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Study on Hybrid Air- Conditioning System Using Natural Ventilation in Office Space

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ABSTRACT

In Japan, the middle-height and high-rise office buildings have usually been designed with a fixed (permanently-close) window, in order to reduce the infiltration of outdoor air. Here in this research, semi-open indoor space of office design is proposed by introducing the hybrid air-conditioning (cooling) system, utilizing the natural ventilation for controlling the indoor climate in spring and fall seasons. The natural ventilation is applied in the hybrid air-conditioning system, a system which combines the wind induced cross-ventilation with the conventional (usual) mechanical air-conditioning system. During the middle season (spring and fall), it is reasonable to supply the outdoor air into the indoor office space directly, because during that period the fresh outdoor air is thermally comfortable enough. Furthermore, this can contribute to improve the indoor air quality and save the air-conditioning energy consumption.

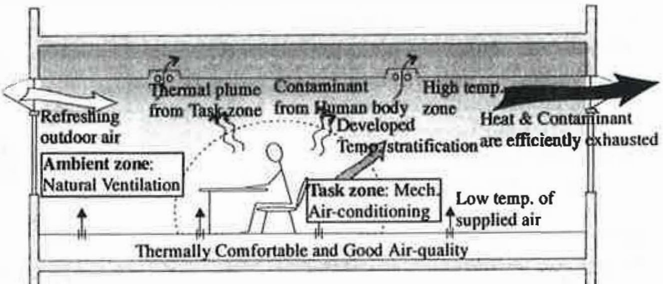
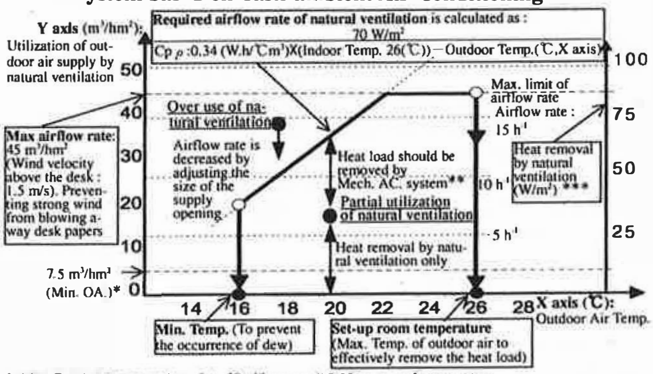


Fig. 1 Concept of Hybrid Air-conditioning (cooling) System based on Task-ambient Air-conditioning



* Min. OA (Minimum outdoor air) = 30m³/h.person X 0.25 person/m², assumed as the same condition as in conventional air-conditioning
** Indoor heatload, which cannot be removed wholly by natural ventilation only
*** Under the condition of outdoor air temp. = 20℃ (X axis)

Fig. 2 Heat removal by Natural Ventilation in Hybrid Air-conditioning (cooling) System

The hybrid air-conditioning is applied in an office space based on the task- ambient air conditioning concept. As the concept of hybrid air conditioning is illustrated in Fig. 1, fine-control of mechanical air-conditioning is applied in the task (occupied) zone and furthermore, the ambient zone air-conditioning is intended to be controlled roughly with the aid of the natural ventilation.

Based on CFD (Computational Fluid Dynamics) simulation results, the ventilation efficiency of the hybrid air-conditioning system is analyzed by investigating the Age of air (Kato et al. 1988) of the fresh outdoor air and the Contribution ratio (NOTE 1) of supply opening of the natural ventilation (Kato et al. 1992). Then the improvement of the air quality at each point of the office room is examined, based on the Perceived air quality concept (Fanger 1989).

The research of the hybrid air-conditioning (cooling) system using natural ventilation (from here on, will be referred as hybrid air-conditioning) in an office space is intended to analyze the energy conservation effect, the effect to human perceived air quality and, the zone averaged heat and contaminant flow rate between task and ambient zones (NOTE 2). Finally, the efficiency and feasibility of the hybrid air-conditioning system itself are analyzed.

KEYWORDS
Hybrid Air-conditioning, Natural Ventilation, Ventilation Efficiency, Perceived Air Quality, Task-Ambient Air-conditioning, CFD.

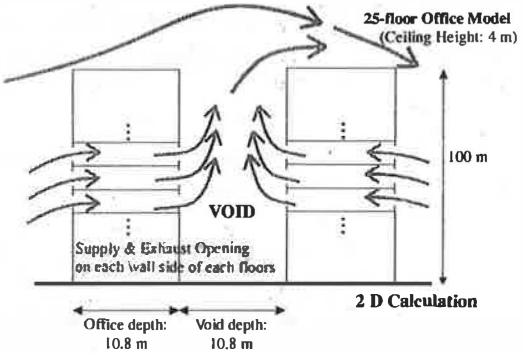


Fig. 3 Flow Network Analysis of Ventilation Circuit

INTRODUCTION

As the concept of hybrid air-conditioning is described in Fig. 1 and 2, energy conservation, comfortable thermal environment and also good air quality can be achieved reasonably by applying hybrid air-conditioning system.

