NUMERICAL ANALYSIS OF NORMALIZED CONCENTRATIONS IN THE OCCUPIED ZONE FOR VARIOUS OFFICE VENTILATION SYSTEMS

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SUMMARY

Ventilation Standard HASS 102 of The Society of Heating, Air-Conditioning, and Sanitary Engineers of Japan (SHASE) was revised in 1997. Ventilation Standard HASS 102 recommends the ventilation design, which takes into account a ventilation effectiveness in an occupied zone, under incompletely mixed condition. The index for evaluating ventilation effectiveness in the standard is called the Normalized Concentration (*Cn*) in an Occupied Zone. The definition of *Cn* is the ratio of the average contaminant concentration in the occupied zone (*Cp*) to that of the completely mixed indoor air (*Ce*). To produce an effective design for a ventilation system, more data about the *Cn* is required. In this paper, the values of *Cn* in an occupied zone in an office space are calculated by CFD simulation techniques for several different ventilation systems. Five different ventilation systems were chosen and were investigated under two thermal conditions. The thermal boundary conditions are cooling in the summer season and heating in the winter season. Furthermore, five conditions were chosen for the supply air according to the changes in the Archimedes number (*Ar*) and Reynolds number (*Re*) under a constant heat load. In addition, changes in the value of *Cn* are evaluated by the parameters *Ar* and *Re* for the supply air. For most ventilation systems, the *Cn* is around 1.0, but it changes greatly depending on *Re* and *Ar* for displacement ventilation system.

KEYWORDS

Normalized Concentration in the Occupied Zone, CFD, Archimedes number

INTRODUCTION

The contamination of room air is a problem that is not only related to building materials, furniture and fixtures, the use of chemical products, insecticides and others, but is also significantly related to the air tightness and ventilation systems used in modern buildings [1]. The contamination of room air is mainly due to economic and social pressures leading to the increased use of chemical substances

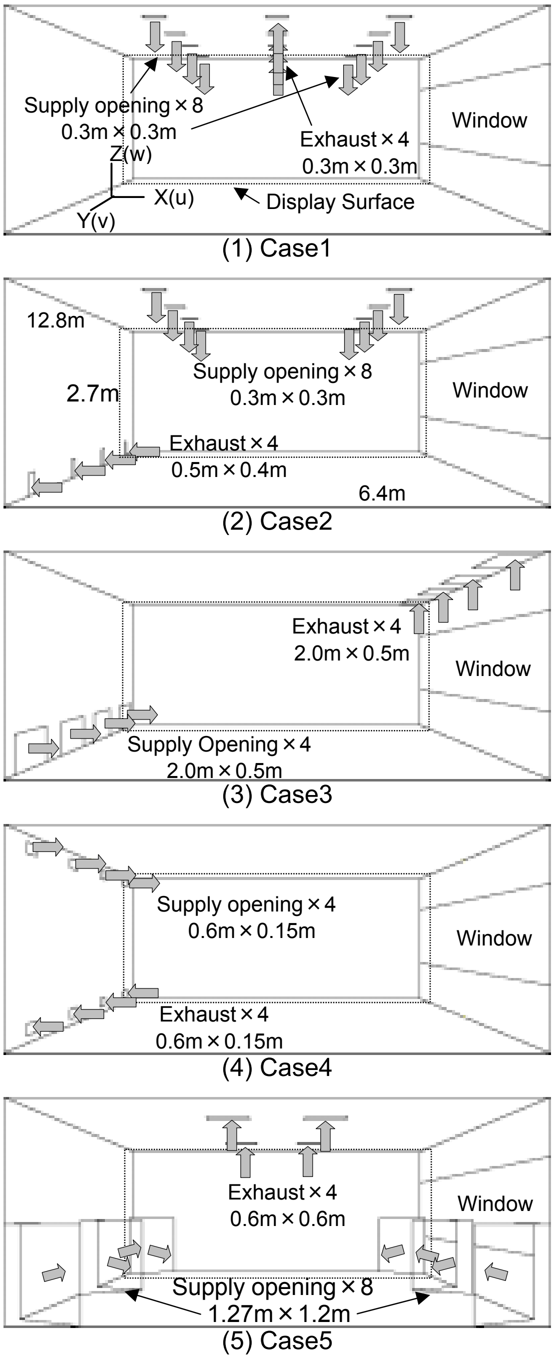


Figure 1 Office Model Analyzed and Ventilation System

and reductions in the air change rate to decrease in the energy consumption of air conditioning. The tendency to minimize the ventilation rate is likely to increase in the future with efforts to prevent global warming and to control the discharge of carbon dioxide gas [2]. In order to safeguard the health of the occupants, it will become necessary to provide suitable ventilation systems that eliminate contaminants efficiently. The ventilation efficiency in the occupied zone is analyzed by CFD (Computational Fluid Dynamics) for various ventilation systems and heat loads, with the aim of offering a database for effective ventilation designs.

