Development of Zonal Model for Predicting Temperature Distribution inside an Office Room with Hybrid Air-conditioning System

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Abstract

Recently, the hybrid air-conditioning system that used natural ventilation together with a mechanical air-conditioning was proposed. Hybrid air conditioning system is expected to be saving energy and maintain keep indoor thermal environment comfort.

The purpose of this study is to develop a simplified numerical model that predicts distribution of indoor air temperature when the office room is air-conditioned by hybrid air-conditioning. An office room with hybrid air-conditioning system with wind-driven ventilation as natural ventilation was used by object of this study. The temperature distribution can be calculated by solving the equations of air balance and heat balance in each zone. The results of CFD analysis are assumed to be a correct answer in this paper. The air temperature prediction of zonal model was compared with the results of CFD analysis to optimum solution.

Keywords: Hybrid Air-conditioning System, Wind-driven Ventilation, CFD, Zonal Model, Temperature Distribution

Introduction

The natural ventilation is of considerable concern as energy saving method in office room building. Natural ventilation can provide the fresh outdoor air for occupants and maintains acceptable indoor air quality. However, it is difficult to keep indoor thermal environment comfort only by the natural ventilation. Recently, the hybrid air-conditioning system that used natural ventilation together with a mechanical air-conditioning was proposed. In fact, the number of building with hybrid air-conditioning system is increasing. There have been various investigations of hybrid ventilation and hybrid air-conditioning [1].

The hybrid air-conditioning system is expected as air-conditioning system that can save energy and maintain keep indoor thermal environment comfort. Temperature and airflow distributions inside a room with hybrid air-conditioning system are more complex than those distributions inside a room with only mechanical air-conditioning. It is useful for the design of hybrid air-conditioning system to investigate the indoor airflow characteristics. We have been clarifying the indoor airflow characteristic of the office room with hybrid air-conditioning system [2], [3].

Recently, computational fluid dynamics (CFD) is used extensively in the analysis of airflow, temperature and contaminant distributions. The CFD analysis is able to simulate the detailed prediction of airflow and temperature. However, it takes a lot of time complexities and economical costs to analyze with CFD method. CFD cannot be used at basic design step. It is necessary that temperature and airflow distributions inside room with hybrid air- conditioning are predicted by simple methods.

Objectives

The purpose of this study is to develop a simplified numerical model that can predict distribution of air temperature inside an office room with a hybrid air-conditioning system with wind-driven ventilation as natural ventilation. It is necessary to clarify the indoor air characteristics. CFD analysis was conducted to clarify the air temperature and airflow distribution inside an office room. The results of CFD analysis are used to be correct values in this paper.

The air temperature prediction of zonal model was compared with the results of CFD analysis to optimum solution.

Outline of CFD

Figure 1 shows the area of CFD analysis. The size of office room is 10.8m x 28.8m in plan and ceiling height is 2.6m. The hybrid air-conditioning system of the office room consists of wind-driven ventilation and mechanical air-conditioning system. The wind-driven ventilation opening is settled at near the ceiling. The win-driven opening was called NVO in this paper. The each NVO has the size of 3.6 (W) x 0.1 (H) m and continues. The mechanical air-conditioned air is supplied to the office room through under floor air distribution (UFAD) and flowed out from the room through return slits. The each UFAD has the size of 0.2 (W) x 0.2 (H) m.

Table 1 shows the setting of CFD analysis. Standard k-ε Model was used as the turbulence model. SIMPLEC was used as pressure-velocity coupling algorithm for the calculation with quick as discretization scheme of advection term. The size of meshes is 0.1 (X) × 0.1 (Y) × 0.1 (Z) m in the office room and the total meshes are 808,704. The boundary conditions of velocity are based on results of the calculated ventilation rate for each opening by network model using measured wind pressure coefficient by means of the wind tunnel test. The air change rate by the natural ventilation was set at 5 ACH. Table 2 shows the boundary conditions. 5 ACH was assumed as a round number from the measurement values in the hybrid ventilated building that the authors measured [4]. The outdoor air was supplied horizontally from NVO and air-conditioned air was supplied vertically from UFAD. Boundary conditions of internal heat loads were shown in Table 3. The office room has occupants, notebook PC, OA loads except notebook PC, and lighting equipment on ceiling as internal heat loads. Total of internal heat loads is 14,668W.

