Study on Normalized Concentrations in an Occupied Zone in Office Space
Optimization of Fresh Supply Air Flow Rate and Analysis of Energy Consumption

S. Kikuchi[1], K. Ito[2], N. Kobayashi[2]

[1]Kawamoto-ind Co., Ltd., Kanagawa, Japan
Email: s-kikuchi@kawamoto-ind.co.jp http://www.kawamoto-ind.co.jp
[2]Tokyo Polytechnic University, Kanagawa, Japan

Summary: *The values of the normalized concentration in the occupied zone (Cn) in an office space are calculated by CFD for five different ventilation systems and the minimum ventilation rate which maintains the average concentration in the occupied zone under the regulated value is analyzed. Energy consumptions associated with the change in ventilation rate are analyzed. In this analysis, for most ventilation systems, the value of Cn is around 1.0, but for large circulation flow ventilation systems it changes greatly depending on the supply inlet velocity and temperature. In the case in which the Cn index was applied, the maximum energy consumption decreased by about 4 %, compared with the case in which perfect mixing conditions were assumed.*

Keywords: *Office Space, CFD, Normalized Concentration in the Occupied Zone, Energy Consumption*
Category: *Case Studies*

1 Introduction

In general, ventilation designs are considered under conditions where contaminants are completely mixed in the whole room. However contaminants in the room are not always uniformly distributed and it is necessary to introduce the concept of ventilation efficiency into the ventilation design. From the viewpoint of energy conservation, it is important to minimize the ventilation rate with the indoor contaminant concentration level controlled under the regulation value. In this paper, the values of the normalized concentration in the occupied zone (*Cn*) in an office space are analyzed by CFD simulation for five different ventilation systems [1], and the minimum ventilation rate which maintains the average concentration in the occupied zone under the regulation value is analyzed. Furthermore, the annual energy consumption associated with the change in ventilation rate is analyzed.

2 Analysis of Ventilation Efficiency and Annual Energy Consumption

In this analysis, the Normalized Concentration (*Cn*) is adopted as the index for evaluating the ventilation efficiency in the occupied zone. A schematic of commonly-used HVAC systems is shown in Figure 1. The definition of *Cn* is as follows:

$$C_n = \frac{C_a - C_0}{C_p - C_0} \quad (1) \qquad C_p = \frac{M}{Q_p} + C_0 \quad (2)$$

Here, *Ca* is the average pollutant concentration in the occupied zone, C_p is the completely-mixed concentration, C_0 is the outdoor concentration, and *M* is the contaminant generation rate. The occupied zone indicates the space in the room where the occupants reside and act (a space 1.8 m in height from the floor) in this analysis. Q_p is the ventilation rate which defines the perfect mixing concentration (C_p). Using the *Cn* data, the final ventilation rate (*Q*) at the design stage that controls the C_a level to be under C_p is estimated as follows:

