COUPLING STRATEGY OF HVAC SYSTEM SIMULATION AND CFD PART 2: STUDY ON MIXING ENERGY LOSS IN AN AIR-CONDITIONED ROOM

Satoru Iizuka[1], Mina Sasaki[1], Gyuyoung Yoon[2], Masaya Okumiya[1],
Junya Kondo[3], and Yuka Sakai[4]
[1]Nagoya University, Nagoya, Japan
[2]Nagoya City University, Nagoya, Japan
[3]Kajima Corporation, Tokyo, Japan
[4]Toyama City Hall, Toyama, Japan

ABSTRACT

A coupled analysis of heating, ventilation and air-conditioning (HVAC) system simulation tool and computational fluid dynamics (CFD) model was carried out to assess the mixing energy loss in an air-conditioned room where heating and cooling operate in the perimeter and interior zones simultaneously. To evaluate the mixing energy loss, we conducted two simulations; one was the case with airflow mixing between the perimeter and interior zones and the other was the case without airflow mixing between the zones. By comparing the required coil loads between the two cases, the mixing energy loss was estimated. The accuracy of the coupled analysis was assessed by comparing its results with those from an experiment conducted by Ito (1988).

INTRODUCTION

In recent years, effective energy management and energy saving strategies are essential in the context of the issue of global warming. In office buildings, for instance, further improvement in the efficiency of air-conditioning system is required in order to achieve high energy saving.

To examine the performance of air-conditioning system, various simulation tools (e.g., Life Cycle Energy Management (LCEM) tool, HVACSIM+, TRNSYS, DeST, EnergyPlus) have been developed. The system simulations performed with those tools are usually carried out under the assumption of complete mixing in the target room. However, without the consideration of spatial and temporal distributions of indoor environment, it is difficult to perform more sophisticated energy saving control.

Coupled analysis of HVAC system simulation and CFD is a promising strategy to overcome the above problem. CFD is considered to be a powerful tool for analyzing airflow and thermal environments, therefore detailed information of flow field and air temperature can be obtained. As reviewed in another paper for this conference (Yoon et al., 2011), several intensive studies on coupled analysis of HVAC system simulation and CFD have been conducted (e.g., Lam et al., 2001; Zhai et al., 2003, 2005, 2006; Iida et al., 2008).

In this study, coupled simulations of a HVAC system simulation tool and a CFD model were carried out to assess the mixing energy loss in an air-conditioned room where heating and cooling operate in the perimeter and interior zones simultaneously. The 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.

To evaluate the mixing energy loss, we conducted two simulations; one was the case with airflow mixing between the perimeter and interior zones and the other was the case without airflow mixing between the zones. By comparing the required coil loads between the two cases, the mixing energy loss was estimated.

Furthermore, the effects of the reference points of temperature in the perimeter and interior zones on the mixing energy loss were investigated. The accuracy of the coupled analysis was assessed by comparing its results with those from an experiment conducted by Ito (1988).

OUTLINE OF THE COUPLED SIMULATIONS

Room model and simulated cases

The room model used during the experiment conducted by Ito (1988) was the target of this study. Figure 1 shows an illustration of the facility with its size being 6.0 m (Width; x direction) × 3.9 m (Depth; y direction) × 2.6 m (Height; z direction). The room model was divided equally in two partitions: perimeter and interior zones.

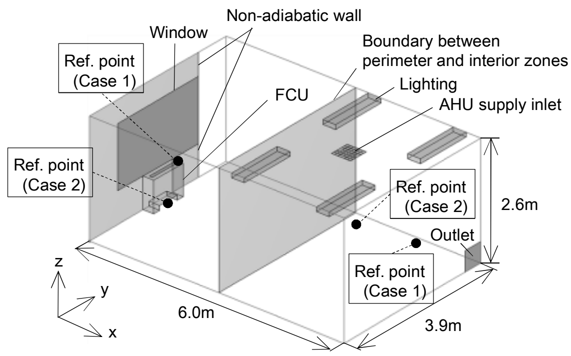


Figure 1 Room model

A fan coil unit (FCU) was installed on the floor of the perimeter zone for heating, while a supply inlet from an air handling unit (AHU) was mounted on the ceiling of the interior zone for cooling. The supply air volumes of FCU and AHU were constant (FCU: 320 m^3/h, AHU: 325 m^3/h). The lighting in the interior zone was considered as the only internal heat source (320 W).