Different from the conventional air-conditioning system, in which the fresh outdoor air is supplied at minimum airflow rate (Min. OA.), hybrid air-conditioning can improve the indoor air quality significantly, since fresh outdoor air is supplied in a large volume. In order to investigate the improvement of indoor air quality, the perceived air quality calculation based on the olf and decipol unit, is carried out.

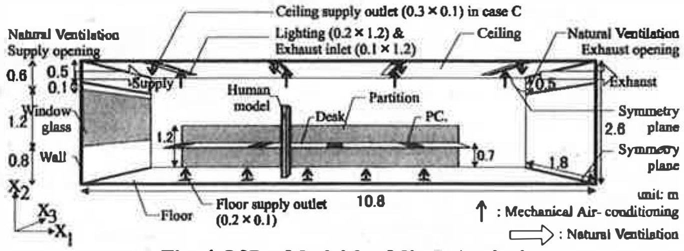


Fig. 4 Office Model for Micro Analysis
(by CFD, half space of the symmetrical room)

Since the hybrid air-conditioning is applied based on the concept of task-ambient air conditioning, in which mechanical air-conditioning (Mech. AC) is used for the task (occupied) zone and natural ventilation is applied in the ambient zone, the analysis of the mixing and diffusion ratio between the air of the natural ventilation and that of Mech. AC. becomes important. This analysis is carried out based on the CFD calculation of the contribution ratio of the natural ventilation supply opening (NOTE 1).

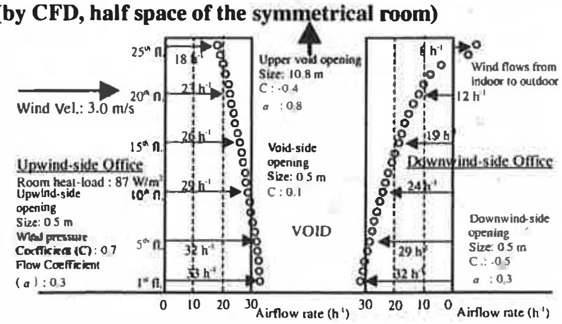


Fig. 5 One of the Macro Analysis Results

Finally, in order to evaluate the efficiency of the supply of outdoor air into the task zone and the removal of heat, contaminant by the hybrid air-conditioning, zone analysis of outdoor air, contaminant and heat flow rate based on CFD is carried out (NOTE 2) (Murakami et al. 1994).

Through these analysis, the air-conditioning energy consumption, thermal comfort and air quality by applying the hybrid air-conditioning system in an office space are investigated.

OFFICE MODEL (Fig. 3, 4)

The office is modeled as a 25-floor office building, with a void space inside (Fig. 3). Due to the wind pressure difference, the wind induced cross-ventilation is generated across the office room at each floor.

In CFD analysis, the width of the calculation area is set as half of the 3.6 m office module (1.8 m), considering the symmetrical configuration (Fig. 4). In case of hybrid air-conditioning, it is modeled that the outdoor air flows in from the opening above the window (0.5 × 1.8 m; Fig. 4, left), and is exhausted through the opening at the other side (0.5 × 1.8 m; Fig. 4, right), while the Mech. AC. is still operating. The cooled-air of the Mech. AC. is supplied by 5 units of floor supply-outlet (0.1 × 0.2 m) and then exhausted through 4 units of ceiling exhaust-inlet (0.1 × 1.2 m).

In case D (Table 1), the size and position of supply outlets and exhaust inlets of the usual (mechanical) air-conditioning are the same as those of the Mech. AC. in the hybrid air conditioning (Case A, E). The cooled-air is supplied by the floor supply-outlets (4 units of ceiling supply-outlet for case C), then exhausted through 4 units of ceiling exhaust-inlet.

For the zone analysis of heat and contaminant flow rate inside the office space (NOTE 2), the space is divided into 3 zones, i.e. Perimeter, Ambient and Task zones. Perimeter zone is an area between the window-glass wall (source of solar radiation heat) and 0.9 m inside from the wall. Task zone is considered as an area within 1.5 m height from the floor, where people are sitting and doing office activities. And the residual area which is not included in both of task and perimeter zones, is considered as the ambient zone.