$$Q = Q_p \times C_n \quad (3)$$

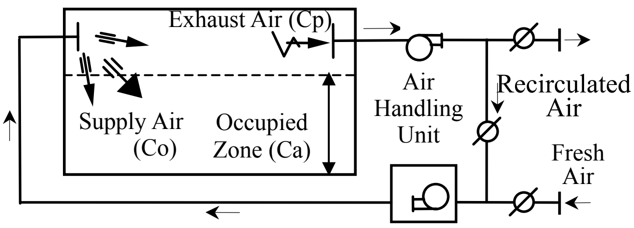


Figure 1 Schematic of HVAC system

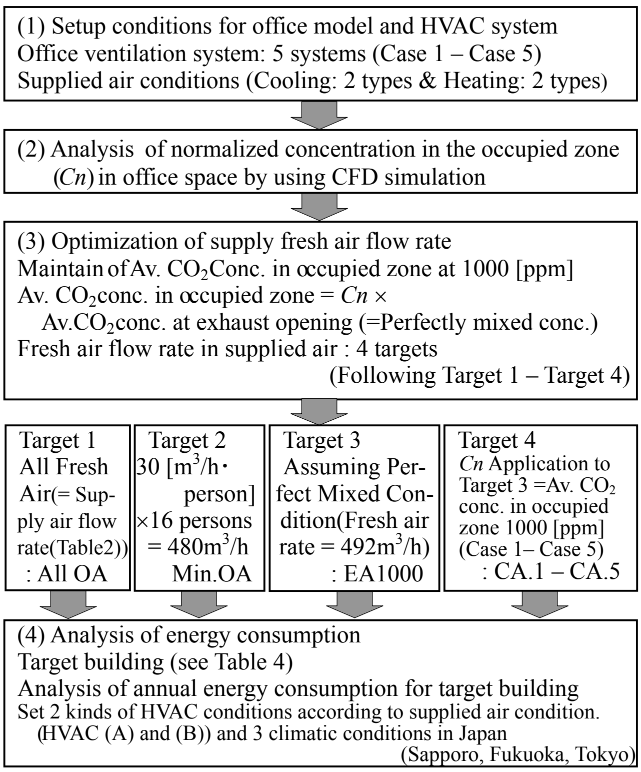


Figure 2 Flowchart of analysis of energy consumption

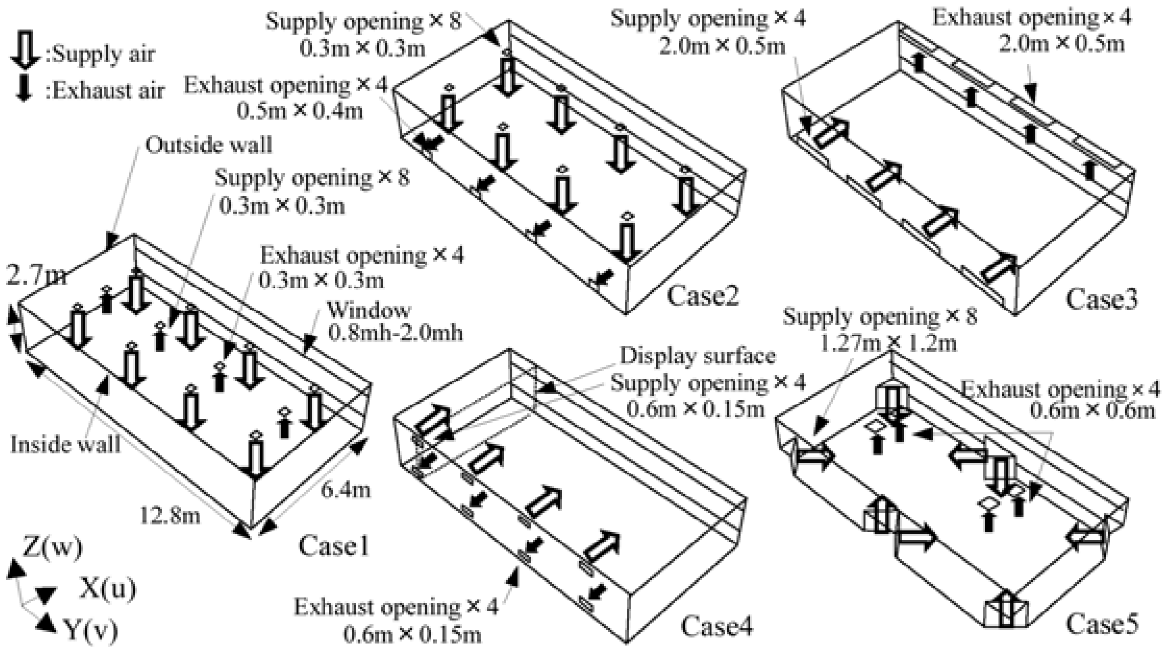


Figure 3 Office model and ventilation system

Energy consumptions associated with the change in ventilation rate from Q_p to Q are analyzed. A flowchart of the analysis of the energy consumption is shown in Figure 2. At the first step, we analyzed *Cn* for each ventilation system under various thermal conditions. At the next step, we estimate the optimized fresh airflow rate by investigating the average CO_2 concentration in the occupied zone with the *Cn* data. In Japan, indoor CO_2 concentration levels are regulated to be less than 1000 [ppm] by the Act for the Maintenance of Sanitation in Buildings. In this study, we observed the CO_2 concentration in the occupied zone and compared the value of the CO_2 concentration in the occupied zone with the value of the CO_2 concentration in the exhaust air. If the ventilation efficiency is high (effective), the CO_2 concentration in the occupied zone will be less than the exhaust CO_2 concentration. In that case, we will be able to reduce the fresh air flow rate rather than the volume of fresh air under perfect mixing conditions at the design stage. In fact, we were able to reduce the fresh air flow rate, and thus save on energy consumption for the air-conditioning system. In this study, the annual energy consumption associated with the change in the fresh air flow rate was analyzed under a variety of climatic conditions occurring in Japan. Annual energy consumptions were calculated for different levels of ventilation rate using the E-passplan software [2]. As shown in Figure 2, Target 1 is the case which targeted on all fresh air in the supply air. Target 2 is the case which targeted on minimum fresh air rate for the worker (30 m^3/h/person). Target 3 is perfect mixing condition, and which targeted 1000 [ppm] at exhaust CO_2 concentration. Target 4 is the case which maintained 1000 [ppm] in an occupied zone by using ventilation efficiency.