All walls were adiabatic except the window and its surrounding wall in the perimeter zone, making it the only place where heat transfer occurs. The air temperature outside the window and its surrounding wall was set at 5 °C.

The setting temperature in both perimeter and interior zones was 19 °C. The reference points of temperature, as shown in Figure 1, were different for the two test cases (Cases 1 and 2). In Case 1, the reference points in the perimeter and interior zones were located near the supply inlet of FCU and at the center of the wall with an exhaust outlet, respectively, while in Case 2, the reference points in the perimeter and interior zones were set at the suction opening of FCU and at the center of the interior zone, respectively.

In the experiment conducted by Ito (1988), for both Cases 1 and 2, additional tests with a vinyl-sheet partition at the boundary between the perimeter and interior zones were carried out. The mixing energy loss in the experiment was estimated by comparing the required coil loads between the cases with and without the vinyl-sheet partition.

In the coupled simulations conducted in this study, we performed additional simulations with a "virtual" adiabatic wall at the boundary between the perimeter and interior zones. Like the estimation in the experiment, we assessed the mixing energy loss in the room model by comparing the required coil loads between the cases with and without the virtual adiabatic wall.

HVAC system simulation

The HVAC system simulations were performed using LCEM tool ver.3.02 developed under the supervision of the Ministry of Land, Infrastructure, Transport and Tourism (MLIT) of Japan. Table 1 shows the specifications of FCU (perimeter zone) and AHU (interior zone) determined based on the results of heat load calculations.

Table 1 Specifications of FCU and AHU

FCU (PERIMETER ZONE)	
Supply air volume	: 320 m^3/h
Hot water volume	: 5.0 *l*/min
Hot water temperature	: 55-50 °C
Heating capacity	: 3.2 kW
AHU (INTERIOR ZONE)	
Supply air volume	: 325 m^3/h
Chilled water volume	: 3.6 *l*/min
Chilled water temperature	: 7-12 °C
Cooling capacity	: 1.3 kW

CFD

The airflow and thermal simulations in the room model (cf. Figure 1) were conducted using a commercial CFD software, STREAM ver.8. The analysis conditions are shown in Table 2. We used the standard k-ε model as the turbulence model. The total number of the grid points was 70,848. In both Cases 1 and 2, the value of the wall coordinate at the first grid point adjacent to the wall was around 20-50.

Table 2 CFD analysis conditions

Grid points	64 (x) × 41 (y) × 27 (z) = 70,848
Scheme for convection terms	1st-order upwind scheme for all governing equations
Turbulence model	Standard k-ε model
Inlet boundary condition	**FCU (PERIMETER ZONE)** Velocity : 320 m^3/h Temp. : PID control (22-26 °C) Humidity : Results from HVAC system simulation k : 1.07×10^{-2} m^2/s^2 ε : 4.68×10^{-3} m^2/s^3
	AHU (INTERIOR ZONE) Velocity : 325 m^3/h Temp. : PID control (14-18 °C) Humidity : Results from HVAC system simulation k : 2.98×10^{-3} m^2/s^2 ε : 4.56×10^{-4} m^2/s^3
Outlet boundary condition	Zero-gradient conditions for all variables
Wall boundary condition	Velocity : Logarithmic law Temp. : Overall heat transfer coefficients for window and its surrounding wall were 3.33 W/m^2K and 0.50 W/m^2K, respectively. For other walls, adiabatic conditions were used. Humidity : Impermeable conditions without condensation
Heat generation	Lighting (interior zone): 320 W

Coupling of HVAC system simulation and CFD

The detailed information on the coupling method of HVAC system simulation (LCEM tool) and CFD (STREAM) is described in another paper for this conference (Yoon et al., 2011).