VENTILATION EFFICIENCY

In this analysis, the Normalized Concentration (*Cn*) and the Age of Air in the occupied zone are adopted as the index for evaluating the ventilation efficiency in the occupied zone. The definition of *Cn* is the ratio of the average pollutant concentration in the occupied zone (*Cp*) to that of the completely mixed indoor air (*Ce*). This becomes to equal to the ratio of the average Age of Air in the occupied zone (*Ap*) to a nominal time constant (τ_n).

$$Cn = Cp/Ce = Ap/\tau_n \qquad (1)$$

The *Cn* is the most basic index of ventilation efficiency, and if the pollutant generation rate and the *Cn* are known, the ventilation rate of fresh air to the room, which is to maintain the average concentration in the occupied zone (*Cp*) at suitable level, can be directly calculated. The occupied zone indicates the space in the room where the occupants reside and act (a space 1.8m in height from the floor) in this analysis

OFFICE MODEL

The office models shown in Figure 1 are used for analyzing the *Cn*. The office model has a volume of 6.4m (x) × 12.8m (y) × 2.7m (z). To examine the property of the *Cn* index, five different ventilation systems were chosen and were investigated under two thermal conditions; cooling in the summer season and heating in the winter season. Furthermore, five supply air conditions were chosen according to the change in the Archimedes number (*Ar*) and Reynolds number (*Re*) under a constant heat load. In addition, changes in the value of *Cn* are evaluated by the parameters *Ar* and *Re* for the supply air [Note]. The thermal (Heat Load) conditions and the conditions of the air conditioning system are shown in Tables 1 and 2.

NUMERICAL METHODS AND BOUNDARY CONDITIONS

CFD was carried out using a low Reynolds type k-ε model (Abe-Nagano model). A second-order center difference scheme was used for the convection term.

Table 1 Heat Load Conditions

Heat Source	Lighting	Human Body	Equipment	Solar Heat	Heat transmission (Window)	Heat storage (Wall)	Total
Cooling Load	1638 (20)	1104 (13)	3200 (39)	2765 (34)	625 (8)	—	9332 (114)
Heating Load	—	—	—	—	-2028 (-25)	-6810 (-83)	-8838 (-108)

Units: [W],() Per-floor area [W/m²],
The direction from outdoor to indoor is +

Table 2 Conditions of the Air Conditioning System

Case	Supplied air Condition (Difference of temp.)	Target Temp. [℃]	Inlet Temp [℃]	Air exchange rate [h⁻¹]	U_{in} [m/s]	*Ar*	*Re* (×10⁶)
1-1	Isothermal	-	-	10.0	0.85	0	0.16
1-2	Cooling (-16℃)	26	10	7.9	0.67	-3.13	0.13
1-3	Cooling (-10℃)	26	16	12.6	1.07	-0.76	0.21
1-4	Cooling (- 3℃)	26	23	42.0	3.58	-0.02	0.69
1-5	Heating (+10℃)	22	32	11.9	1.02	0.85	0.20
1-6	Heating (+ 3℃)	22	25	39.8	3.39	0.02	0.65
2-1	Isothermal	-	-	10.0	0.85	0	0.16
2-2	Cooling (-16℃)	26	10	7.9	0.67	-3.13	0.13
2-3	Cooling (-10℃)	26	16	12.6	1.07	-0.76	0.21
2-4	Cooling (- 3℃)	26	23	42.0	3.58	-0.02	0.69
2-5	Heating (+10℃)	22	32	11.9	1.02	0.85	0.20
2-6	Heating (+ 3℃)	22	25	39.8	3.39	0.02	0.65
3-1	Isothermal	-	-	10.0	0.15	0	0.03
3-2	Cooling (-16℃)	26	10	7.9	0.12	-96.6	0.02
3-3	Cooling (-10℃)	26	16	12.6	0.19	-23.6	0.04
3-4	Cooling (- 3℃)	26	23	42.0	0.65	-0.64	0.12
3-5	Heating (+10℃)	22	32	11.9	0.18	26.29	0.04
3-6	Heating (+ 3℃)	22	25	39.8	0.61	0.71	0.12
4-1	Isothermal	-	-	10.0	1.71	0	0.33
4-2	Cooling (-16℃)	26	10	7.9	1.34	-0.78	0.24
4-3	Cooling (-10℃)	26	16	12.6	2.15	-0.19	0.41
4-4	Cooling (- 3℃)	26	23	42.0	7.17	-0.01	1.38
4-5	Heating (+10℃)	22	32	11.9	2.04	0.02	0.39
4-6	Heating (+ 3℃)	22	25	39.8	6.79	0.09	1.31
5-1	Isothermal	-	-	10.0	0.05	0	0.01
5-2	Cooling (-16℃)	26	10	7.9	0.04	-900	0.01
5-3	Cooling (-10℃)	26	16	12.6	0.06	-220	0.01
5-4	Cooling (- 3℃)	26	23	42.0	0.21	-5.94	0.04
5-5	Heating (+10℃)	22	32	11.9	0.06	245	0.01
5-6	Heating (+ 3℃)	22	25	39.8	0.20	6.62	0.04