Results of CFD

Distribution of indoor air temperature

Figure 2 shows the CFD analyzed results of the distribution of the temperature at A-A' section (Figure 1). The dark color of figure means a high temperature and the light color of figure means a low temperature. The air temperature changes approximately 22~25 [deg. C] at most areas except the vicinity of the supply openings of air-conditioning and natural ventilation. The temperature rises while going to the leeward side. The temperature difference between the area near the supply natural ventilation openings and the exhaust natural ventilated openings is large and that is about 3~4 [deg. C]. It can be said that the difference of the thermal environment is large by the position of occupant. For provide to occupants the acceptable thermal environment, it is necessary that temperature distribution is predicted by the simplified method when designing a hybrid air-conditioning system.

Distribution of velocity

Figure 3 shows the CFD analyzed results of the distributions of velocity. The speed is equal to or less than 0.2m/s in most areas except the vicinity of the supply openings of air-conditioning and natural ventilation. The velocity of the fresh outdoor airflow reduces after flowing into the room, and the velocity is about 0.5m/s when it drops into the low level of the room from the ceiling.

Figure 4 shows the CFD analyzed results of the vertical distribution of velocity vector and temperature together. At the low level of the room, the outdoor air divided into two directions, flows in the leeward side of natural ventilation and the opposite direction of that. The high temperature of indoor air near the ceiling causes the outdoor airflow along the ceiling to drop in the lower level of the room. The direction of high temperature of indoor air is opposite direction of inflow outdoor air. Because the updraft at about 7m position away from the wall of the leeward side flows until vicinity that the inflow outdoor air is felled from ceiling.

Outline of Zonal model

The office room in the zonal model was assumed to be the area of two dimensions. Figure 5 shows the zone division of the calculated area. The office room was divided horizontally every 3.6m. In addition, the office room was divided vertically into the occupied zone and the non-occupied zone. The occupied zone is area from floor to the height of 1.8m. The office room is consists of 16 zones.

We assumed 4 airflow elements in the office room referring to the results of CFD. The 4 airflow elements in the office room is following as; (1) the airflow of natural ventilation, (2) the supplied airflow from UFAD, (3) the circulating flow caused by convection for wide area of office room, (4) the airflow mixed between occupied zone and non-occupied zone in the each zone. Figure 6 shows the outline of the zonal model.

Several assumptions were made in the zonal model. (1) The outdoor airflow from natural ventilation flow into the office room along the ceiling. The outdoor airflow flowed into the office room along the ceiling. The outdoor airflow entrained the ambient air falls down from the ceiling at $X_{\max}$. The outdoor airflow drops in the occupied zone of the same zone and the close zone. The position of $X_{\max}$ is determined by the following equation [5]:

$$X_{\max} = \left(0.925 Ar^{-\frac{2}{3}} \right) \cdot H_0 \cdot K \tag{1}$$

where:

H_0 = the height of supply natural ventilation opening

K = centerline velocity constant [6]

The amount of the outdoor air at the X [m] far from natural ventilation supply opening Q_x is described by the equation [7]:

$$Q_x = 0.248 \sqrt{\frac{X}{H_0}} \cdot Q_o \tag{2}$$

where:

Q_0 = flow rate of inflow outdoor air

The same flow rate as outdoor airflow rate flows in leeward side at zones except zones near supply and exhaust natural ventilation openings, the flow rate flows out through exhaust natural openings. At zones near supply and exhaust natural ventilation openings, the followings were assumed.

