2000 Elsevier Science Ltd. All rights reserved.
Air Distribution in Rooms, (ROOMVENT 2000)
Editor: H.B. Awbi

13340

ROOM AIR DISTRIBUTION AND COOLING LOAD OF HYBRID AIR-CONDITIONING SYSTEM UTILIZING NATURAL VENTILATION IN AN OFFICE

Shinsuke Kato[1], Shuzo Murakami[1], Hyunjae Chang[2], Tomoyuki Chikamoto[3], Taeyeon Kim[1]

[1]Institute of Industrial Science, University of Tokyo, Tokyo, Japan
[2]Graduate School, University of Tokyo, Tokyo, Japan
[3]Environmental Eng. Group, Nikken Sekkei, Ltd., Tokyo, Japan

ABSTRACT

The characteristics of a hybrid air-conditioning system, utilizing natural and mechanical ventilation, is investigated in an office setting. The characteristics of the indoor environment are examined with CFD(Computational Fluid Dynamics) simulation under various conditions of incoming outdoor air. The control of the room air conditioning system (VAV system) is included in the calculation through changing the supply air volume to keep the task zone's temperature at a target temperature. When the temperature of the inflow outdoor air rises, the inflow jet flows deeper into the room and mixes well with the indoor air. A vertical temperature gradient appears at the task zone with the rise in the incoming outdoor air temperature. The distribution of the age of air is under the strong influence of flow fields. The cooling load of the mechanical AC system increases with the increase of the outdoor air temperature and the decrease of the incoming outdoor air volume. From the viewpoint of energy conservation, the window opening (vertical width of the opening) should be made larger for ease of outdoor air intake.

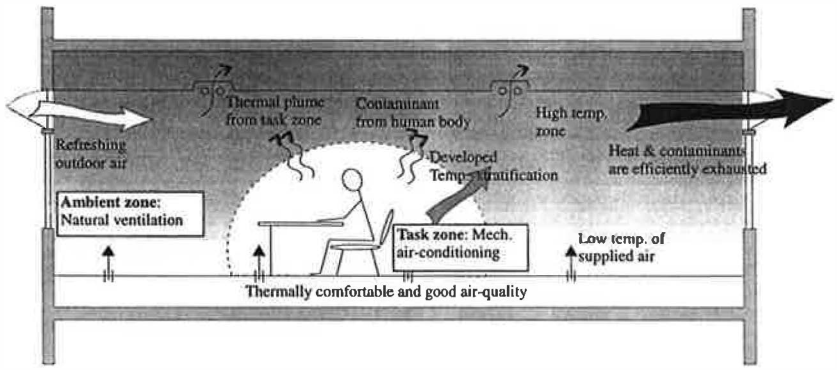


Figure 1: Concept of hybrid air-conditioning (cooling) system based on task-ambient air-conditioning

KEYWORDS

CFD, VAV, Task/ambient air-conditioning, Relative humidity, Age of air, Cooling load

INTRODUCTION

The hybrid air-conditioning (AC) system is based on the concept of utilizing both natural and mechanical ventilation. In the intermediate seasons of Spring and Autumn, since outdoor air has the ideal temperature and humidity, a large volume of outdoor air may be allowed into the room directly (without conditioning) to maintain a comfortable indoor air-temperature and humidity. Hybrid air-conditioning is applied to an office space based on the task-ambient air conditioning concept. The concept of hybrid air conditioning is illustrated in Figure 1. The indoor environment is controlled with the aid of natural ventilation and fine-control of mechanical air-conditioning being applied to the task (occupied) zone. The application of this hybrid AC system, is expected to improve the indoor air quality (IAQ) on the one hand, and lower the energy consumption on the other.