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/m^2] The direction from outdoor to indoor is +

Table 2 Conditions for the air-conditioning system

Supplied Air Condition (diff. of temp. $\Delta\theta$)	Target Temp.[°C]	Heat Load [W/m^2]	Inlet Temp [℃]	Supply Air Flow Rate [m^3/h]
(1) Cooling (-10˚C)	26	+114	16	2786
(2) Cooling (- 3˚C)	26	+114	23	9287
(3) Heating (+10˚C)	22	-108	32	2639
(4) Heating (+ 3˚C)	22	-108	25	8769

3 Office Model and HVAC System

The office model is shown in Figure 3. The office model has a volume of 6.4m (x) × 12.8m (y) × 2.7m (z). Five different ventilation systems (Cases 1 to 5) were chosen and were investigated under two thermal conditions: cooling in the summer season and heating in the winter season. Case1 and 2 are selected as typical air-conditioning system of office space in Japan. Case3 and 4 are selected as an air-conditioning system which generate circulated flow fields in the space. Case5 is intended as displacement ventilation system. Furthermore, four supplied air conditions were chosen according to the change in the Archimedes number (*Ar*) and the Reynolds number (*Re*) under a constant heat load. The conditions for the heat load and the air-conditioning system are shown in Tables 1 and 2. In Table 2, the supply air flow rate in which is constant value in this analysis, indicates the sum of the fresh air flow rate (outdoor air) and re-circulated air flow rate (return air) (see Figure 1).

4 Numerical Methods and Boundary Conditions

CFD simulations were carried out using a low Reynolds number type k-ε model. The contaminant concentration distributions were analyzed under the condition of uniform and continuous contaminant generation throughout the space. Numerical conditions are shown in Table 3. In the analysis of annual energy consumption, the climatic conditions in three cities in Japan (Sapporo, Tokyo, and Fukuoka) were targeted.

5 Methods and Conditions for Energy Consumption Analysis

In the analysis of annual energy consumptions, two cases were set for the HVAC conditions according to the supply air flow rate (HVAC (A): (1) Cooling (-10˚C)+ (3) Heating (+10˚C), and HVAC (B): (2) Cooling (- 3˚C)+ (4) Heating (+ 3˚C) in Table 2). Description of the building features and HVAC conditions are shown in Table 4. Table 5 shows the operating conditions of the HVAC equipment and of the annual energy consumed in the eight-story office building. The scale of target office building is one of the typical one among the mesoscale building in Japan. This building has a total of 48 rooms. The index for the equivalent hour on full load is used for the analysis of energy demand as indicated in Table 4. The heat load of the air-conditioning system is assumed to be constant in this analysis. In general, the outdoor fresh air load will become a larger heat load than the others. We introduced the index for normalized concentration in the occupied zone (*Cn*) into the ventilation design and analysis of the annual energy consumption. It is possible to reduce the outdoor fresh air load by using the *Cn* index. However, it is also possible that the fresh air load will increase when the ventilation efficiency is lower (ineffective) than under perfect mixing condition.

6 Results and Discussion

The predicted results for the flow field, temperature, and contaminant concentration distribution in Case 4 ((1) Cooling (-10 [˚C]) condition) are shown in Figure 4. Other analytical results are omitted due to limitations of paper space.