Table 1 Analyzed Cases (Floor area used for calculation = 19.4 m^2)

Cases	Cooling Load W	Air-conditioning system	Mechanical Air-conditioning (Mech. AC) Supply outlet position	Burden Heat Load W	Air change rate m^3/h	Outdoor air supply m^3/h	Supply outlet vel. m/s	Temp ℃	Natural Ventilation Burden Heat Load W	Airflow Rate m^3/h	Supply Vel. m/s	Temp ℃
A	1700	Hybrid	Floor	579	194	0	0.5	19	1121	505	0.16	21
B				546					1154		0.78	
C		Usual air-conditioning	Ceiling	1700	583	150	1.4		none			
D			Floor				1.6					
E		Hybrid		1158	542	0	1.2		542	505	0.16	24
F		Natural Vent.	none						1700			18

Table 2 Boundary & Calculation Condition for CFD

Supply (opening) outlet	U_{in}: Airflow velocity $k_{in} = 3/2\,(U_{in} \times 0.05)^2$ $\varepsilon_{in} = C_{\mu} \times k_{in}^{3/2} / l_{in}$ $l_{in} = 1/7 \times$ supply outlet/ opening width
Exhaust (opening) inlet	Vel.: Mass balanced K_{out}, ε_{out}: Free- slip.
Wall	Vel.: Standard log-law, Symmetry plane: Free slip. Heat : Fixed convective heat load
CFD grid Points	76 (X_1)× 18 (X_2)× 16 (X_3) = 21888 (Case A) 76 (X_1)× 18 (X_2)× 18 (X_3) = 24624 (Case D, E, F)

Table 3 Cooling load (Floor area used for calculation =19.4 m^2)

Heat Sources (W)	Lighting (4 units)	Solar heat (window)	PC. (4 units)	Human body (1 body)	Floor (human body)	Total	Per- floor area (W/m^2)
Heat load	400	225	800	55	220	1700	87.5

Table 4 Sensory Pollution Load , olf

olf sources	Floor	Wall & Ceiling	Desk.	Human model (non- smoking)	PC	Partition	Total
Rate	0.05 olf/m^2	0.01 olf/m^2	0.14 olf/m^2	1 olf/person	0.3 olf/unit	0.01 olf/m^2	8.16 olf

METHODS

Flow Network Macro Analysis (Fig. 3, 5)

First in Flow Network analysis, the airflow rate of the natural ventilation is estimated. The airflow rate of the natural ventilation is calculated based on the network analysis of ventilation-circuit. The ventilation-circuit network is calculated under various different conditions of wind direction, velocity and supply opening area. One of results is illustrated in Fig. 5, where wind velocity: 3.0 m/s, supply opening: 0.5 m and the wind direction : 90^0 normal to the window surface. Air change rate by the natural ventilation on the middle height floors are predicted to be between 15- 30 h^{-1}; at the low floors, they are about 30 h^{-1} and at the top floors, they are between 5- 20 h^{-1}.

For Micro analysis (CFD simulation), considering the combination-use with the Mech. AC. and also the expectable air change rate by natural ventilation in usual condition, the airflow rate by the natural ventilation is set at 10 h^{-1} for all cases (shown as 505 m^3/h in Table 1).

Micro Analysis /CFD Simulation (Fig. 4)

Indoor airflow and thermal distribution field are calculated based on 3-D CFD simulation, using the standard k- ε model. The office cooling load is given from the pre-calculation using radiative-convective coupled simulation where the distribution of convective heat transfer coefficient is assumed. The boundary condition for CFD simulation is shown in Table 2 and the heat load of the office area used for calculation is described in Table 3.

Based on the CFD results, the age of air of the fresh outdoor air and the contribution ratio of natural ventilation supply opening are analyzed. Then due to the generation of the Sensory pollution load (olf) from the office equipment, human body, floor, etc. (Table 4), the Perceived air quality (decipol) of the office room is also analyzed.

Finally based on CFD, zone analysis of outdoor air, contaminant and heat flow between perimeter, ambient and task zone of the office space, is then carried out.

ANALYZED CASES (Table 1)

4 cases (Case A, D, E, F) are selected to be presented in this paper, and are analyzed comparatively with case A (Hybrid air-conditioning) as the basic case (Table 1).

In case of hybrid air-conditioning (Case A, E), the fresh outdoor air is supplied directly by the natural ventilation only. All the cooled-air supplied through the floor supply-outlets of Mech. AC. is the returned-air which is recirculated through the ceiling exhaust-inlets.

In case D, the air supplied by the floor supply-outlets is the mixture of the returned-air (from the ceiling exhaust-inlets) and 150 m^3/h (Min. OA.: 30 m^3/h.person for 5 persons) of fresh outdoor air. This condition is assumed as the same as the common air-conditioning system.

For the case of cooling by natural ventilation only (in case F), the supply and exhaust openings of the natural ventilation are the same as those of case A and E (hybrid air-conditioning).