METHODS

In the office setting shown in Figure 2, flow, temperature, humidity and age of air are numerically analyzed based on a 3-D CFD (Computational Fluid Dynamics) simulation, using the standard k-ε model. The numerical method was confirmed as a valid tool by comparing the results of room airflow simulation with experimental results (Kato, *et al.*, 1989, 1999). The width of the calculation area is set

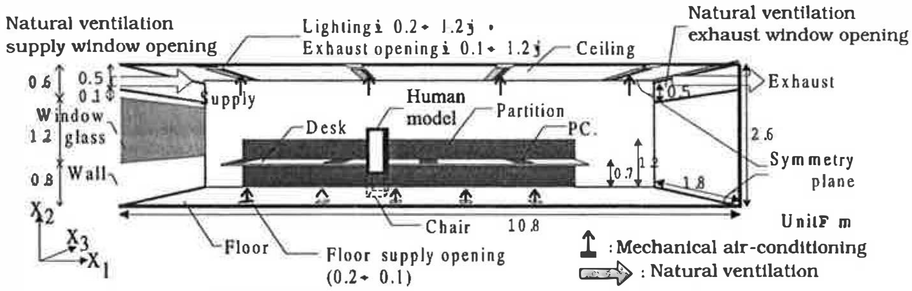


Figure 2: Office model for analysis
(by CFD, half space of the symmetrical office module)

TABLE 1
BOUNDARY & CALCULATION CONDITIONS FOR CFD

Supply opening	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 opening width T_{in} : T_{supply} AH_{in}: AH_{supply}
Exhaust opening	Vel.: Mass balanced k_{out}, ε_{out}, T, AH : Free slip
Wall	Vel.: Standard log-law Heat : Fixed convective heat transfer coefficient Absolute humidity [Human model, Floor : Fixed flux Other walls : Gradient of AH = 0]
Turbulence model	Standard k-ε model (Viollet model for ε eq.) with wall function for wall boundary
CFD grid points	88 (X_1) × 17 (X_2) × 15 (X_3) = 22440

TABLE 2
HEAT SOURCES IN OFFICE MODEL
(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)
Cooling load	400	225	800	55(SH) 28(LH)	220(SH) 111(LH)	1700(SH) 139(LH)	87.5(SH)

Nomenclature

U : velocity [m/s]
k : kinetic energy [m^2/s^2]
ε : kinetic energy dissipation rate [m^2/s^3]
l : specific length scale [m]
T : temperature [°C]
AH : absolute humidity [kg/kg]
SH : sensible heat [W]
LH : latent heat [W]
Suffix [in : inlet
out : outlet]

[redacted]

at half of the 3.6m office module (1.8 m), in consideration of the symmetrical configuration. Hybrid air-conditioning is modeled as the outdoor air flows into the room from the upper opening of the window (0.5 × 1.8 m; Fig. 2, left), and is expelled through the opening at the other side (0.5 × 1.8 m; Fig. 2, right), while the mechanical AC is still operating. Return air through the four ceiling exhaust openings (0.1 × 1.2 m) is conditioned by the mechanical AC system, and then supplied again from the five floor supply openings (0.1 × 0.2 m).

The CFD simulation leads to a condition where the average task zone temperature is kept at the target temperature of 26°C by changing the supply air volume of the air-conditioning system, but keeping the supply air temperature constant. The absolute humidity of the incoming outdoor air was set at a constant value of 0.0093 [kg/kg] in each case, considering that the absolute humidity of outdoor air is almost constant throughout the day. The relative humidity of the supply air for mechanical air-conditioning was set at a constant value of 80% at 19°C. The boundary and calculation conditions for the CFD simulation are summarized in Table 1. The sensible and latent heat generation rates are shown in Table 2. One simplified body shape was arranged in the center of the office to investigate the flow field around it. The effects of the other 4 human bodies were modeled as sensible heat and latent heat sources which are given on the floor.

CASES ANALYZED

Cases analyzed are grouped into three as shown in Table 3. Group I is composed of cases which change the outdoor air temperature of the cross ventilation, but keep the airflow volume constant. Group II is composed of cases which change the outdoor airflow rate, but maintain the outdoor air temperature. Finally, Group III is composed of cases which change the opening area of the window for natural cross ventilation. Case C in Group I also belongs to the other two groups II and III, and is used as the basic case.