6.1 Analysis Results for *Cn* and Mean CO_2 Concentration

Target 1: Analysis results for *Cn* under the all fresh air condition (All OA) are shown in Table 6. In this case, the recirculated air flow rate of the HVAC system is 0 [%]. As for the supply air flow rate, see in Table 2. The grayed cells in Table 6 are

Table 3 Numerical conditions

Mesh	CASE 1: 78(x)×72(y)×35(z)＝196,560 CASE 2,3: 67(x)×63(y)×27(z)＝113,967 CASE 4: 72(x)×68(y)×51(z)＝249,696 CASE 5: 64(x)×64(y)×27(z)＝110,592
Turbulence Model	Low Re type k-εmodel
Inflow Boundary Conditions	U_{in}: see Note, $k_{in} = 3/2(U_{in}\times0.05)^2$, $\varepsilon_{in} = C_\mu \cdot k_{in}^{3/2} / L_{in}$, $C_\mu = 0.09$, $L_{in} = 1/7$ scale length of supply opening
Outflow Boundary Conditions	U_{out}, k_{out}, ε_{out} = free slip
Wall Treatment	No slip
Contaminant	CO_2, generation rate =0.02m^3/h/person (restful work , 0.2 person/m^2) Passive, uniformly and continuously generated throughout the space

Table 4 Target building and HVAC system

Size	Floor area: 4000 m^2 (= 48 office rooms in Fig. 3)
Number of Floors	8
HVAC conditions (see Table 2)	HVAC (A): Low supply air flow rate (1) Cooling (-10[°C])+ (3) Heating (+10[°C]) HVAC (B): High supply air flow rate (2) Cooling (- 3[°C])+ (4) Heating (+ 3[°C])
Energy Demand	Equivalent operating hours on full load Sapporo: Cooling: 533 [h], Heating; 425 [h] Tokyo, Fukuoka: Cooling: 608 [h], Heating; 293 [h]

Table 5 Conditions of annual office building consumption

Heat Source	Direct fired double effect absorption water chiller boiler, Hot water boiler Depreciation Period: 15 years Annual Interest Rate: 3 [%]
Energy Consumption Rate	Electricity: 10.26 [MJ/kW] CO_2 Emission Rate: 143 [g-C/kW] Gas : 46.5 [MJ/m^3] CO_2 Emission Rate: 858 [g-C/m^3]

Table 6 Analysis results for *Cn* under all OA condition [-]

Supplied Air Conditions (diff. of temp. ⊿θ)	Recirc. Air Rate [%]	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
(1) Cooling (-10)	0	0.94	0.98	0.78	0.82	0.80
(2) Cooling (- 3)	0	0.93	1.01	1.19	0.87	0.76
(3) Heating (+10)	0	1.14	1.01	1.42	0.85	1.13
(4) Heating (+ 3)	0	0.92	1.01	1.21	0.94	0.81

* grayed cells: *Cn* above 1.0 for low ventilation efficiency

Table 7 Analysis results of averaged CO_2 conc. in occupied zone under all OA condition [ppm]

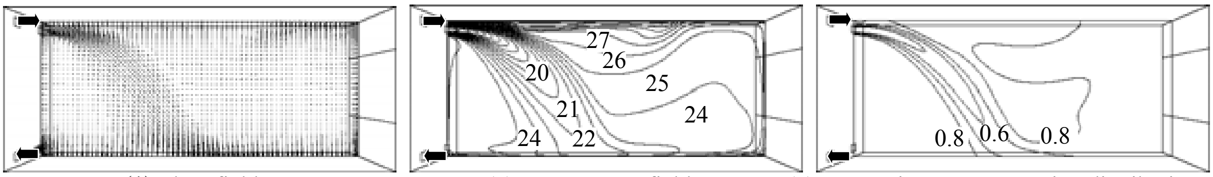
Supplied Air Conditions (diff. of temp.⊿θ)	Recirc. Air Rate [%]	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
(1) Cooling (-10)	0	437	456	363	381	372
(2) Cooling (- 3)	0	358	388	458	334	292
(3) Heating (+10)	0	537	476	669	401	533
(4) Heating (+ 3)	0	355	390	468	363	313

* grayed cells: *Cn* above 1.0 for low ventilation efficiency

Table 8 Analysis results of *Cn'* (Min.OA [30 m^3/h person]) [-]

Supplied Air Conditions (diff. of temp.⊿θ)	Recirc. Air Rate [%]	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
(1) Cooling (-10)	83	0.97	0.99	0.90	0.92	0.91
(2) Cooling (- 3)	95	0.97	1.00	1.07	0.95	0.91
(3) Heating (+10)	82	1.06	1.00	1.19	0.93	1.06
(4) Heating (+ 3)	95	0.97	1.00	1.08	0.98	0.93

* grayed cells: *Cn* above 1.1 or below 0.9



2m/s (1) Flow field (2) Temperature field (3) Contaminant concentration distribution

Figure 4 Prediction results for Case 4 (1)

for the case where *Cn* exceeds 1.0, and therefore it becomes necessary to increase the fresh air flow rate from the view point of ventilation efficiency. The analysis results for the average CO_2 concentration in the occupied zone under the all fresh air condition is shown in Table 7. The average CO_2 concentration in the occupied zone is below 1000 [ppm] in all cases. A *Cn* value above 1.0 means that the ventilation efficiency is worse than for perfect mixing conditions. However, the average CO_2 concentration in the occupied zone is less than 1000 [ppm] for every case. As a matter of course, when we design the fresh air flow rate, we should examine both *Cn* and the average CO_2 concentration in the occupied zone.