RESULTS

Characteristics of Flowfields (Fig. 6)

In case A (Fig. 6(a)), the outdoor air flows into the office room, and is directly slanted towards the floor surface ($Ar = g \cdot \beta \cdot \Delta \Theta \cdot L / U^2 = 4.77$, NOTE 3), then drops down. But in case E (Fig. 6(c)) since the negative buoyancy effect is decreased (Ar= 2.20), the outdoor air is flowing farther and deeper inside the room (reaching the human model). On the other hand, due to a large temperature difference, the negative buoyancy effect becomes much stronger in case F (Fig. 6(d), Ar= 6.04). The outdoor air drops down to the floor directly after flowing in from the supply opening, then flows along the window-wall surface and the floor surface, until reaching the other side wall.

In case D (Fig. 6(b)), the jet cooled-air supplied from the floor supply-outlets of Mech. AC., rises up straightly toward the ceiling, then is re-circulated around the area between the supply-outlets.

Characteristics of Temperature Fields (Fig. 7)

In case A (Fig. 7(a)), due to the strong effect of natural ventilation, the temperature around the left side area of human model is almost uniformly distributed (26℃). But on the right side area, the effect of floor supply-outlets of the Mech. AC is pretty strong that a temperature stratification is formed.

On the other hand in case E (Fig. 7(c)), due to the strong effect from the floor supply-outlets of Mech. AC (airflow rate is increased), temperature stratification is formed all over the office room. In case F (Fig. 7(d)), temperature stratification is produced also. Here in case F, the task zone average temperature is 2.8℃ lower than the room average temperature under perfect-mixing condition. This indicates that supplying low temperature of outdoor air by the natural ventilation can effectively cool down the task zone, especially the lower level area of the room.

In case of usual air-conditioning (Case D, Fig. 7(b)), due to the strong effect of floor supply-outlets, temperature stratification is produced, particularly in the ambient zone. But compared to other cases, the lowest average temperatures are observed, indicating that an efficient air-conditioning is realized.

The Age of Air (Fig. 8)

Here, the age of air means the age of

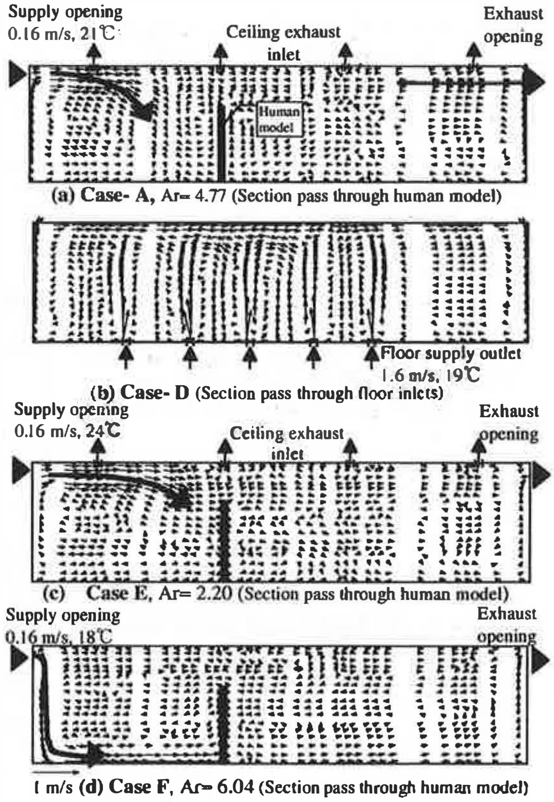
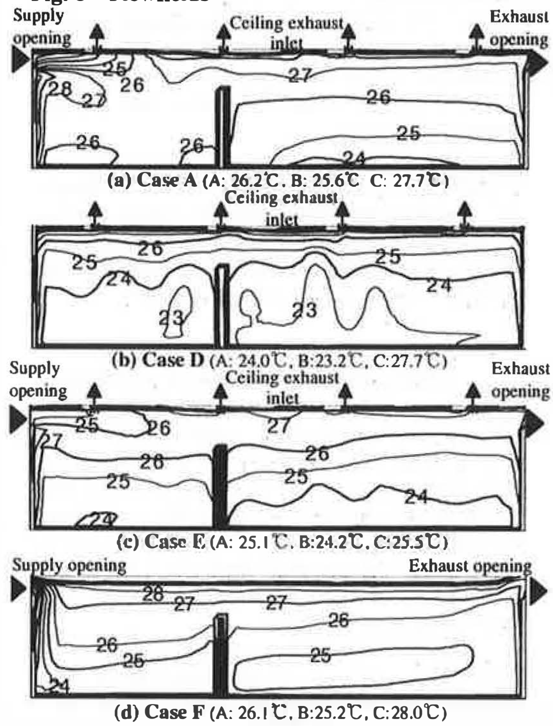


Fig. 6 Flowfields



A: Room average temperature, B: Task zone average temperature
C: Room average temperature under perfect mixing condition

Fig. 7 Temperature Distribution (Section through human model) the fresh outdoor air, which is supplied by natural ventilation. The calculation results in Fig. 8 are non-dimensionalized by the Nominal Time Constant, τ_n (the inverse of air exchange rate). τ_n for cases of A, E, F is 360 sec, and for case D is 1200 sec.

