A Sensitivity Analysis on Mixing Energy Loss in Air-Conditioned Rooms by Using CFD

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ABSTRACT

In some office buildings, in winter, heating and cooling often operate in perimeter and interior zones simultaneously. In such situations, mixing energy loss often occurs. This loss is defined as the difference between the sum of the net heating/cooling load and that of the actual heating/cooling energy supplied to the air-conditioned room. A mixing energy loss increases the energy consumption of air-conditioning; therefore, it is very important to prevent occurrences of mixing energy loss for energy-saving strategies.

In this study, we performed a sensitivity analysis on the mixing energy loss (or mixing energy gain) in air-conditioned rooms by using computational fluid dynamics (CFD) techniques. Particularly, the effects of (1) the setting temperatures in the perimeter and interior zones, (2) the locations of reference temperature points in the perimeter and interior zones, and (3) the shape of air-conditioned room on mixing energy loss (or mixing energy gain) were investigated. To evaluate the mixing energy loss/gain, we conducted two simulations for each test case; one was a simulation without airflow mixing between the perimeter and interior zones, and the other was a simulation with airflow mixing between the two zones. In the simulation without airflow mixing, we introduced a virtual adiabatic wall at the boundary between the perimeter and interior zones. In the simulation with airflow mixing, on the other hand, we removed the virtual adiabatic wall. Finally, by comparing the sensible heat loads between the two simulations (the cases with and without airflow mixing), the mixing energy loss/gain was estimated.

KEYWORDS

Mixing energy loss/gain, Interior and perimeter zones, Computational fluid dynamics, Sensitivity analysis

INTRODUCTION

Effective energy management and energy saving strategies are now essential in the context of the issue of global warming. In office buildings, further improvement in the efficiency of air-conditioning systems is required to achieve high energy savings. To achieve effective and sophisticated air-conditioning, it is essential to understand the detailed spatial and temporal

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distributions of indoor environment. CFD is a very powerful tool to analyze/predict detailed indoor thermal and airflow environments spatially and temporally.

In this study, we introduced CFD analysis to assess mixing energy loss in air-conditioned rooms where heating and cooling operated in the perimeter and interior zones simultaneously. A mixing energy loss is defined as the difference between the sum of the net heating/cooling load and that of the actual heating/cooling energy supplied to the room. It increases the energy consumption of air-conditioning; therefore, to prevent occurrences of mixing energy loss is a key issue for energy-saving strategies.

To evaluate mixing energy loss (or mixing energy gain) by using CFD, we conducted two simulations. One was a simulation without airflow mixing between the perimeter and interior zones. The other was a simulation with airflow mixing between the two zones. By comparing the sensible heat loads between the two simulations (the cases with and without airflow mixing), the mixing energy loss/gain was estimated. Here, we specifically investigated the effects of (1) the setting temperatures in the perimeter and interior zones, (2) the locations of reference temperature points in the perimeter and interior zones, and (3) the shape of air-conditioned room on mixing energy loss/gain.

OUTLINE OF CFD SIMULATIONS

Room models

Two room models (A and B) were used in this study. The only difference between the two rooms was the depth. Rooms A and B were 10.0 m (width; x direction) × 10.0 m (depth; y direction) × 2.9 m (height; z direction) and 10.0 m (width) × 20.0 m (depth) × 2.9 m (height), respectively. Figure 1 is an illustration of Room A.

In both models, the depth of the perimeter zone was set to 3 m, and two fan coil units (FCU) were installed on the floor of each perimeter zone for heating, while supply inlets (Room A: 6 inlets, Room B: 8 inlets) from an air handling unit (AHU) were mounted on the ceiling of each interior zone for cooling. In short, heating and cooling operated simultaneously in both room models. The supply air volumes of FCU and AHU were constant (uniform); however, note that the amounts were different depending on the setting temperature and the size of the room model (cf. Table 2).

Human bodies (60 W/person), lighting (20 W/m^2), and equipment (Interior: 20 W/m^2, Perimeter: 10 W/m^2) were considered as the internal heat sources. All walls in both room models were adiabatic except for the window, making it the only place in which heat transfer occurs. The air temperature outside the window was set at 4.5 °C.

Simulated cases

Table 1 shows the simulated cases (16 cases). In this study, as described above, the focus was on the effects of (1) the setting temperatures in the perimeter and interior zones, (2) the locations of reference temperature points in the perimeter and interior zones, and (3) the shape of air-conditioned room on mixing energy loss (or mixing energy gain).