Target 2: Analysis results for *Cn* under the condition of the minimum fresh air flow rate (Min.OA) is shown in Table 8. Here, *Cn'* is the *Cn* value which takes into consideration the recirculated air flow of the HVAC system. The grayed cells in Table 8 indicate *Cn'* values which exceed 1.0 ± 0.1. Analysis results for the average CO_2 concentration in the occupied zone under the Min.OA condition are shown in Table 9. The average CO_2 concentration in the occupied zone exceeded 1000 [ppm] in 9 cases out of the total of 20 cases analyzed. These cases keep the fresh air flow rate at 30 [m^3/h person] according to the Act for the Maintenance of Sanitation in Buildings in Japan. When an incompletely-mixed concentration distribution is formed in the room, cases which exceed the design standard CO_2 concentration exist. The HVAC designer should examine both the total fresh air flow rate by using the *Cn* index and the mean CO_2 concentration in the occupied space. In this analysis, the average CO_2 concentration at the exhaust opening is 1017 [ppm] under the condition of 0.2 [person / m^2].

Target 3: Perfectly-mixed conditions are assumed in this analysis. Therefore, the average CO_2 concentration in the occupied zone is equal to that at the exhaust opening.

Target 4: Analysis results for *Cn'* maintaining the average CO_2 concentration in the occupied zone at 1000 [ppm] are shown in Table 10. When the average CO_2 concentration in the occupied zone is lower than the perfectly-mixed concentration of 1000 [ppm], *Cn'* shows below 1.0. In this case, we can reduce the fresh air flow rate until the average CO_2 concentration in the occupied zone reaches 1000 [ppm]. That is, it is possible to save energy by maintaining the air quality level in the occupied zone. Table 11 shows the fresh air flow rate calculated by using *Cn'* as shown in Table 10. The fresh air flow rate in CASE 4 is maintained lower in all supply air conditions ((1)-(4)).

Table 9 Analysis results of averaged CO_2 conc. under Min.OA [30 m^3/h person] condition [ppm]

Supplied Air Conditions (diff. of temp.⊿θ)	Recirc. Air Rate [%]	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
(1) Cooling (-10)	83	989	1007	914	933	924
(2) Cooling (- 3)	95	990	1021	1090	967	924
(3) Heating (+10)	82	1083	1021	1215	946	1078
(4) Heating (+ 3)	95	986	1021	1098	993	943

* Fresh air flow rate needs at least 492 m^3 for maintaining average CO_2 conc. at exhaust opening 1000 [ppm]

* Grayed cells: Average CO_2 conc. at exhaust above 1000 [ppm]

Table 10 Analysis results for *Cn'* to maintain average CO_2 conc. in occupied zone 1000 [ppm] [-]

Supplied Air Conditions (diff. of temp.⊿θ)	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
(1) Cooling (-10)	0.97	0.99	0.90	0.92	0.91
(2) Cooling (- 3)	0.97	1.00	1.07	0.95	0.91
(3) Heating (+10)	1.07	1.00	1.20	0.93	1.06
(4) Heating (+ 3)	0.97	1.00	1.08	0.98	0.93

* Grayed cells: *Cn* above 1.0

Table 11 Fresh air flow rate to maintain average CO_2 conc. in occupied zone 1000 [ppm] [m^3/h]

Supplied Air Conditions (diff. of temp.⊿θ)	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
(1) Cooling (-10)	472	485	426	436	431
(2) Cooling (- 3)	473	495	555	457	431
(3) Heating (+10)	548	495	708	444	544
(4) Heating (+ 3)	470	495	563	476	443

* Volume of fresh air keeps value of 492 [m^3] to maintain exhaust mean CO_2 conc. at a value of 1000 [ppm]

* Grayed cells: Volume of fresh air more than 492 [m^3/h]