At zones near supply natural ventilation openings, Q_x is called as $Q_{jet(noi)}$ when *X* belongs to zone *NOi*. The amount of jet flow at zone *NOi* $Q_{jet(noi)}$ dropped in occupied zone from non-occupied zone *NOi*, the outdoor airflow $Q_{jet(noi)}$ is flowed in the occupied zone *Oi* of the same zone and the close zone. In this paper, $Q_{jet(noi-1)}$ is flowed in the zone *Oi-1* and *Oi*, and $Q_{jet(noi-1)}$ is described by the following equation:

$$Q_{jet(noi-1)} = Q_{jet(noi-1,oi-1)} + Q_{jet(noi-1,noi)} \tag{3}$$

$$Q_{jet(noi-1,oi-1)} = A \cdot Q_{jet(noi-1)} \tag{4}$$

$$Q_{jet(noi-1,noi)} = (1-A) \cdot Q_{jet(noi-1)} \tag{5}$$

where:

$Q_{jet(i,j)}$ = jet airflow rate that flowed in zone j from zone i

A = ratio of $Q_{(noi-1,oi-1)}$ to $Q_{jet(noi-1)}$

At zone *O7* near exhaust natural ventilation openings, the same flow rate $Q_{(o6,o7)}$ as outdoor airflow rate is divided airflow rate $Q_{(o7,no7)}$ and $Q_{(o7,o8)}$. The relation is described by the following equation:

$$Q_{(o6,o7)} = Q_{(o7,no7)} + Q_{(o7,o8)} \tag{6}$$

$$Q_{(o7,no7)} = B \cdot Q_{(o6,o7)} \tag{7}$$

$$Q_{(o7,o8)} = (1 - B) \cdot Q_{(o6,o7)} \tag{8}$$

where:

$Q_{(o7,no7)}$ = airflow rate that flowed in non-occupied zone *NO7* from occupied zone *O7*

$Q_{(o7,o8)}$ = airflow rate that flowed in occupied zone *O8* from zone *O7*

B = ratio of $Q_{(o7,\,no7)}$ to $Q_{(o6,\,o7)}$

(2) The supplied airflow from UFAD does not flow to the near occupied zone, and flow only to non-occupied zone in the same zone. The same amount of supplied air from UFAD in zone Oi flows to zone NOi, and flowed out the office room through return air opening. This relationship is described as following equation:

$$Q_{SA(oi)} = Q_{RA(noi)} \tag{9}$$

where :

$Q_{SA(oi)}$: flow rate of supply air of air-conditioning in zone Oi

$Q_{RA(noi)}$: flow rate of return air in zone NOi

(3) There is the airflow of the circulating flow on the wide area. This circulating airflow by convection exists in the zone except the close zone of supply opening and exhaust opening of natural ventilation. This circulating airflow rate Q_{α} is assumed to be in zone 2 ~ zone6 referring to the results of CFD.

(4) There is the mixed airflow rate between occupied zone and non- occupied zone. This airflow is in the whole zone except the close zone of the supply opening and exhaust opening of natural ventilation. The airflow rate that flowed in occupied zone *NOi* from occupied zone *Oi* is designated as Q_{ui}. The airflow rate that flowed in occupied zone *Oi* from occupied zone *NOi* is designated as Q_{di}. This relationship is described as following equation:

$$Q_{di} = Q_{ui} \tag{10}$$

The temperatures for all zones are calculated by solving air balance equation and heat balance equation for each zone. Table 4 shows the parameters that were used to calculate the optimum solution. At first, the temperature differences for all zones between CFD results and zonal model results are calculated by means of the each parameter's combination. The mean absolute value of temperature difference between CFD results and zonal model results ΔT defined by following equation:

$$\Delta T = \frac{\left| T_{CFD(i)} - T_{ZONE(i)} \right|}{N} \tag{11}$$

Where:

$T_{CFD(i)}$: average temperature of zone i calculated by CFD analysis

$T_{ZONE(i)}$: average temperature of zone i calculated by zonal model

N : number of zone

The optimum solutions of Q_α and Q_d were founded as the minimum value of the calculated ΔT on each case. The constant A and B were calculated in the same method. The values were calculated in order of constant A, B.