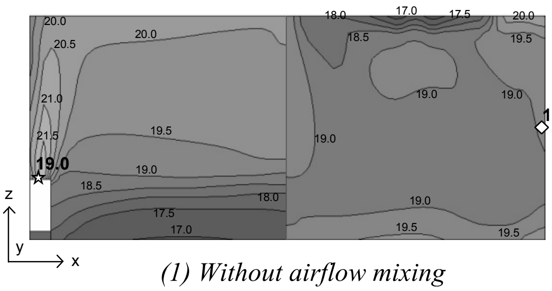
In the coupled simulations conducted here, for both FCU and AHU, the air temperature and humidity at the exhaust outlet calculated by the CFD simulation were imposed to the HVAC system simulation. The humidity at the supply inlet obtained with the HVAC system simulation was given to the CFD simulation as boundary conditions. The temperature at the supply inlet was provided by a PID control. Those processes were performed every 30 seconds.

RESULTS AND DISCUSSION

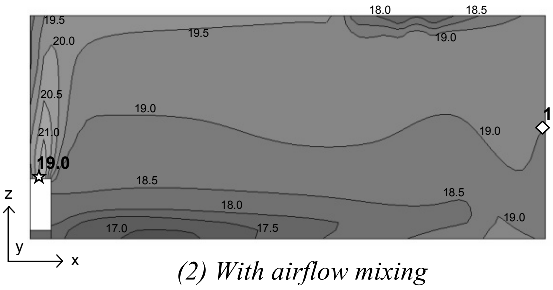
The following results obtained with the coupled simulations were those at 6000 seconds from the start of the simulations. The indoor environment at that time was considered as a statistically steady state.

Air temperature distributions

Figures 2 and 3 show the distributions of air temperature at the center section (y = 1.95 m) obtained with the coupled simulations for Cases 1 and 2, respectively. The star-shaped and diamond-shaped symbols represent the reference points of the temperature. For reference, Figures 4 and 5 depict the distributions of velocity vectors at the same section (y = 1.95 m) for Cases 1 and 2, respectively.