TABLE 3
ANALYZED CASES

Cases		Heat gener-ation [W]	Mechanical AC					Natural ventilation						Room temp. assum-ing perfect mixing [°C]
			Cooling load [W] (simulated result)	Supply vel. [m/s] (simulated result)	Supply air vol.[1] [m³/h] (simulated result)	Supply air temp.[°C] /Relative humidity[%]	Out-door air vol. [m³/h]	Cooling load [W] (simulated result)	Supply vel. [m/s]	Air exhange rate [h^{-1}]	Temp.[°C] /Relative humidity[%]	Window opening area (vertical width) [m]	Ar. number[2] of inflow	
Group I	A	1700	0	0	0	19.0/80	0	1700	0.16	10	18.8/69	0.5	-4.83	28.6
	B		146	0.07	25.2			1554			19.5/66		-4.36	28.8
	C		366	0.31	111.6			1334			21.0/60		-3.36	28.7
	D		627	0.49	176.4			1073			22.5/54		-2.35	28.9
	E		843	0.66	237.6			857			24.0/50		-1.34	29.1
Group II	F		882	0.64	230.4			818	0.08	5	21.0/60		-13.42	30.3
	G		563	0.39	140.4			1137	0.12	7.5			-5.97	30.0
	H		129	0.09	32.4			1571	0.20	12.5			-2.15	28.4
	I		0	0	0			1700	0.23	15			-1.49	27.6
Group III	J		491	0.46	165.6			1209	0.39	10		0.2	-0.21	27.9
	K		545	0.55	198.0			1155	0.78	10		0.1	-0.03	27.5

1) Supply air volume in VAV mechanical air-conditioning is simulated through the CFD so as to achieve the average temperature of 26°C within the task zone.

2) Archimedes number of inflowing outdoor air is defined as $Ar = g \cdot \beta \cdot \Delta\Theta \cdot L/U^2$, where, g is the gravitational acceleration [m/s^2], β is the coefficient of volumetric expansion (=1/300) [$°C^{-1}$], ΔΘ is the temperature difference between the inflow outdoor air and the temperature at the task zone [°C], L is the vertical width of the window opening [m] and U is the velocity of the inflow outdoor air [m/s].

RESULTS

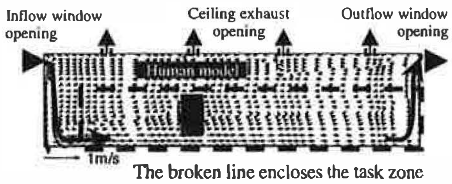
Only the CFD simulation results of Case A, Case C and Case E in Group I are shown in this paper on account of limited space.

Flow Fields

The flow fields of three cases in Group I are shown in Figure 3. The vertical sections including the human model are shown. The inflow outdoor air temperatures for Case A are set at 18.8°C. The results show a tendency for the inflow air to drop near the window opening because of the negative buoyancy effect in the room. The air flows near the floor at low speed, and rises along the wall surface, exiting through the exhaust opening. There is an anti-clockwise air circulation pattern being observed in the room. In Cases C and E, the inflow outdoor air temperatures are set at 21.0°C and 24.0°C respectively. The inflow air flows deeper into the interior direction and mixes well with the indoor air, as shown in Figure 3. There is a rising stream around the human model which generates heat.

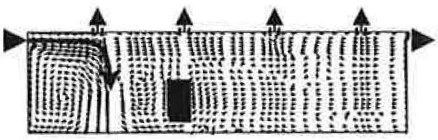
Temperature Fields

Figure 4 shows the temperature distributions. The average task zone temperature is kept exactly at 26°C by utilizing the under-floor VAV AC system. In Case A, there is no need AC cooling energy because the task zone is kept at the target temperature with only natural ventilation. Temperature fields show a tendency to form temperature stratifications inside the room. In Cases C and E, with an increase in outdoor air temperature, inflow air tends to flow further into the room and to mix well with the indoor air. The air volume of the air-conditioning, which is supplied at a constant temperature of 19°C, increases to keep the average temperature of the task zone at 26°C. With the increase in air volume of the AC, the temperature gradient becomes steep at the task zone.