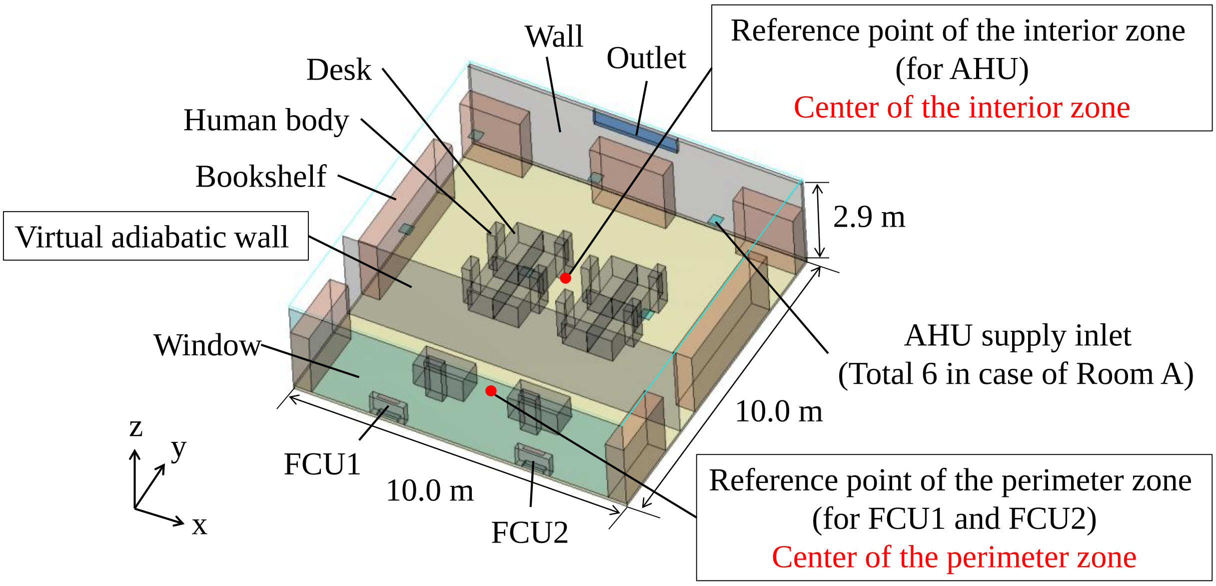

Figure 1. *Room model A*

Table 1. *Simulated cases*

	(1) Setting temperature		(2) Reference temperature point	(3) Shape of room model (Width : Depth)
	Perimeter	Interior		
Case A1	22 °C	22 °C	Center of each zone FL+1.0m	Room A 1 : 1 (10m : 10m)
Case A2		24 °C		
Case A3	24 °C	22 °C		
Case A4		24 °C		
Case A5	22 °C	22 °C	Center of each zone FL+0.5m	
Case A6		24 °C		
Case A7	24 °C	22 °C		
Case A8		24 °C		
Case B1	22 °C	22 °C	Center of each zone FL+1.0m	Room B 1 : 2 (10m : 20m)
Case B2		24 °C		
Case B3	24 °C	22 °C		
Case B4		24 °C		
Case B5	22 °C	22 °C	Center of each zone FL+0.5m	
Case B6		24 °C		
Case B7	24 °C	22 °C		
Case B8		24 °C		

The setting temperatures in the perimeter and interior zones were 22 °C or 24 °C. The reference temperature points were in the center of the perimeter and interior zones with two heights, FL+1.0 m or FL+0.5 m. The only difference between the shape of Room A and Room B was the depth. As described above, Room A and Room B were 10 m and 20 m deep, respectively.

In this study, to evaluate the mixing energy loss (or mixing energy gain), we conducted two simulations for each test case, one without and one with airflow mixing between the perimeter and interior zones. In the simulation without airflow mixing, we introduced a virtual adiabatic wall at the boundary between the perimeter and interior zones (cf. Figure 1). In the simulation with airflow mixing, on the other hand, we removed the virtual adiabatic wall. Finally, by comparing the sensible heat loads between the simulations with and without airflow mixing, the mixing energy loss (or mixing energy gain) was estimated.

CFD analysis conditions

The airflow and thermal simulations in the room models (cf. Figure 1) were conducted using commercial CFD software, STREAM ver.8. The analysis conditions are shown in Table 2.