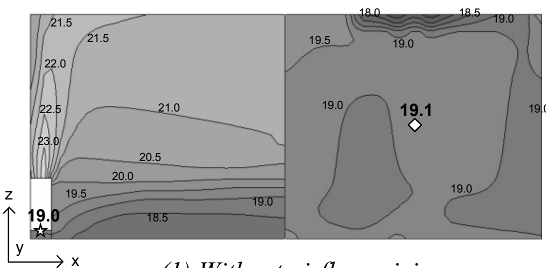


(1) Without airflow mixing

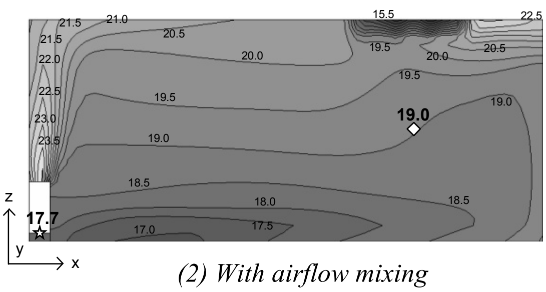


(2) With airflow mixing

Figure 2 Distribution of air temperature in Case 1

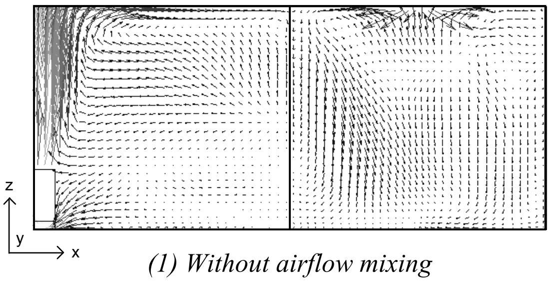


(1) Without airflow mixing

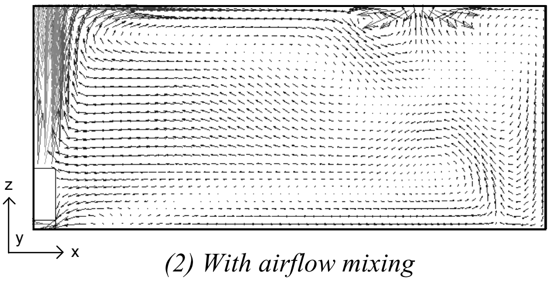


(2) With airflow mixing

Figure 3 Distribution of air temperature in Case 2

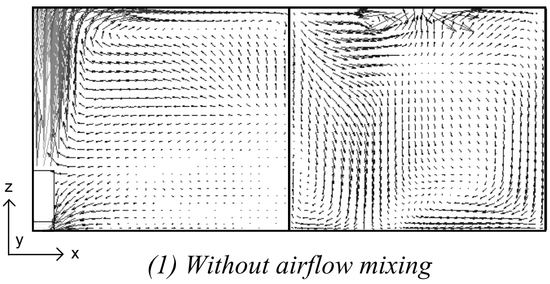


(1) Without airflow mixing

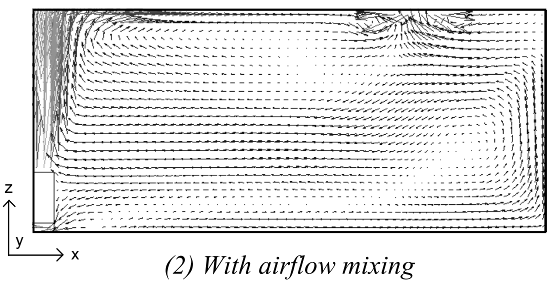


(2) With airflow mixing

Figure 4 Distribution of velocity vectors in Case 1



(1) Without airflow mixing



(2) With airflow mixing

Figure 5 Distribution of velocity vectors in Case 2

Figures 2(1) and 3(1) illustrate the results of the case "without airflow mixing" (with the virtual adiabatic wall) between the perimeter and interior zones. Figures 2(2) and 3(2) show the results of the case "with airflow mixing" (without the virtual adiabatic wall) between the perimeter and interior zones.

In both Figures 2(1) and 3(1), there are large differences between the perimeter and interior zones due to the existence of the virtual adiabatic wall. In the perimeter zone, for both Cases 1 and 2, the warm supply air from FCU rises (cf. Figures 4(1) and 5(1)), and the cold air caused by the heat transfer through the window and its surrounding non-adiabatic wall stays in the lower region. We can observe an apparent thermal stratification in both perimeter zones. On the other hand, in the interior zone, unstable flow is formed by the effect of the cold supply air from the ceiling, and thus the air in the entire interior zone is well mixed for both Cases 1 and 2. The air temperature is around 19 °C (the setting temperature), except for the vicinity of the supply inlet on the ceiling.

In both Figures 2(2) and 3(2), airflow mixing arises between the perimeter and interior zones because there is no partition at the boundary. The warm supply air from FCU rises and flows into the upper region of the interior zone (cf. Figures 4(2) and 5(2)). In the lower region of the perimeter zone, the cold air region expands due to not only the heat transfer through the window and its surrounding non-adiabatic wall but also the cold flow from the interior zone. A remarkable thermal stratification is formed in the entire room in both Figures 2(2) and 3(2).

As for the reference temperature, only the reference temperature in the perimeter zone for Case 2 with airflow mixing is different from the setting temperature (19 °C) (cf. Figure 3(2)). The temperature is 1.3 °C lower than the setting temperature.

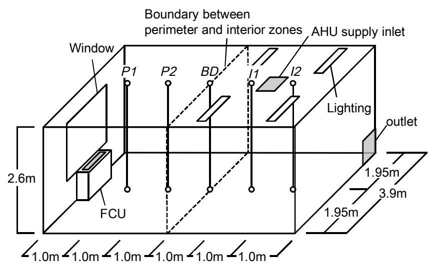
The reference point in the perimeter zone for Case 2 is located in a lower position than that of Case 1. Therefore, the reference temperature in the perimeter zone of Case 2 is often lower and thus the temperature of the supply air from FCU is higher. On the other hand, in case with airflow mixing, the temperature of the supply air from AHU in the interior zone is lower due to the effect of the warmer supply air from FCU. As a result, the reference temperature in the perimeter zone for Case 2 with airflow mixing decreases further.