Results and Discussion

Figure 7 shows the mean absolute value of temperature differences $\Delta T_{(\alpha, d, 0, 0)}$ between CFD results and zonal model results for all zones by combining Q_α and Q_d. Here, constant A and B were 0. $\Delta T_{(\alpha, d, A, B)}$ means that the temperature difference was calculated by using α, d, A and B as Q_α, Q_d, A and B.

The value of ΔT changes 0.7 ~ 1. 2 [deg. C]. The smallest value of $\Delta T_{(\alpha, d, 0, 0)}$ is 0.7 [deg. C] when Q_d is 300 [m^3/h] and Q_α is 1700 [m^3/h]. The values of constant A and B were calculated by these values.

Figure 8 shows the mean absolute value of temperature differences $\Delta T_{(1700, 300, A, 0)}$ between CFD results and zonal model results for all zones when constant A is changed from 0 to 1 at 0.1 intervals. The value of constant B was 0. The smallest value of $\Delta T_{(1700, 300, A, 0)}$ calculated for all zones is 0.47 [deg. C] when A is 0.8. Figure 9 shows the calculated $\Delta T_{(1700, 300, 0.8, B)}$ when constant B is changed from 0 to 1 at 0.1 intervals. The smallest value of $\Delta T_{(1700, 300, 0.8, 0)}$ calculated for all zones is 0.37 [deg. C] when B is 0.3.

Figure 10 shows the temperatures of CFD results and zonal model results when using the calculated values. 1700 [m^3/h], 300 [m^3/h], 0.8 and 0.3 were used by the values of Q_α, Q_d, constant A and constant B. The temperature of the office room has the distribution of 20~26 [deg. C]. The temperature results of zonal model in each zone are similar with CFD results at the most zones. However, in zone 1 of the windward and zone 7, 8 of the leeward side, the temperature difference between zonal model results and CFD results is large. It is necessary to clarify a current of air characteristic around the natural ventilation openings.

Temperature of return air, removed heat load rate by air-conditioning

Figure 11 shows the return air temperatures. The calculated return air temperature by zonal model is lower than that by CFD. Figure 12 shows the comparison of the removal rate of heat load by air-conditioning and natural ventilation. These values were calculated by using air temperature of inflow and outflow side opening of natural ventilation and air-conditioning. The removal rate of the heat load by air-conditioning in CFD results is about 8.5% larger than that by zonal model. Because the temperature of each zone was calculated by assuming complete mixture and it was not able to calculate temperature distribution of the vicinity of ceiling.

Conclusions

We tried to develop the simplified prediction model that can predict temperature distribution inside the office room with hybrid air-conditioning system. CFD analysis was conducted to clarify the air temperature and airflow distribution inside an office room. The results of CFD analysis are used as correct values in this paper.

The temperature distribution predicted from zone model agreed approximately with CFD results at the most occupied zones except the windward and leeward window side. However, the differences of the return air temperature near the ceiling between CFD results and zone model results are not small. It can be said that the zone model is effective to predict the temperatures of occupied zones. However, the zone model is not effective to predict the removal rate of heat loads by the air-conditioning and the natural ventilation. For applying the zone model to various conditions, more investigations will be needed. Especially, it is necessary to organize data concerning the airflow rates in each zone, constant A, and B.