Table 2. *CFD analysis conditions*

Room model	**Room A**	**Room B**
Grid points	90 (x) × 90 (y) × 30 (z) = 243,000	90 (x) × 161 (y) × 30 (z) = 434,700
Scheme for convection terms	QUICK scheme for all governing equations	
Turbulence model	RNG k-ε model (high-Reynolds number type)	
Inlet boundary condition	**FCU (Perimeter zone)**	
	Setting temperature: 22 °C	**Setting temperature: 22** °C
	Velocity : 0.076 m^3/s (273 m^3/h) Temperature : PID control (24-36 °C) k : 1.41×10^{-2} m^2/s^2 ε : 6.62×10^{-2} m^2/s^3	Velocity : 0.076 m^3/s (273 m^3/h) Temperature : PID control (24-36 °C) k : 1.41×10^{-2} m^2/s^2 ε : 6.62×10^{-2} m^2/s^3
	Setting temperature: 24 °C	**Setting temperature: 24** °C
	Velocity : 0.121 m^3/s (435 m^3/h) Temperature : PID control (24-36 °C) k : 3.59×10^{-2} m^2/s^2 ε : 2.68×10^{-2} m^2/s^3	Velocity : 0.121 m^3/s (435 m^3/h) Temperature : PID control (24-36 °C) k : 3.59×10^{-2} m^2/s^2 ε : 2.68×10^{-2} m^2/s^3
	AHU (Interior zone)	
	Setting temperature: 22 °C	**Setting temperature: 22** °C
	Velocity : 0.076 m^3/s (273 m^3/h) Temperature : PID control (12-20 °C) k : 8.25×10^{-2} m^2/s^2 ε : 2.95×10^{-3} m^2/s^3	Velocity : 0.095 m^3/s (343 m^3/h) Temperature : PID control (12-20 °C) k : 1.30×10^{-2} m^2/s^2 ε : 5.84×10^{-2} m^2/s^3
	Setting temperature: 24 °C	**Setting temperature: 24** °C
	Velocity : 0.057 m^3/s (205 m^3/h) Temperature : PID control (12-20 °C) k : 4.64×10^{-3} m^2/s^2 ε : 1.24×10^{-3} m^2/s^3	Velocity : 0.072 m^3/s (258 m^3/h) Temperature : PID control (12-20 °C) k : 7.32×10^{-3} m^2/s^2 ε : 2.47×10^{-3} m^2/s^3
Outlet boundary condition	Zero-gradient conditions for all variables	
Wall boundary condition	Velocity : Logarithmic law Temperature : Overall heat transfer coefficient for window was 4.8 W/m^2K. For other walls, adiabatic conditions were used.	
Heat generation	Human body : 60 W/person Lighting : 20 W/m^2 Equipment : 20 W/m^2 (Perimeter zone), 10 W/m^2 (Interior zone)	

We used the RNG k-ε model (high-Reynolds number type) as the turbulence model. The total number of grid points was 243,000 (Room A) and 434,700 (Room B). In the CFD simulations conducted here, the supply air temperatures for FCU and AHU were determined by a PID control (every 30 seconds) based on the reference temperatures.

RESULTS AND DISCUSSION

The following results were those at 3,600 seconds from the start of the CFD simulations. The indoor environment at that time was considered as a statistically steady state.

Air temperature distributions and amount of mixing energy loss/gain

As an example, Figure 2 shows the vertical distributions of the air temperature at the center section (x = 5.0 m) for Case A5. Figure 2(1) illustrates the result of the simulation "without airflow mixing" (with the virtual adiabatic wall) between the perimeter and interior zones. Figure 2(2) shows the results of the simulation "with airflow mixing" (without the virtual adiabatic wall) between the perimeter and interior zones.

In Figure 2(1), there are large differences between the perimeter and interior zones due to the existence of the virtual adiabatic wall. In the perimeter zone (heating zone), an apparent thermal stratification is formed. On the other hand, in the interior zone (cooling zone), the air in the entire interior zone is well mixed because unstable flow arises by the effect of the cold supply air from the ceiling.

In Figure 2(2), airflow mixing arises between the perimeter and interior zones because there is no partition at the boundary. The warm air in the perimeter zone flows into the upper region of the interior zone. As a whole, a thermal stratification is formed in the entire room.

Tables 3 (Room A) and 4 (Room B) show the results of sensible heat loads for FCU and AHU and the amounts of mixing energy loss/gain. Here, concerning the mixing energy loss/gain, the positive values indicate a mixing energy loss, and the negative values, a mixing energy gain.