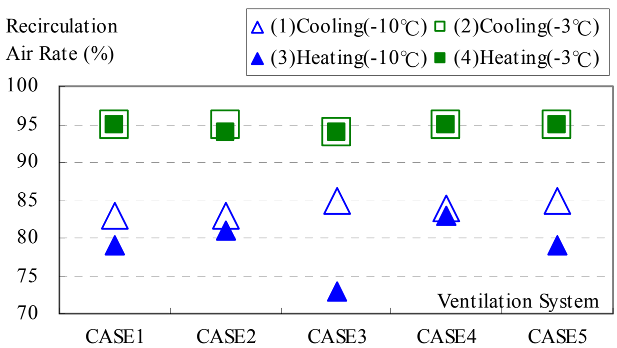


Figure 5 Recirculated air flow rate to maintain the average CO_2 concentration in occupied zone at 1000 [ppm] for each case

6.2 Upper Limit of the Recirculated Air Flow Rate in the HVAC System

Figure 5 shows the percentage of the recirculated air flow rate in the supplied air. This percentage shows the maximum recirculated air flow rate to maintain the average CO_2 concentration in the occupied zone at 1000 [ppm]. A high percentage for the recirculated air flow rate means that the fresh air flow rate in the supplied air is low. In this case, a reduction in energy consumption is achieved by lowering the fresh air flow rate. The percentage of the recirculated air flow rate in CASE 4 shows a comparatively high value under all cooling and heating conditions.

6.3 Predicted Results for the Differences in the Fresh Air Flow Rates

The differences in the fresh air flow rate in each case as an increase or decrease from the fresh air flow rate under completely-mixed conditions (=492 [m^3/h]) are shown in Table 12. For CASE 4(1) - (4) in Table 12, the fresh air flow rate can be reduced during the year. In this case, the heat load of the fresh air is decreased, and this can be expected to conserve energy. Table 13 shows the fresh air heat load in three cities for each target (Targets 1 to 4 in Figure 2). The cooling load in Fukuoka is bigger than that in Tokyo. The heating load in Sapporo is bigger than that in Tokyo.

6.4 Definition of Increase Ratio in the Heat Load

In this analysis, the increase ration in the heat load is defined as follows:

$$\eta = \left(\frac{L_a + L_r}{L_b + L_r} - 1 \right) \times 100 \qquad (4)$$

Here, η is the increase ratio in the heat load [%], L_a is the fresh air heat load under each analysis condition [W], L_b is the fresh air heat load under perfectly mixed conditions [W], and L_r is the heat load indoors [W].

6.5 Analysis Results for the Increase Ratio in the Heat Load

The increase ratio in the heat load with the change in the fresh air flow rate is shown in Table 14 for the three cities. The effect of energy conservation can be expected in CASE 4 under HVAC condition (A) and in CASE5 under HAVC (B) from the viewpoint of the increase in the heat load. In CASE 4 and In CASE5, the increase in the heat load is about -3 [%] as shown in Tables 14-1, 14-2, and 14-3. In CASE 1 under HVAC condition (B), the increase in the heat load is -0.5 [%] and some energy conservation can be expected. For CASE 3 under HVAC condition (B) in Table 14, the increase in the heat load is +4 [%], therefore the energy consumption will increase in this case.

Table 12 Difference of fresh air flow rate compared to that under conditions of perfectly mixed contaminants [m^3/h]

Supplied Air Conditions (diff. of temp.⊿θ)	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
(1) Cooling (-10)	-20	-7	-66	-56	-61
(2) Cooling (- 3)	-19	3	63	-35	-61
(3) Heating (+10)	56	4	216	-48	52
(4) Heating (+ 3)	-22	3	71	-16	-49

* Grayed cells indicate the possibility to reduce the fresh air by means of low *Cn* (high ventilation efficiency)

Table 13 Heat load of fresh air for each climatic condition (Includes sensible heat load and latent heat load) [W]