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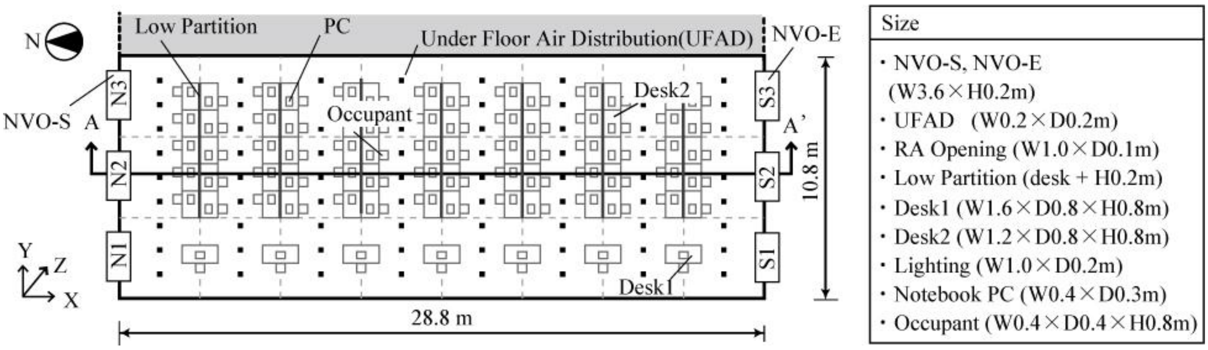
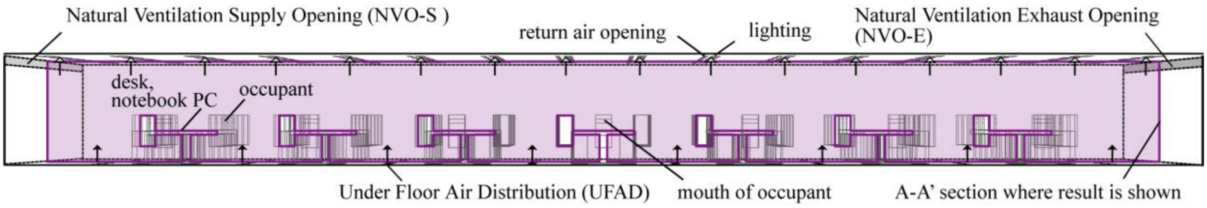


(a) Layout of office room



(b) Section where result is shown

Figure 1 Area of CFD analysis

Table 1 Setting of CFD analysis

Turbulence model	Standard k-ε
Algorithm	Steady state (SIMPLEC)
Analysed area	28.8m(X) x 10.8m(Z) x 2.6m(Y)
Meshes	108(X) x 208(Z) x 26(Y)
Discretization scheme for advection term	QUICK
Wall boundary condition	Standard log-low, Adiabatic

Table 2 Boundary conditions

Velocity [m/s]	Air-conditoning (flow rate : 3,261 m^3/h)		UFAD	0.31
			Exhaust Opening	-0.944
	Natural Ventilation (flow rate : 4,032 m^3/h)	NVO-S	N1	0.50
			N2	0.52
			N3	0.54
		NVO-E	S1	-0.51
			S2	-0.52
			S3	-0.53
Temperature [deg.C]	Air-conditoning			20 deg.C
	Natural Ventilation			18 deg.C

Table 3 Internal heat loads

Heat load		Number
Person	60 W	77
Notebook PC	30 W	77
Lighting on ceiling	32W (60% on setting position, 40% on floor and desks)	96
OA loads except PC	30 W/m^2	

Table 4 Parameters that were used to calculate the optimum solution

Q_{α}[m^3/h]	0,500, from 1000 to 3000 at 100 intervals, 3500, 4000
Q_d [m^3/h]	from 0 to 2000 at 100 intervals
A [-]	from 0 to 1 at 0.1 intervals
B [-]	from 0 to 1 at 0.1 intervals

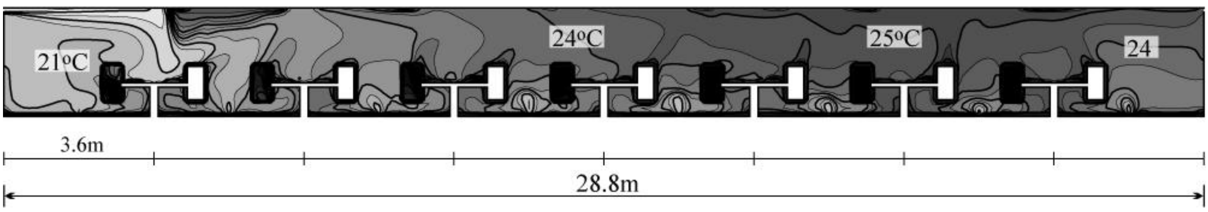


Figure 2 Distribution of Temperature at A-A' Section [deg. C]