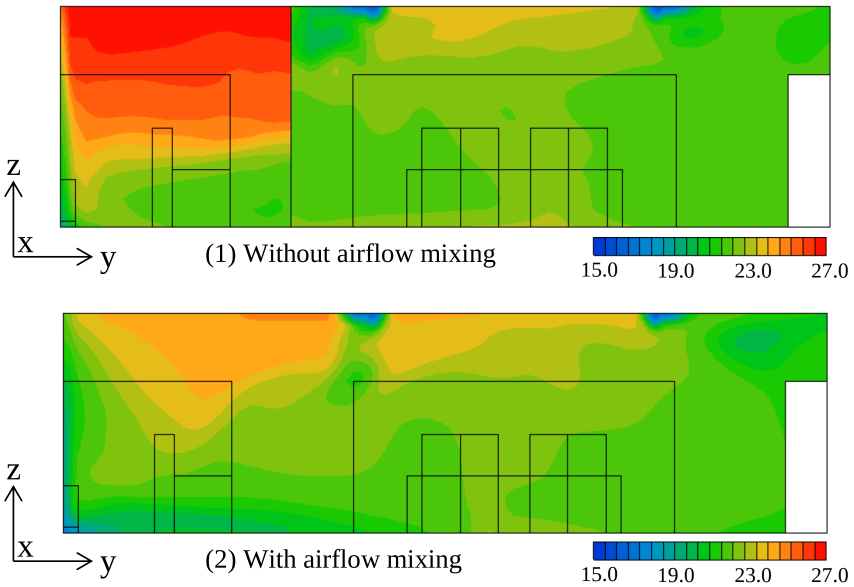

Figure 2. *Vertical distribution of the air temperature in Case A5*

Table 3. *Sensible heat loads and amount of mixing energy loss/gain in Cases A1-A8*

	Airflow mixing	Sensible heat load [kW]			Total sensible heat load [kW]	Mixing energy loss [kW]
		FCU1	FCU2	AHU		
Case A1	Without airflow mixing	0.54	0.54	3.10	4.18	0.09
	With airflow mixing	0.63	0.64	3.00	4.27	
Case A2	Without airflow mixing	0.55	0.55	2.94	4.05	-1.32
	With airflow mixing	0.23	0.21	2.29	2.73	
Case A3	Without airflow mixing	0.59	0.59	3.00	4.18	3.53
	With airflow mixing	1.63	1.61	4.47	7.71	
Case A4	Without airflow mixing	0.58	0.58	2.93	4.09	1.19
	With airflow mixing	0.85	0.86	3.57	5.28	
Case A5	Without airflow mixing	0.62	0.62	2.97	4.21	0.41
	With airflow mixing	0.74	0.73	3.15	4.62	
Case A6	Without airflow mixing	0.62	0.61	2.94	4.17	-1.07
	With airflow mixing	0.34	0.34	2.42	3.10	
Case A7	Without airflow mixing	0.65	0.62	3.03	4.31	3.96
	With airflow mixing	1.81	1.81	4.65	8.27	
Case A8	Without airflow mixing	0.67	0.67	2.93	4.27	1.49
	With airflow mixing	1.02	1.03	3.71	5.76	

Table 4. *Sensible heat loads and amount of mixing energy loss/gain in Cases B1-B8*

	Airflow mixing	Sensible heat load [kW]			Total sensible heat load [kW]	Mixing energy loss [kW]
		FCU1	FCU2	AHU		
Case B1	Without airflow mixing	0.52	0.52	7.50	8.55	0.49
	With airflow mixing	0.60	0.61	7.82	9.04	
Case B2	Without airflow mixing	0.52	0.52	7.12	8.17	-1.19
	With airflow mixing	0.21	0.17	6.60	6.98	
Case B3	Without airflow mixing	0.60	0.61	7.65	8.86	3.36
	With airflow mixing	1.48	1.47	9.26	12.22	
Case B4	Without airflow mixing	0.59	0.60	7.13	8.33	0.51
	With airflow mixing	0.70	0.70	7.44	8.84	
Case B5	Without airflow mixing	0.62	0.62	7.58	8.81	0.67
	With airflow mixing	0.71	0.71	8.07	9.49	
Case B6	Without airflow mixing	0.61	0.61	7.28	8.50	-0.81
	With airflow mixing	0.35	0.34	7.00	7.69	
Case B7	Without airflow mixing	0.64	0.60	7.64	8.88	4.34
	With airflow mixing	1.73	1.73	9.76	13.22	
Case B8	Without airflow mixing	0.66	0.64	7.19	8.49	1.12
	With airflow mixing	0.90	0.90	7.82	9.61	