In both cases of A and E (Fig. 8(a), (c)), the age of air around the human model is between 0.8- 1.0 and around the upper-right side area of the human model is higher (about 1.2). In case of natural ventilation only (Case F, Fig. 8(d)), the low age of air (0.8>) is distributed only around the lower part near the floor surface.

The area around the human model in case D (Fig. 8(b)), the age of air of fresh outdoor air which is supplied from the floor supply-outlets (Min. OA.), is almost uniformly distributed (0.9). Since the nominal time constant of case D is 3 times bigger than those of cases of A, E and F, the age of fresh outdoor air in case D becomes so high (old), that the quality of the outdoor air is worse than that of cases of A, E, F.

Perceived Air Quality (Fig. 9)

The decipol (Perceived air quality) value distributed around the human model in cases of A, E, F (Fig. 9(a), (c), (d)), is almost the same, around 0.5 (corresponds to 8.2 percent of dissatisfied). Since the fresh outdoor air always reaches the area around the left side of the human model, the decipol value of that area is lower than that of the right side area of the human body, but in case F the low value (0.5 decipol) is distributed almost in the entire office room.

Since in the case of usual air conditioning (Case D, Fig. 9(b)), the fresh outdoor air is supplied in Min. OA., the decipol value becomes much higher (1.9 decipol, 24.8 percent of dissatisfied) than the values of cases of hybrid air-conditioning and natural ventilation only. It is suggested that Min. OA.. (30 m^3/h.person) of fresh outdoor air supply is not enough to improve the perceived air quality of the office room at all.

Outdoor Air Flow Rate Between Zones (Fig. 10, 11)

The outdoor air here means the air, which is supplied by the natural ventilation. The outdoor air flow rate between zones (Perimeter, task and ambient zone) are predicted based on the contribution ratio of the natural ventilation supply opening calculation, as shown in Fig. 10 (NOTE 1).

In case A and E (Fig. 10(a), (b)), the

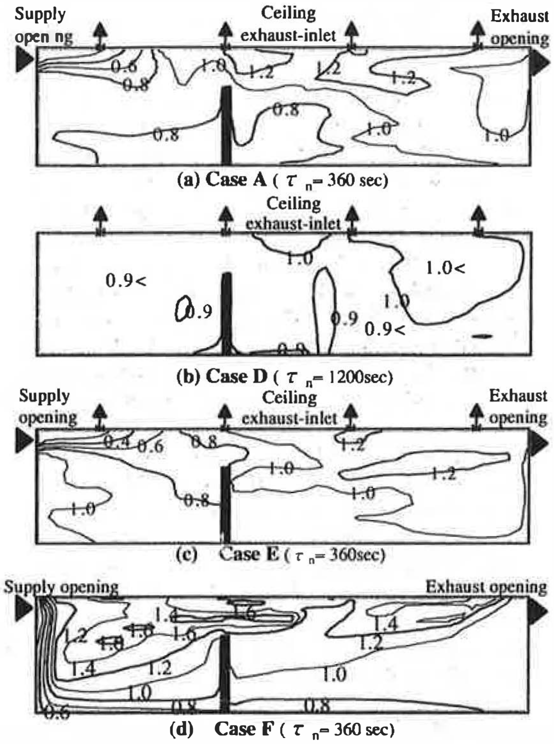


Fig. 8 The Age of Air, (nondimensional)
(Section pass through human model)

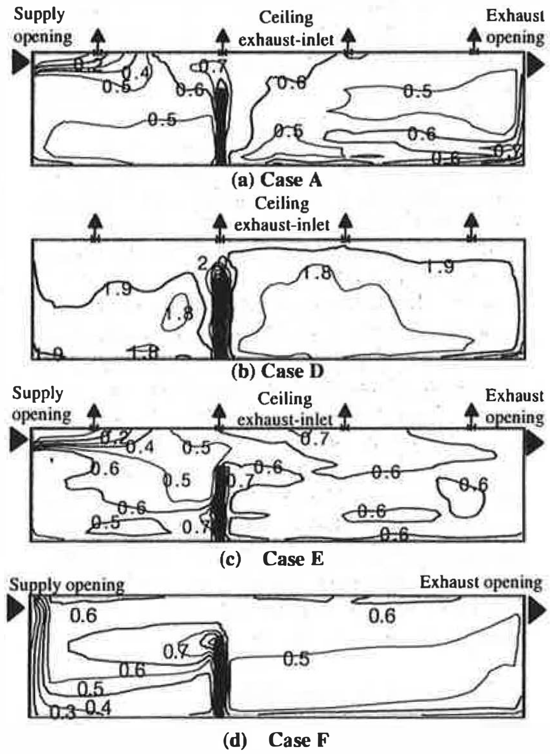


Fig.9 Decipol Distribution (Section through human model)

outdoor air which is flowing from the supply opening of the natural ventilation is circulated from perimeter, ambient, and then task zone (clockwise turn). (As shown in the Fig. 10 (a), the airflow rate of recirculation is larger than the supply airflow rate). In case F (Fig. 10(c)), due to strong effect of negative buoyancy of the outdoor air, the flow direction is reversed (counter-clockwise, Perimeter→Task→Ambient).