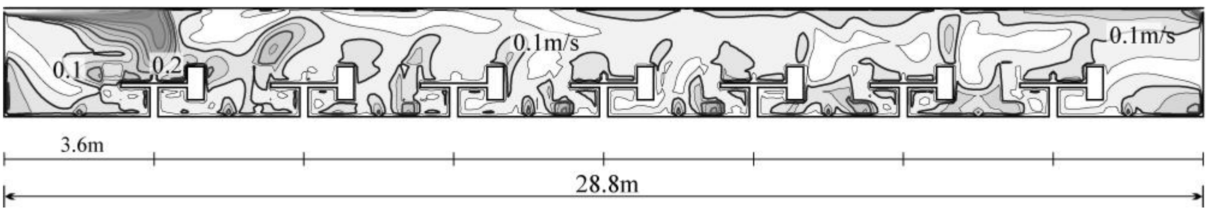


Figure 3 Distributions of velocity at A-A' Section [m/s]

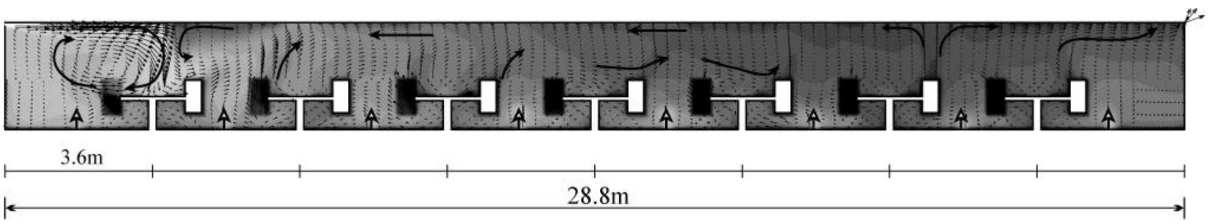


Figure 4 Distributions of velocity vector and temperature at A-A' Section

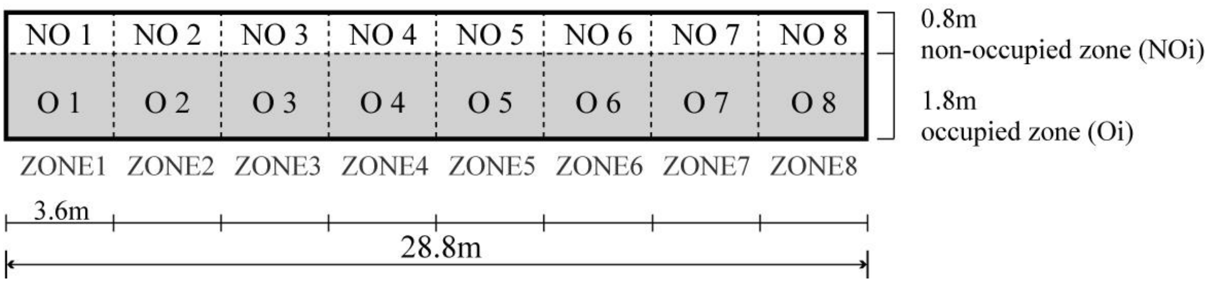
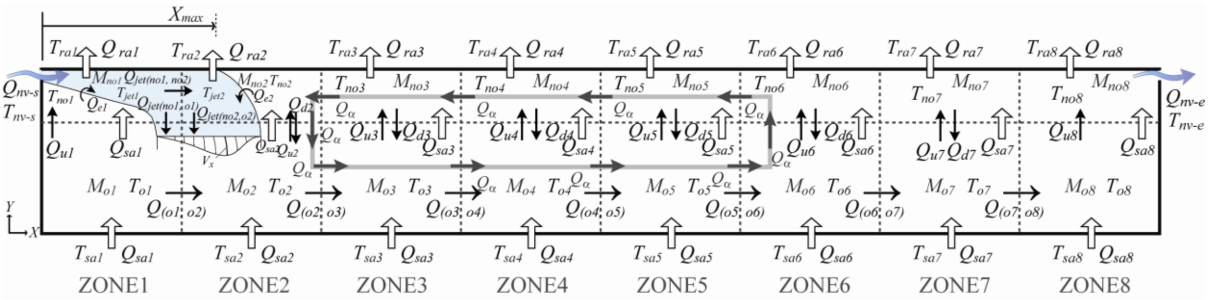