Effect of the setting temperatures in the perimeter and interior zones

In all cases in which the setting temperature in the perimeter zone is 2 °C higher than that in the interior zone (Cases A3 and A7, and Cases B3 and B7), mixing energy loss occurs. The amounts of mixing energy loss in Cases A3, A7, B3, and B7 are much larger than those in other test cases. When the setting temperature in the perimeter zone is higher than that in the interior zone, the effect of the location of the reference temperature point and that of the shape

of air-conditioned room introduced in this study on mixing energy loss become relatively small.
In those cases in which the setting temperatures in the perimeter and interior zones are the same (Cases A1, A4, A5, and A8, and Cases B1, B4, B5, and B8), mixing energy loss also occurs. As reported below, in these cases, the amount of mixing energy loss becomes smaller as the location of the reference temperature point is lower.
On the other hand, in all cases in which the setting temperature in the perimeter zone is 2 °C lower than that in the interior zone (Cases A2 and A6, and Cases B2 and B6), mixing energy gain occurs. As reported in previous studies (e.g., Nakahara et al. 1987), such treatment (setting temperature in the perimeter zone < setting temperature in the interior zone) is effective to prevent mixing energy loss.

Effect of the location of the reference temperature point

In all cases in which mixing energy loss occurs, the amount of mixing energy loss becomes larger as the location of the reference temperature point is lower (Cases A1 and A5, Cases A3 and A7, Cases A4 and A8, Cases B1 and B5, Cases B3 and B7, and Cases B4 and B8).
In the perimeter zone, even in the case with airflow mixing, a relatively apparent thermal stratification is formed (cf. Figure 2 (2)). Therefore, the required supply heat increases to satisfy the setting temperature as the location of the reference temperature point is lower; namely, the supply air temperature of FCU becomes higher. The warmer air from FCU flows into the upper region of the interior zone, and, thus, the supply air temperature of AHU becomes lower. As a result, the amount of mixing energy loss is larger as the location of the reference temperature point is lower. Due to the similar physical structure, in all cases in which mixing energy gain occurs, the amount of mixing energy gain becomes smaller as the location of the reference temperature point is lower (Cases A2 and A6, and Cases B2 and B6).

Effect of the shape of the air-conditioned room

By comparing the cases with the same number for Cases A and B, the effect of the shape of air-conditioned room on the mixing energy loss/gain becomes evident. Although the amounts of mixing energy loss/gain differ slightly in each comparison, there is not a significant difference between Cases A and B. Mixing energy loss/gain occurs through the boundary between the perimeter and interior zones. In this study, however, we set the same area of the boundary between the perimeter and interior zones for Cases A and B (Rooms A and B); therefore, there seems to be little difference created by the shape of the room introduced in this study.

CONCLUSION

In this study, we performed a sensitivity analysis on mixing energy loss/gain in air-conditioned rooms in which heating and cooling operated in the perimeter and interior zones simultaneously by using CFD. Needless to say, it is very difficult to evaluate mixing energy loss/gain correctly with simple heat load calculations (heat balance analyses) which are usually performed on the basis of the assumption of complete mixing. CFD can analyze detailed information of thermal and airflow environments needed for estimating mixing energy loss/gain accurately.

We investigated the effects of (1) the setting temperatures in the perimeter and interior zones, (2) the locations of reference temperature points in the perimeter and interior zones, and (3) the shape of air-conditioned room on mixing energy loss/gain. The effect of the setting temperature was the largest, in particular, in the case in which the setting temperature in the perimeter zone was higher than that in the interior zone. The effect of the location of the reference temperature point was relatively large. The amount of mixing energy loss (mixing energy gain) became larger (smaller) as the location of the reference temperature point was lower. The effect of the shape of the room was not clear in this study. Further investigation into the effect of room shape is still required.

REFERENCE

Nakahara, N., Kajiwara, T., Ito, H. 1987. Studies on prevention of mixing energy loss in air-conditioned room, Part 1: Experimental study with actual sized equipment, Transaction of the Society of Heating, Air-Conditioning Sanitary Engineers of Japan, 33, 1-12 (in Japanese).