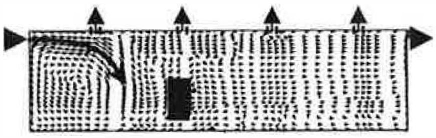


The broken line encloses the task zone

(a) Case A (Ar. number of inflowing outdoor air= -4.83

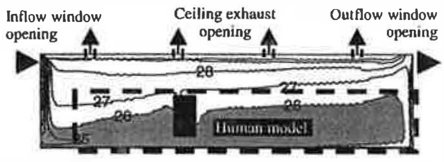


(b) Case C (Ar. number = -3.36)



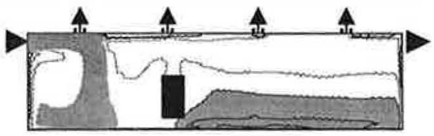
(c) Case E (Ar. number = -1.34)

Figure 3: Flow fields in Group I
(Sections including human model)

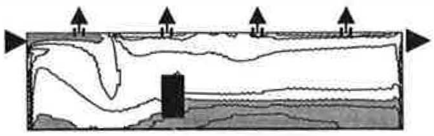


The broken line encloses the task zone

(a) Case A (room averaged temp. =



(b) Case C (room averaged temp. =



(c) Case E (room averaged temp. =

Figure 4: Temperature fields in Group I
(Sections including human model)

Humidity Fields

The relative humidity distributions are shown in Figure 5. Because the absolute humidity distribution is almost constant throughout the whole space, the relative humidity distribution is determined by the temperature fields. With the increase in outdoor air temperature, the supply air volume of the AC system increases and the relative humidity becomes high around the floor.

The Age of Air

The age of the fresh outdoor air was calculated based on the resulting flow fields and is shown in Fig 6. The calculation results are non-dimensionalized by the nominal time constant, τ_n (inverse of the air exchange rate). τ_n for all cases in Group_is 360 sec. The distribution of the age of air is under the strong influence of flow fields. In Case A, young air is observed in the lower part of the room because the fresh outdoor air passes through the task zone; i.e. the lower part of the room. In Case C, the fresh outdoor air mixes well with the indoor air around the perimeter zone, and the age of air becomes old accordingly as it flows deeper into the interior. In Case E, the age distribution becomes complicated. The incoming outdoor air forms a narrow zone with the low age of air (<1) at the right side of the human model.

Characteristics of Cooling Load for Mechanical Air-conditioning System

Figure 7/(a) shows the cooling loads for the mechanical AC system to maintain the average temperature of 26°C only at the task zone in Group_. It also shows the cooling loads with the assumption of perfect mixing condition. Cooling loads are considered only for the sensible heat_because the absolute humidity of conditioned air is higher than the return air. With the increase in the temperature of the inflow outdoor air, the cooling loads in both conditions increase at a constant rate. In comparison to the assumed perfect mixing condition, the cooling load required to maintain this temperature only at the task zone is 470 W lower, which is about one third of the total heat generation

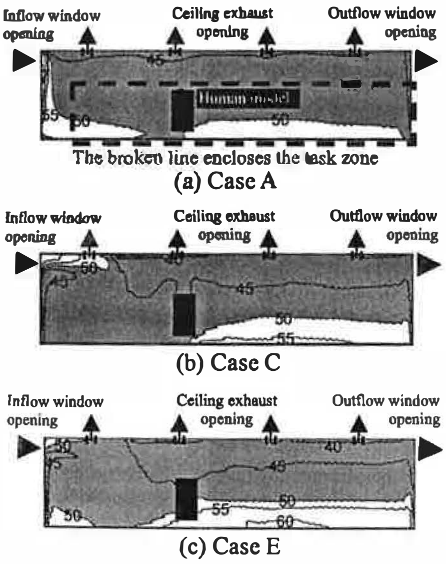


Figure 5: Relative humidity fields in Group_
(Sections including human model)