Table 3 Numerical Conditions

Mesh	CASE1 ; 78(x)×72(y)×35(z)＝196,560 CASE2, 3; 67(x)×63(y)×27(z)＝113,967 CASE4 ; 72(x)×68(y)×51(z)＝249,696 CASE5 ; 64(x)×64(y)×27(z)＝110,592
Turbulence model	Low Re Type k-ε model
Inflow Boundary Condition	U_{in}; show table 3, $k_{in}=3/2(U_{in}\times0.05)^2$, $\varepsilon_{in}=C_\mu \cdot k_{in}^{3/2}/L_{in}$, $C_\mu=0.09$, L_{in}=1/7 scale of supply opening
Outflow Boundary Condition	U_{out}, k_{out}, ε_{out}＝free slip
Wall Treatment	No slip
Contaminant	Passive, uniformly and continuously generated throughout the space

The contaminant concentration distributions were analyzed by using the prior prediction results of the flow fields. It was assumed that the contaminants were generated uniformly and continuously throughout the space, and moved passively. Under this assumption, the air mass from the supply inlet is gradually polluted, and the pollution level is taken to be proportional to the elapsed time from when the air mass enters the room space until it reaches the point or region. Therefore, for a uniform and continuous contaminant generation throughout the space, the concentration at a point or region corresponds to the Age of Air (*Ap*), and the *Cn* value corresponds to the *Ap*, which normalized by a nominal time constant [3, 4]. Numerical conditions are shown in Table 3.

RESULTS AND DISCUSSION

Flow and Temperature Fields

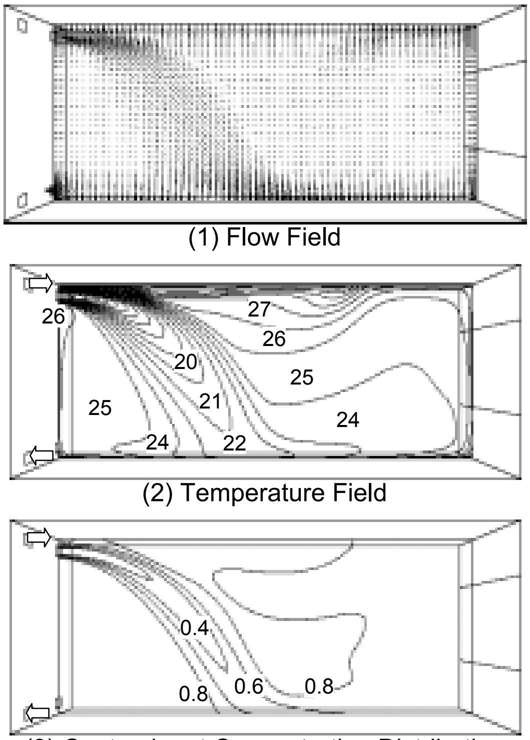
The prediction results for the flow and temperature field by CFD are shown in Figure 2. This shows the representative result (Case 4-2) among all analyzed cases, but the remaining results are omitted.