Figure 5 Zone division of the calculated area



Flow Rate

Q_{sai}	: flow rate of supply air of air-conditioning in zone i
Q_{rai}	: flow rate of return air in zone i
$Q_{nv\text{-}s}$	: flow rate of supply air by natural ventilation
$Q_{nv\text{-}e}$	: flow rate of exhaust air by natural ventilation
Q_{ei}	: entrained flow rate of outdoor air in zone NOi
Q_{ui}	: flow rate that flows from zone Oi to zone NOi
Q_{di}	: flow rate that flows from zone NOi to zone Oi
$Q_{(noi,\ noi+1)}$	: flow rate that flows from zone NOi to zone NOi+1
$Q_{(oi,\ oi+1)}$	: flow rate that flows from zone Oi to zone Oi+1
Q_a	: large circulation airflow by convection in zone 2~6
Q_{jeti}	: flow rate of jet flow of inflow outdoor air+ Q_{ei}

Temperature and Heat Load

T_{sai}	: temperature of supply air of air-conditioning in zone i
T_{rai}	: temperature of return air
$T_{nv\text{-}s}$	: temperature of supply air by natural ventilation
$T_{nv\text{-}e}$	: temperature of exhaust air by natural ventilation
T_{noi}	: temperature of zone NOi
T_{oi}	: temperature of zone Oi
T_{jeti}	: temperature of jet flow in zone NOi
M_{noi}	: heat load in zone NOi (lighting)
M_{oi}	: heat load in zone Oi (occupant, note PC, OA equipment)

Constant

A	: $Q_{jet(no1,\ o1)} / Q_{jet1}$
B	: $Q_{u7} / Q_{(o6,\ o7)}$

Figure 6 Outline of the zonal model

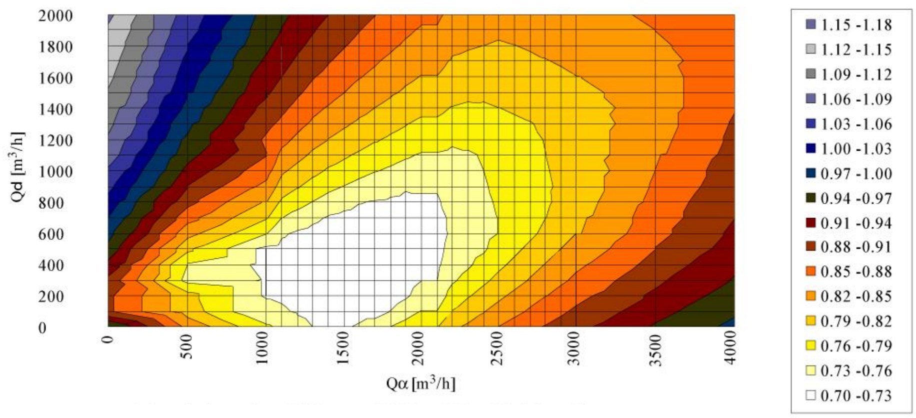


Figure 7 Mean absolute value of temperature differences $\Delta T\,(Q_\alpha, Q_d, A, B)$ between CFD results and zonal model results for all zones by combining Q_α and Q_d (constant A=0, B=0)