In case A, in which the outdoor air flows into the perimeter zone, 30% of it is exhausted through ceiling exhaust-inlets of Mech. AC. and 70% through the exhaust opening of the natural ventilation. But in case E, since the negative buoyancy effect is weaker (than that of case A), the amount of the outdoor air which is flowing into the task zone, is decreasing and tend to be directly exhausted through the ceiling exhaust-inlets of Mech. AC. (increased by almost 50%). In case F, the outdoor air is flowing into the task zone in the biggest volume (753 m^3/h), compared to other cases. It indicates that the low temperature of outdoor air supply by natural ventilation, can introduce the outdoor air into the task zone effectively and improves the air quality there.

In Fig. 11 (a), the relationship between the outdoor air temperature and the amount of outdoor air which flows from task to ambient zone (T→A), is shown. The amount of T→A outdoor air flow is decreasing (or the flow direction is changing/ reversing), when the outdoor air temperature becomes higher. This is because the effect of negative buoyancy is weakened. Based on this relationship, the flowfields of the outdoor air (from the natural ventilation) in the office room can be differentiated into 3 patterns (Fig 11(b), (c), (d)), i.e. strong, medium and weak negative buoyancy effect on the indoor airflow field.

Due to strong effect of negative buoyancy (Fig. 11(b)), the cool outdoor air drops down in the perimeter (P) zone, rises up in the task (T) zone and then is exhausted in the ambient (A) zone.

In case of medium negative-buoyancy effect (Fig. 11(c)), the outdoor air is flowing into the task zone from the upper part of task zone (ambient zone). But in case of weak negative buoyancy effect (Fig. 11(d)), the warm outdoor air tend to just pass through the ambient zone and then being exhausted directly, almost without flowing into the task zone.

Contaminant Transport Rate Between Zones (Fig. 12)

The contaminant here means the sensory pollution load, olf (cf. Table 4). The total generated contaminant load is standardized to 1000.

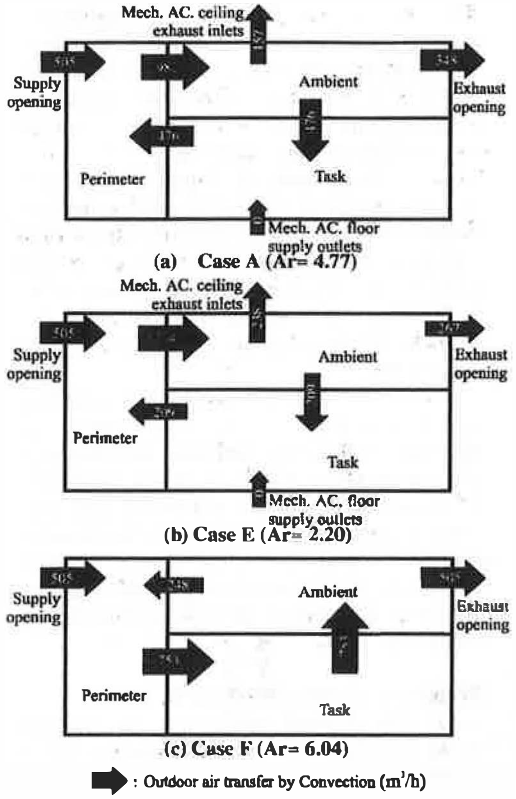


Fig. 10 Outdoor Air Flow Rate Between Zones

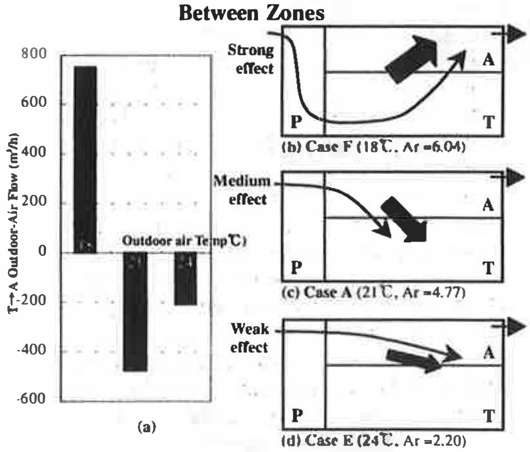


Fig. 11 Negative Buoyancy Effect on Flowfields

In case of hybrid air-conditioning (Case A, E; Fig. 12(a), (c)), the contaminant from task zone is mostly transported to perimeter zone, then move into the ambient zone and finally exhausted through the exhaust opening of the natural ventilation and pretty large part is re-circulated back to task zone by the Mech. AC. It is shown that the return-air mechanism of the