In Case 1, the supply air reaches the floor under cooling conditions. Under heating conditions, the supply air flows upward due to buoyancy and does not reach the occupied zone; thus, the temperature in the occupied zone does not rise. In Case 2, the flow field is similar to Case 1. The average temperature in the occupied zone in Case 2 is higher than that in Case 1. In Case 3, an anti-clockwise re-circulating flow is formed in the room under cooling conditions. Under heating conditions, the supplied air immediately flows upward due to buoyancy; thus thermal stratification is formed in the room. In Case 4, the supplied air immediately descends to the floor level under cooling conditions. Under heating conditions, a clockwise re-circulating flow is formed in the room and the room average temperature is uniformly distributed. In Case 5, thermal stratification is formed in the room under cooling conditions. A complex flow is formed under heating conditions.

Ventilation Efficiency (*Cn*)

The prediction result for the contaminant concentration distribution in Case 4-2 is shown in Figure 2 (3). The concentration values are normalized by the completely mixed concentration.

Table 4 shows the prediction results of the *Cn* in each case. In Cases 1 and 2, the *Cn* becomes about 1.0, except in Case 1-5. In Case 3, the *Cn* becomes larger compared with the other cases. When a supply opening is set up in the vicinity of the floor, the height and width of the supply opening greatly influences the average concentration in the occupied zone. In Case 4, the *Cn* becomes 1.0 or less in all the cases analyzed. In Case 5, the *Cn* becomes 0.58 - 0.85 under cooling conditions. The *Cn* in case 5 is the smallest under cooling conditions. Figure 3 shows the relationship between the *Re* number and the *Cn* index, and Figure 4 shows the relationship between the *Ar* number and the *Cn* index. In Cases 2 and 4, the *Cn* value becomes about 1.0 regardless of the *Re* and *Ar* number. In Case 5, when the *Ar* number is negative, the *Cn* becomes 0.9 or less. As a whole, when the exhaust opening is set up in the upper part of the room and the *Ar* number is large, the *Cn* greatly exceeds 1.0. When the *Re* number is small, the *Cn* varies greatly, and when the *Re* number is large, the *Cn* becomes near 1.0 because the room air is almost completely mixed by the supply air jet.



(1) Flow Field

(2) Temperature Field

(3) Contaminant Concentration Distribution

Figure 2 Prediction Results for the Flow, Temperature, and Contaminant Distribution (Case 4-2)

Table 4 Analysis Results of *Cn*

Supply Air Condition	Case1	Case2	Case3	Case4	Case5
-1 Isothermal	0.92	1.02	1.25	0.84	0.74
-2 Cooling (-16℃)	0.95	0.97	1.09	0.86	0.58
-3 Cooling (-10℃)	0.98	1.03	0.90	0.89	0.73
-4 Cooling (- 3℃)	0.96	1.00	1.33	0.87	0.85
-5 Heating (+10℃)	1.53	0.99	1.24	0.85	1.52
-6 Heating (+ 3℃)	0.96	1.00	1.55	0.99	1.49

CONCLUDING REMARKS

(1) Flow, temperature field and contaminant concentration distributions are analyzed using CFD for five different ventilation systems.

(2) The characteristics of the contaminant concentration distributions due to the differences in the ventilation systems are shown.

(3) The ventilation systems are evaluated using the *Cn* index. No clear relationships could be found between the *Re* number, the *Ar* number, and the *Cn* index.

(4) Under these analytical conditions, when the *Ar* number is large and the *Re* number is