Target	HVAC Conditions		Fresh Air Load Per Office Room (Sapporo)	(Fukuoka)	(Tokyo)
Target 1 (All OA)	HVAC (A)	Cooling	25,640	32,329	27,870
		Heating	44,182	30,279	30,897
	HVAC (B)	Cooling	85,470	107,767	92,903
		Heating	147,263	100,921	102,981
Target 2 (Min OA)	HVAC (A), (B)	Cooling	4,418	5,570	4,802
		Heating	8,036	5,507	5,620
Target 3 (Ex.1000)	HVAC (A), (B)	Cooling	4,528	5,709	4,922
		Heating	8,237	5,645	5,760
Target 4 (CA.1-CA.5)	HVAC (A), (B)	Cooling	(Fresh air load) = *Cn'* × (fresh air load of Target 3)		
		Heating			
Outdoor Conditions (°C /RH)	Summer		29.0/66	33.3/63	32.6/64
	Winter		–9.4/70	1.5/56	0.6/362

Table 14-1 Increase in heat load with change in fresh air flow rate in Sapporo [%]

HVAC Conditions	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
HVAC(A)	-1.2	-0.4	-4.1	-3.5	-3.8
(diff. of temp 10[°C])	5.5	0.4	21.2	-4.7	5.1
HVAC(B)	-1.2	0.2	3.9	-2.2	-3.8
(diff. of temp 3[°C])	-2.2	0.3	7.0	-1.6	-4.8

* Grayed cells indicate possibility to reduce the heat load by low *Cn'*

Table 14-2 Increase in heat load with change in fresh air flow rate in Fukuoka [%]

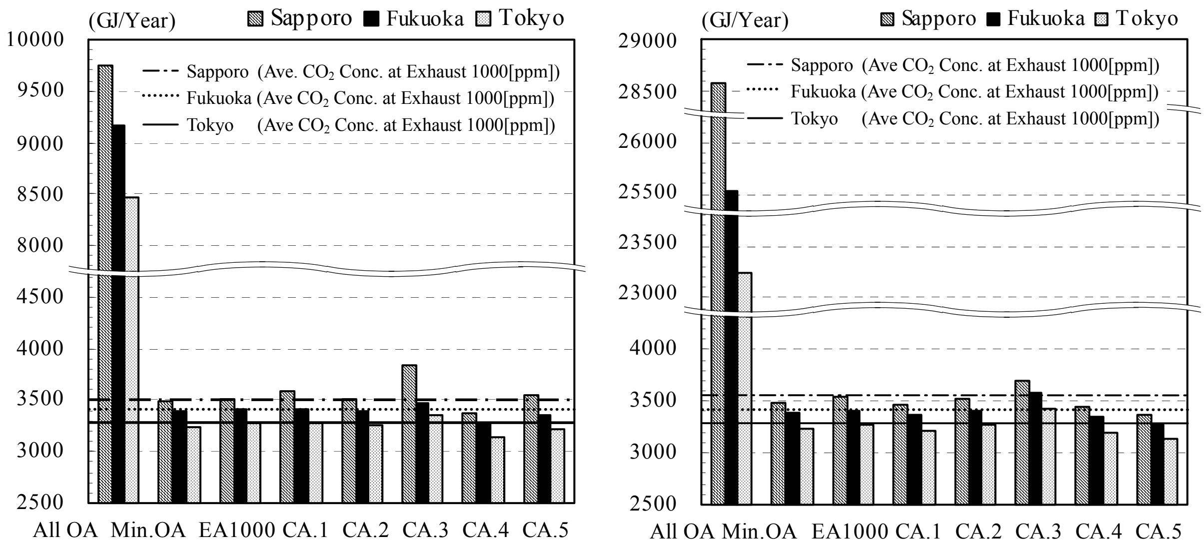
HVAC Conditions	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
HVAC(A)	-1.5	-0.5	-4.8	-4.1	-4.4
(diff. of temp 10[°C])	5.4	1.3	18.2	-2.9	5.1
HVAC(B)	-1.4	0.2	4.6	-2.6	-4.4
(diff. of temp 3[°C])	-0.6	1.4	6.9	-0.1	-2.8

* Grayed cells indicate the possibility to reduce the heat load by low *Cn'*

Table 14-3 Increase in heat load with change in fresh air flow rate in Tokyo [%]

HVAC Conditions	CASE 1	CASE 2	CASE 3	CASE 4	CASE 5
HVAC(A)	-1.3	-0.5	-4.4	-3.7	-4.0
(diff. of temp 10[°C])	4.5	0.3	17.3	-3.8	4.2
HVAC(B)	-1.3	0.2	4.2	-2.3	-4.0
(diff. of temp 3[°C])	-1.8	0.2	5.7	-1.3	-3.9

* Grayed cells indicate the possibility to reduce the heat load by low *Cn'*



(1) HVAC condition (A)
((1) Cooling (-10 [℃]) + (3) Heating (+10 [℃])