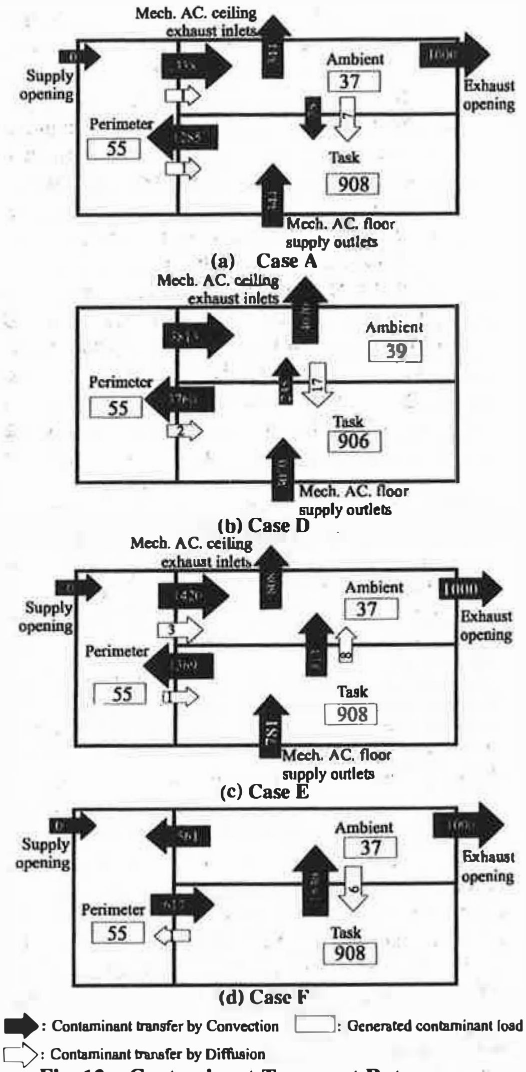


Fig. 12 Contaminant Transport Rate Between Zones

Mech. AC. in the hybrid air-conditioning cases, worsen the office air quality, particularly that in the task zone.

In case F (Fig. 12(d)), the contaminant load from task zone is mostly transported to ambient zone, then exhausted through the exhaust opening of natural ventilation.

In case D (Usual air-conditioning, Fig. 12(b)), due to the return-air mechanism of the Mech. AC., the circulated contaminant between zones are much bigger than those of cases of A, E, F. This caused the air quality of the usual mechanical air-conditioning case to become much worse, than that of the cases in which natural ventilation is applied.

Heat Flow Rate Between Zones (Fig. 13)

The analysis of heat flow rate between each zones are calculated based on the task zone average temperature as shown in Fig. 13 (NOTE 4).

In case A (Fig. 13(a)), 52% (409 W) of the heat generated in task zone (784 W) is absorbed by the cooled-air from the floor supply-outlets of Mech. AC. Then, the rest is flowing into perimeter and ambient zone almost equally. Almost the same in case F (Fig. 13(d)), the heat from task zone is also equally divided, flowing into perimeter and ambient zone.

In case A, the removal of the indoor heat by natural ventilation (1121 W, as it is calculated from the amount of heat removed by the supply and exhaust-opening of Natural ventilation, as it is shown in Fig. 13 (a)) is twice bigger than that by Mech. AC (579 W, as it is calculated from the amount of heat removed by the supply-outlet and exhaust-inlet of Mech. AC.). It shows that applying natural ventilation in the hybrid air-conditioning system can decrease the air-conditioning energy consumption. Almost 66% of the whole cooling heat-load which is generated in the office room, is removed by natural ventilation.

But in case E (Fig. 13(c)), due to an increase of the Mech. AC airflow rate (583 m^3/h, 2.2 times of that of case A) and the outdoor air supply temperature (24℃; 3℃ higher than that of case A, cf. Table 1), the heat removal by natural ventilation (542 W) is decreasing, compared to that of case A. The value becomes only half of the Mech. AC. heat removal (1158 W).

In case D (Fig. 13(b)), due to the outflow of cool air from task zone to ambient zone, the heat load of task zone is increased by 211 W (173W+38W, as it is calculated from the amount of heat flow rate between ambient and task zone, as it is shown in Fig. 13(b)). It is indicated that the heat load removed by the floor supply-outlets is also increasing.

CONCLUSION

(1). In applying the hybrid air-conditioning system using the natural ventilation, it is important to consider the effect of negative buoyancy, due to temperature difference between the outdoor and the inside room. This negative buoyancy effect is strongly related to the entire indoor air flowfields of the office space.

(2). Applying hybrid air-conditioning system using natural ventilation does not only keep the task (occupied) zone thermally comfortable, but also can contribute to the air-conditioning energy conservation (almost about 66% saving in maximum (in Case A)).