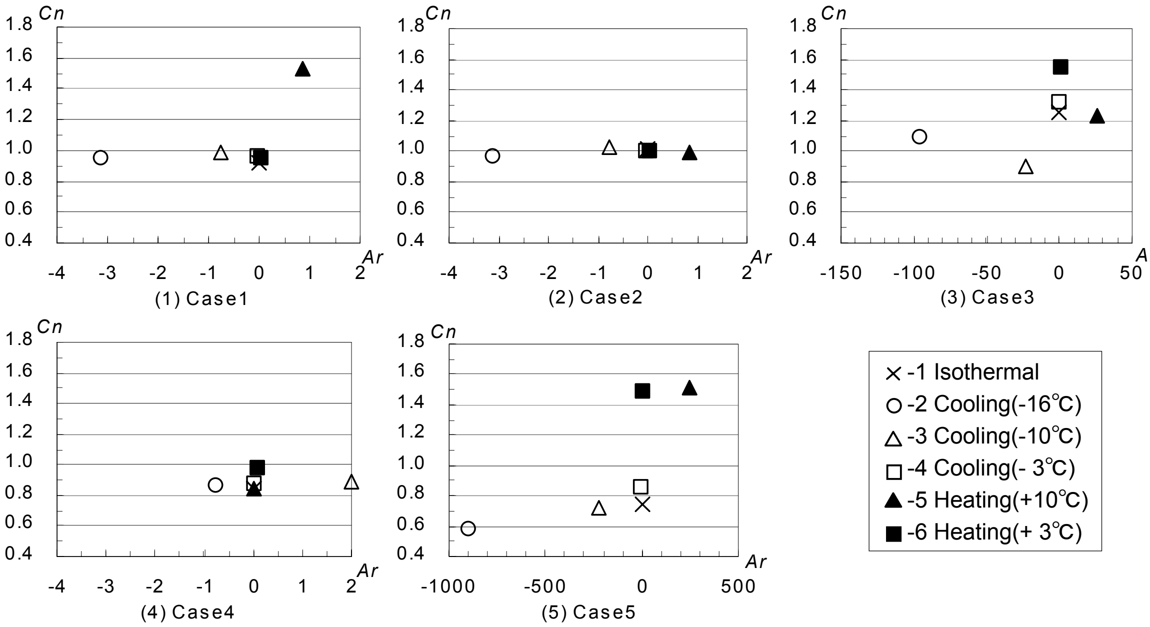


Figure 3 The relationship between the *Ar* number and *Cn*

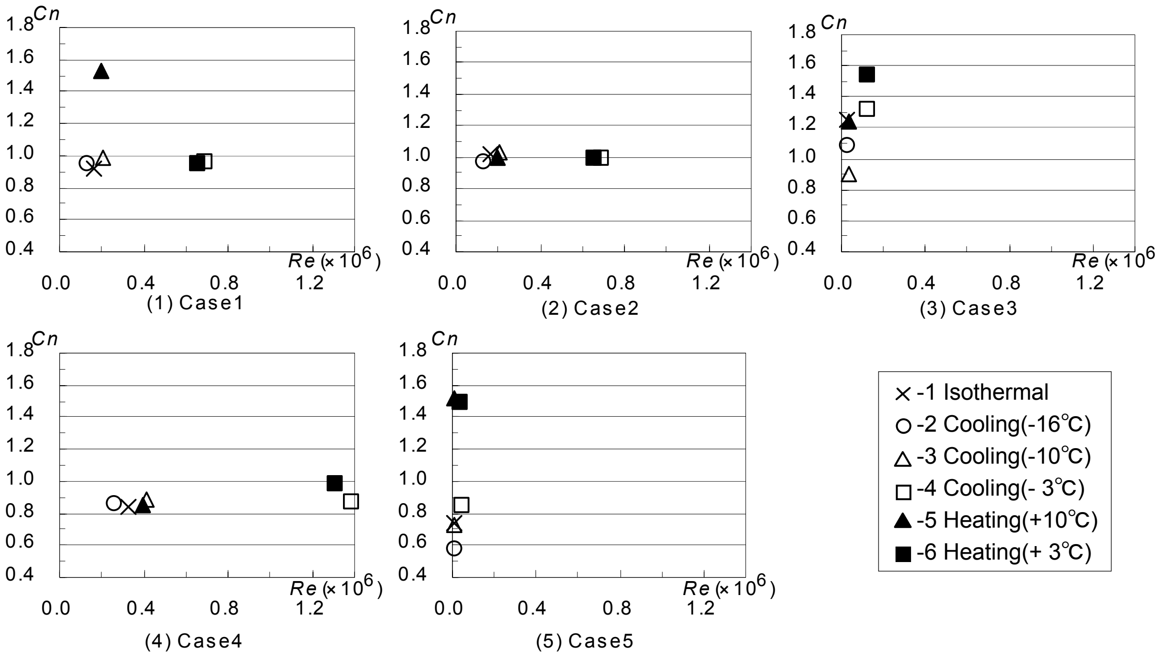


Figure 4 The Relationship between the *Re* number and *Cn*

small, the *Cn* tends to become a value greater than 1.0.

NOTE

In this paper, the *Re* number and *Ar* number are defined as follows; $Re = U_{in} L_0/\nu$; $Ar = g \beta \Delta\theta L_0/U_{in}^2$ Here, L_0 is the representative length scale (= 2.7m = room height between floor and ceiling).

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