(2) HVAC condition (B)
((2) Cooling (-3[℃]) + (4) Heating (+3[℃]))

Min.OA : 30[m^3/h・person]
EA1000: Average CO_2 conc. at exhaust 1000 [ppm]
CA.1 – CA.5: Average CO_2 conc. in occupied zone 1000 [ppm]
All.OA: Supplied air full of fresh air

Broken line: Sapporo Annual Energy Consumption in EA1000.
Dotted line: Fukuoka Annual Energy Consumption in EA1000.
Solid line: is Tokyo Annual Energy Consumption in EA1000.

Figure 6 Prediction results for annual energy consumption

6.6 Predicted Results for Annual Energy Consumption in Each City

The analysis results for the annual energy consumption are shown in Figure 6. The annual energy consumption in Case 4 was minimized in all cities. In Case 4 (HVAC(A)) and CASE5(HVAC(B)) in which *Cn* are applied, the annual energy consumption decreased by 4 % compared with the case of perfect mixing conditions. As a matter of course, in the case with low ventilation efficiency (*Cn*>1.0), the annual energy consumption increased.

7 Concluding Remarks

(1) When the fresh air flow rate is calculated using the condition of 30 [m^3/h/person], the average CO_2 concentration in the occupied zone occasionally exceeds 1000 [ppm]. The HVAC designer should examine both the total fresh air flow rate by using the *Cn* index and the mean CO_2 concentration in the occupied space.
(2) In Case 4, *Cn* was below 1.0 under both cooling and heating conditions of all HVAC conditions. In Case1 and in Case5, *Cn* was below 1.0 under cooling and heating conditions of only HVAC(B).
(3) The analysis of the annual energy consumption was carried out under the condition of an average CO_2 concentration 1000 [ppm] in the occupied zone. In this analysis, the annual energy consumption in CASE 4 (HVAC(A)) and in CASE 5 (HVAC(B)) were reduced by 4 [%] compared to perfectly-mixed conditions.

References
[1] N Kobayashi, S Kikuchi, K Ito (2002). Numerical Analysis of Normalized Concentration in the Occupied Zone for Various Office Ventilation Systems. *ROOM-VENT2002*, Proceedings, pp. 129-132.
[2] E-passplan, E&E Planning Co. Ltd.

Note CFD Cases

Case	Supplied air Condition (Difference of temp.)	Target Temp. [℃]	Inlet Temp [℃]	Air ex-change rate [h^{-1}]	$U_{in.}$ [m/s]	*Ar*	*Re* ($\times10^6$)
1-1	Cooling (-10℃)	26	16	12.6	1.07	-0.76	0.21
1-2	Cooling (- 3℃)	26	23	42.0	3.58	-0.02	0.69
1-3	Heating (+10℃)	22	32	11.9	1.02	0.85	0.20
1-4	Heating (+ 3℃)	22	25	39.8	3.39	0.02	0.65
2-1	Cooling (-10℃)	26	16	12.6	1.07	-0.76	0.21
2-2	Cooling (- 3℃)	26	23	42.0	3.58	-0.02	0.69
2-3	Heating (+10℃)	22	32	11.9	1.02	0.85	0.20
2-4	Heating (+ 3℃)	22	25	39.8	3.39	0.02	0.65
3-1	Cooling (-10℃)	26	16	12.6	0.19	-23.6	0.04
3-2	Cooling (- 3℃)	26	23	42.0	0.65	-0.64	0.12
3-3	Heating (+10℃)	22	32	11.9	0.18	26.29	0.04
3-4	Heating (+ 3℃)	22	25	39.8	0.61	0.71	0.12
4-1	Cooling (-10℃)	26	16	12.6	2.15	-0.19	0.41
4-2	Cooling (- 3℃)	26	23	42.0	7.17	-0.01	1.38
4-3	Heating (+10℃)	22	32	11.9	2.04	0.02	0.39
4-4	Heating (+ 3℃)	22	25	39.8	6.79	0.09	1.31
5-1	Cooling (-10℃)	26	16	12.6	0.06	-220	0.01
5-2	Cooling (- 3℃)	26	23	42.0	0.21	-5.94	0.04
5-3	Heating (+10℃)	22	32	11.9	0.06	245	0.01
5-4	Heating (+ 3℃)	22	25	39.8	0.20	6.62	0.04