(3). In the case of hybrid air-conditioning, the age of air of the fresh outdoor air around the human model is significantly young, compared to that of the usual air-conditioning system case. This means that the hybrid air-conditioning system using natural ventilation is very efficient from the view point of ventilation efficiency.

(4). Introducing a large volume of fresh outdoor air in hybrid air-conditioning which produced relatively young age of outdoor air in the task zone, can also decrease the Percent of Dissatisfied (Improvement of the Perceived air quality).

(5). Applying hybrid air-conditioning system can improve the air quality of the office room greatly, since the re-circulated amount of contaminant transport rate between zones in the office room becomes smaller, compare with the usual air-conditioning system.

NOTE 1:
The contribution ratio of supply opening means the mixing ratio between the air supplied from the supply opening in concerns, with the air from other supply openings. It is estimated by calculating/ measuring the tracer gas concentration on each points of the room, in which the tracer gas is initially generated (injected) into the air of the supply opening in concerns.

NOTE 2:
Zone analysis of outdoor air, contaminant, heat flow rate between zones is carried out based on the airmass, contaminant and heat flux between zones (perimeter, ambient and task zones) from CFD calculation result (micro analysis) (Murakami et al. 1994)

NOTE 3:
The Archimedes number (Ar) is calculated based on the flowing-in outdoor air supplied from the supply opening of the natural ventilation.

$$Ar = g.\beta.\Delta\Theta.L/U^2 \quad (1)$$

g :	gravitational acceleration (= 9.8)	[m/s²]
β :	coefficient of volumetric expansion(=1/30)	[℃⁻¹]
$\Delta\Theta$:	temperature difference between the natural ventilation supply and exhaust opening	[℃]
L :	vertical-width of the natural ventilation supply opening (= 0.5)	[m]
U :	outdoor air supply velocity	[m/s]

NOTE 4:
The analysis of heat flow rate between zones are calculated based on the equation (2).

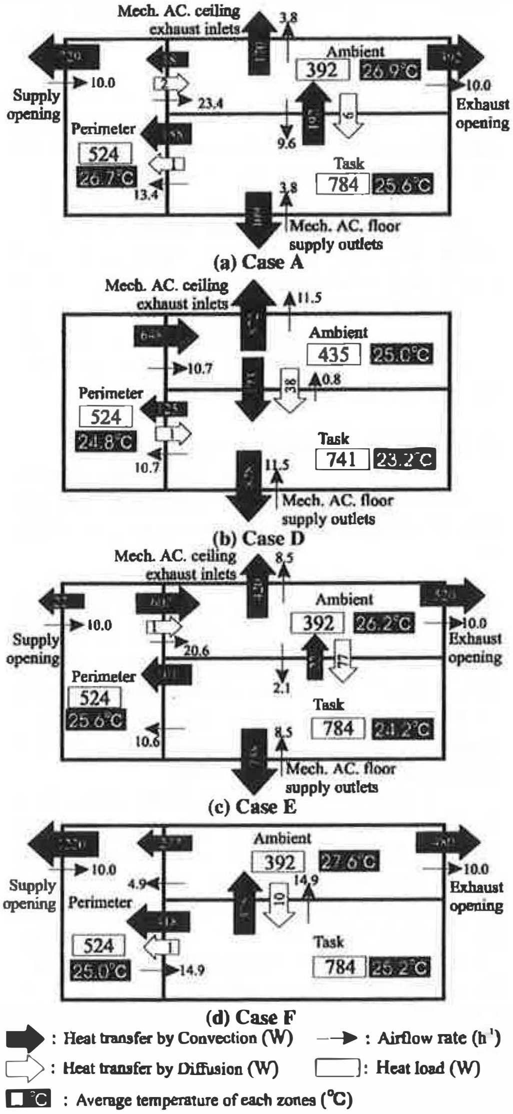
Heat flow rate = Heat flux by convection + Heat flux by diffusion (2),

where each heat flux is calculated as:

$$\text{Heat flux by convection} = \sum U_n \cdot Cp \cdot \rho \cdot (T - Tref) \cdot A \quad (3)$$

$$\text{Heat flux by diffusion} = \sum -\rho\, Cp \frac{(\nu + \nu_t)}{\sigma_\theta} \frac{\partial T}{\partial x} \cdot A \quad (4)$$

Cp :	specific heat at constant pressure	[J/kg℃]
T :	temperature	[℃]
Tref:	temperature reference (the average temperature of task zone)	[℃]
A :	area	[m²]
ν_t :	turbulence viscosity	[m²/s]
ν :	molecular viscosity	[m²/s]
σ_θ :	turbulent Prandtl number(= 0.9)	[-]



Outward arrow indicates outflow of heat from the room/ system (Heat sink)

Fig. 13 Heat Flow Rate Between Zones (NOTE 4)

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