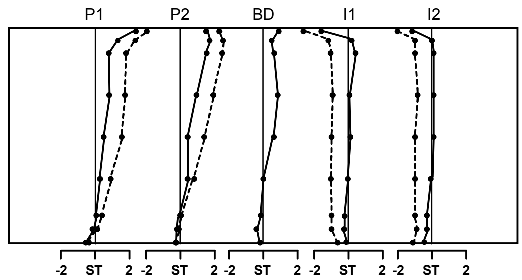
Actually, in Case 2 with airflow mixing, the temperatures of the supply air from FCU and AHU were the maximum value (26 °C) and almost the minimum value (14.1 °C) of the PID controls, respectively.

Figures 6 and 7 compare the vertical profiles of air temperature between the experimental and simulated results ("ST" means the setting temperature (19 °C)). In general, the results obtained with the coupled simulations correspond well to the experimental results of Ito (1988).

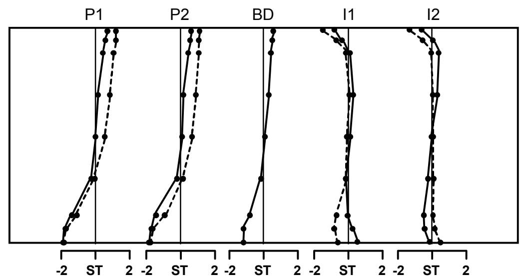
However, there are some differences between the experimental and simulated results; 1) In Cases 1 and 2 both with and without airflow mixing, the temperatures in the lower region of the perimeter zone predicted by the coupled simulations are lower than those of the experiment. Therefore, in the perimeter zone, the temperature differences in the vertical direction given from the coupled simulations are larger than the experimental results. 2) In Cases 1 and 2 without airflow mixing, the temperatures in the interior zone predicted by the coupled simulations are higher than those of the experiment.

Those discrepancies might be related to the problems with the experiment such as air leakage from the vinyl-sheet partition and those with the accuracy of CFD such as the performance of turbulence model.



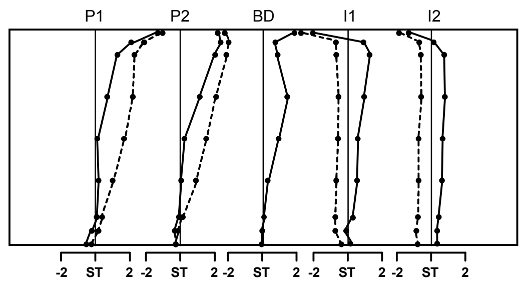


(1) Experiment

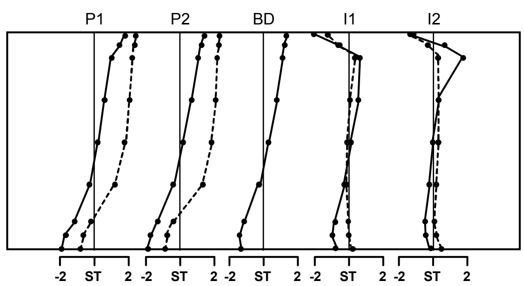


(2) Coupled simulation

Figure 6 Vertical profiles of air temperature in Case 1 (dot line: without airflow mixing; solid line: with airflow mixing)



(1) Experiment



(2) Coupled simulation

Figure 7 Vertical profiles of air temperature in Case 2 (dot line: without airflow mixing; solid line: with airflow mixing)

Table 3 Supply air temperature, return air temperature, and required coil load in the coupled simulations

	PERIMETER ZONE: FCU					
	Without airflow mixing			With airflow mixing		
	Supply air temperature	Return air temperature	Coil load	Supply air temperature	Return air temperature	Coil load
Case 1	23.2 °C	17.7 °C	0.590 kW	22.5 °C	17.7 °C	0.516 kW
Case 2	24.6 °C	19.0 °C	0.622 kW	26.0 °C	17.7 °C	0.894 kW
	INTERIOR ZONE: AHU					
	Without airflow mixing			With airflow mixing		
	Supply air temperature	Return air temperature	Coil load	Supply air temperature	Return air temperature	Coil load
Case 1	16.3 °C	19.0 °C	0.409 kW	17.2 °C	18.8 °C	0.289 kW
Case 2	16.2 °C	19.1 °C	0.436 kW	14.1 °C	19.0 °C	0.654 kW