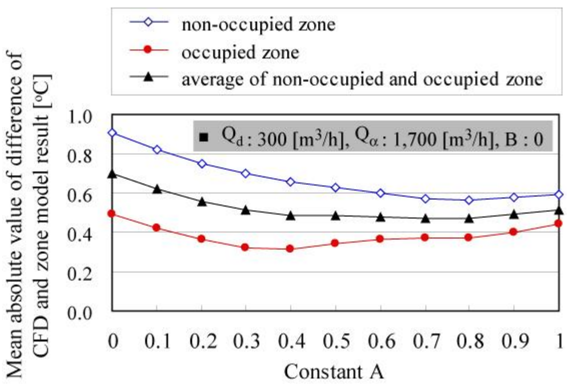


Figure 8 Mean absolute value of temperature differences $\Delta T_{(1700,\,300,\,A,\,0)}$ between CFD results and zonal model results for all zones when constant A is changed from 0 to 1 at 0.1 intervals (B=0)

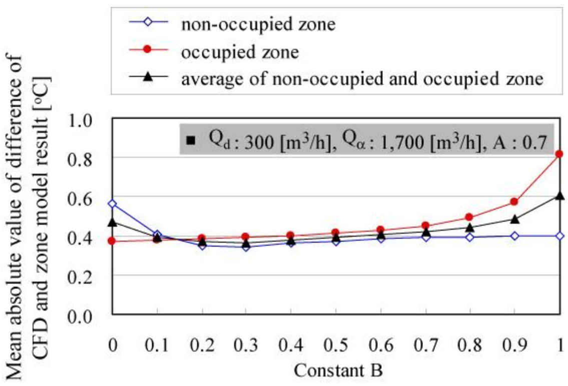


Figure 9 Mean absolute value of temperature differences $\Delta T_{(1700,\,300,\,0.8,\,B)}$ between CFD results and zonal model results for all zones when constant B is changed from 0 to 1 at 0.1 intervals

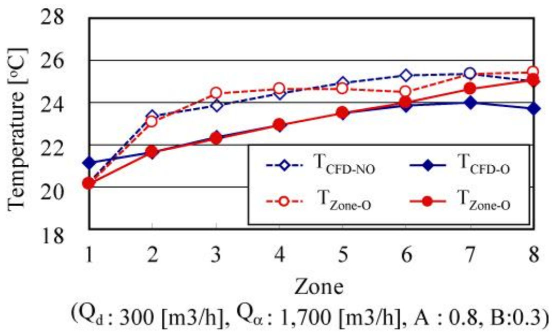


Figure 10 Comparison between temperature of zonal model results and that of CFD results on each zone (Q_α=1700, Q_d=300, A=0.8, B=0.3)

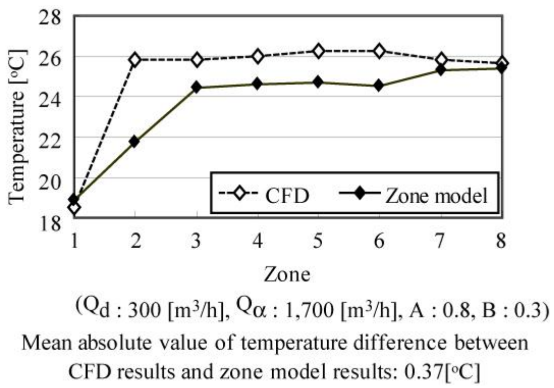


Figure 11 Comparison between return air temperature of zonal model results and that of CFD results

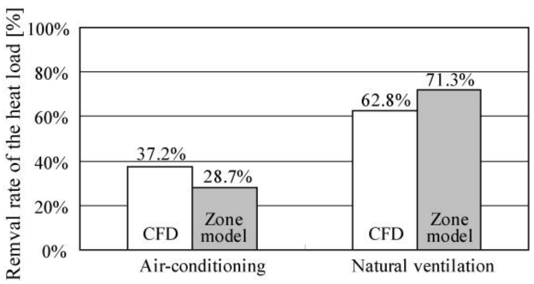


Figure 12 Comparison between removal rate of heat load by air-conditioning and that by natural ventilation