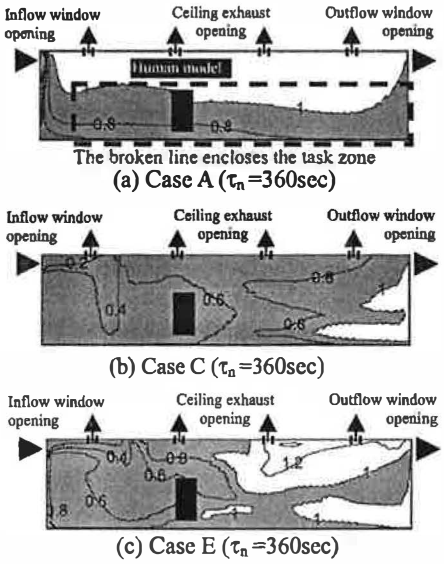
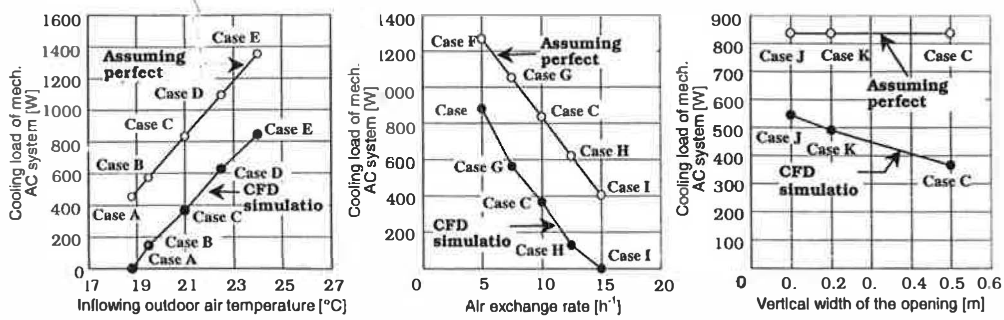


Figure 6: The age of air in Group_
(non-dimensional, sections including human model)



(a) Group I (b) Group II (c) Group III

Figure 7: Cooling load of mech. AC

the task zone can be maintained at 26°C without using the mechanical AC system.

In Group II, as shown in Figure 7/(b), with the increase in the volume of the inflow outdoor air, the cooling load of the air-conditioning system decreases at a constant rate. In comparison to the assumed perfect mixing condition, the cooling load required to maintain this temperature is 450 W lower.

In Group III, as shown in Figure 7/(c), with the increase in the window area (vertical width of the opening), the cooling load of the mechanical AC system decreases at a constant rate in comparison with the assumed perfect mixing case where the cooling load is constant despite the change in the opening area. This can be deduced from the fact that when the window area becomes bigger, the inflow jet of the outdoor air drops down and flows towards the lower part of the room (the task zone), and thus reduces the cooling load of the mechanical AC system needed to achieve the temperature of 26°C at the task zone.

CONCLUSIONS

1. By keeping the air temperature at the task zone constant, when the temperature of the inflow outdoor air increases, there is a tendency for the inflow jet to flow deeper into the room and mix well with the indoor air. These findings can be roughly explained by the Archimedes number of the inflow outdoor air.
2. When the temperature of the inflow outdoor air increases, the air volume from the AC system increases in response, thus creating a big temperature gradient at the task zone.
3. The relative humidity distribution is under the influence of temperature fields because the absolute humidity distribution is almost constant through the whole zone. With the increase in the outdoor air temperature, the relative humidity becomes high.
4. Distribution of the age of the air is strongly influenced by flow fields.
5. The cooling load of the mechanical AC system increases with the increase of the temperature of the inflow outdoor air and with the decrease in the inflow outdoor air volume. In varying the window opening area (vertical width of the opening), it is advantageous to make the opening larger.

References

Kato S., Murakami S., Chang H. et al. (1999). Hybrid Air-Conditioning Based on Natural and Mechanical Ventilation in Office Buildings. *Indoor Air'99* **Vol. 2**, 404 - 409.

Murakami S., Kato S., Ishida Y. (1989). 3-D Numerical Simulation of Turbulent Air Flow in and around Buildings Based on the k-ε Model with Generalized Curvilinear Coordinates. *ASHRAE TRANSACTIONS*, **V.95**, Pt.2.