Table 4 Estimation of mixing energy loss/gain

	PERIMETER ZONE: FCU		**INTERIOR ZONE: AHU**		**TOTAL**	
	Experiment	Simulation	Experiment	Simulation	Experiment	Simulation
Case 1	0.049 kW	-0.074 kW	0.037 kW	-0.120 kW	0.086 kW	-0.194 kW
Case 2	0.312 kW	0.272 kW	0.202 kW	0.218 kW	0.514 kW	0.490 kW

As an example of the investigation into the performance of turbulence model of CFD, Figure 8 shows the results of the RNG k-ε model. The correspondence with the experimental results is worse, compared to the results of the standard k-ε model (cf. Figure 6(2)). In the next phase of this study, we will conduct further investigations into the performance of turbulence model of CFD by introducing more sophisticated models.

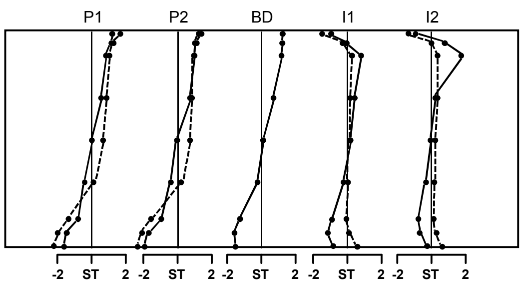


Figure 8 Performance of the RNG k-ε model in Case 1 (dot line: without airflow mixing; solid line: with airflow mixing)

Estimation of mixing energy loss

Table 3 shows the results of supply air temperature, return air temperature, and required coil load obtained with the coupled simulations for Cases 1 and 2 with and without airflow mixing at the boundary between the perimeter and interior zones.

Table 4 compares the mixing energy loss estimated by the experiment (Ito, 1988) and that by the coupled simulations. Here, minus values mean the mixing energy gain. As described above, the mixing energy loss (or gain) was assessed by comparing the required coil loads between the cases with and without the partition (vinyl sheet or adiabatic wall) at the boundary of the perimeter and interior zones.

In Case 1, the mixing energy losses of the perimeter and interior zones estimated by the experiment are 0.049 kW and 0.037 kW, respectively. The total mixing energy loss in the room model is 0.086 kW. On the other hand, in the coupled simulations for Case 1, the mixing energy gain occurs in both perimeter (-0.074 kW) and interior (-0.120 kW) zones. The total mixing energy gain predicted by the coupled simulations is -0.194 kW. In Case 1, therefore, the correspondence of the experimental and simulated results is not good. This implies that it is difficult to predict accurately small amounts of mixing energy loss, although further improvement in the accuracy of the coupled simulation is still required.

In Case 2, the mixing energy losses of the perimeter and interior zones obtained with the experiment are 0.312 kW and 0.202 kW, respectively. The total mixing energy loss in the room model is 0.514 kW. In the coupled simulations for Case 2, the mixing energy loss also occurs in both perimeter (0.272 kW) and interior (0.218 kW) zones. The total mixing energy loss predicted by the coupled simulations is 0.490 kW. In Case 2, the simulated results correspond very well to the experimental results.

CONCLUSION

In this study, the coupled analysis of HVAC system simulation and CFD developed by the authors was applied to estimate the mixing energy loss in an air-conditioned room where heating and cooling operate in the perimeter and interior zones simultaneously.

In case that the mixing energy loss was relatively large, the simulated results of the mixing energy loss were in good agreement with the results obtained with the experiment conducted by Ito (1988). On the other hand, in case that the mixing energy loss was relatively small, the correspondence of the experimental and simulated results was not good.

Although further improvement in the accuracy is still required, such coupled simulations are very useful to analyze indoor environment and to perform effective energy management and energy saving in air-conditioned rooms where complete mixing cannot be assumed.

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