A Coupled Analysis of CFD and HVAC System Simulations for a Multi-Split Air-Conditioning System in the Operation Phase

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ABSTRACT

A coupled analysis system of computational fluid dynamics (CFD) techniques and HVAC system simulation (LCEM tool) was applied for a multi-split air-conditioning system in the operation phase in this study. In particular, the effects of (1) the difference in internal heat sources, (2) that in operational modes (cooling mode/ventilation mode) for indoor units, and (3) that in reference temperature points on the indoor environment and the energy performance of the system were investigated. The effects of energy saving by reducing the internal heat sources and introducing a forced/rotational control of indoor units were quantitatively clarified.

KEYWORDS

Coupled analysis, CFD, HVAC system simulation, Multi-split air-conditioning system, Operation phase

INTRODUCTION

In the context of the issue of global warming, effective energy management and strategies for saving energy are essential. In Japan, particularly after the Fukushima nuclear power plant disaster, high power saving is strongly needed. Under such situations, in office buildings, further improvement of the efficiency of air-conditioning systems is required.

To examine the performance of air-conditioning systems, various simulation tools have been developed. They are quite effective. However, it is difficult to examine detailed and sophisticated energy management and savings by using system simulations exclusively because they are usually performed on the basis of the assumption of complete mixing in the target rooms.

To overcome such problems, we have been developing a coupled analysis system of computational fluid dynamics (CFD) techniques and HVAC system simulation (LCEM tool). CFD can analyze/predict detailed information of indoor thermal and airflow environments needed for detailed and sophisticated energy management and savings of air-conditioning systems. In previous studies, we applied a coupled analysis system of CFD and HVAC system simulations (1) for an under floor air-conditioning system (Yoon et al. 2011) and (2) for

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predicting mixing energy loss in an air-conditioned room in which heating and cooling operate in the perimeter and interior zones simultaneously (Iizuka et al. 2011). The attempts were for the design phase of air-conditioning systems. Through the examples, we clarified the effectiveness of the coupled analysis system and determined that it could accurately analyze/predict both the indoor environment and energy consumption of air-conditioning.

In this study, we applied our coupled analysis system of CFD and HVAC system simulations for a multi-split air-conditioning system in the operation phase. Multi-split air-conditioning systems are currently being used in many office buildings and are expected to achieve considerable energy savings because each indoor unit (hereafter IU) can be individually controlled and operated according to the time schedule of each air-conditioning zone.

On the other hand, there are some operational problems, such as (1) the possibility of reduction in indoor thermal comfort due to interference of supply air between an IU and a total heat exchanger unit (hereafter HEX) or between different IUs and (2) the possibility of a short circuit, i.e., a part of the supply air from IU or HEX flows directly into the suction opening of other IU or HEX. Furthermore, (3) forced On/Off controls will be necessary to save large amounts of power, especially in Japan after the Fukushima nuclear plant disaster. Under such a situation, how to maintain an indoor environment which is not uncomfortable is required. The coupled analysis system of CFD and HVAC system simulations is expected to be effective to examine the problems reported above in the operation phase.

In particular, in this study, the effects of (1) the difference in internal heat sources, (2) that in operational modes for each IU, and (3) that in reference temperature points on the indoor environment and the energy performance of the system were investigated.

OUTLINE OF COUPLED SIMULATIONS

Room model and air-conditioning system

Figure 1 shows the typical floor plan of the target office building in this study. The office building was assumed to be located in Tokyo. The office rooms were arranged in the northern and southern parts of the floor. The air-conditioning area was 1,144 m^2 in the typical floor, and there were four zones in both the northern and southern office areas (N1-N4 and S1-S4).

In this study, only a part of the southern office area, S3 and S4 zones, was analyzed. The size of the S3 and S4 zones was 22 m (width; x direction) × 13 m (depth; y direction) × 2.9 m (height; z direction). The air-conditioning system was a multi-split air-conditioning system in which eight IUs mounted on the ceiling were connected to an outdoor unit. Two HEXs were mounted between IUs in each zone (cf. Figure 2). The target season was summer, and the outdoor air temperature and humidity were set at 30.3 °C and 0.02 kg/kg(DA), respectively.

CFD

The airflow and thermal simulations in the room model (cf. Figure 2) were conducted using commercial CFD software, STREAM ver.8. The radiant environment was also analyzed using this software. To reduce the calculation load of CFD, only the S3 zone (11 m (width) × 13 m (depth) × 2.9 m (height)) was analyzed in CFD. Each IU has four supply inlets for four directions, while each HEX has one supply inlet. All the supply air from IUs and HEXs were provided at an angle of 30 degrees to the ceiling. The heat generation from lighting, equipment, and human bodies was considered as the internal heat sources. Table 1 shows the CFD analysis condition used in this study (NOTE).

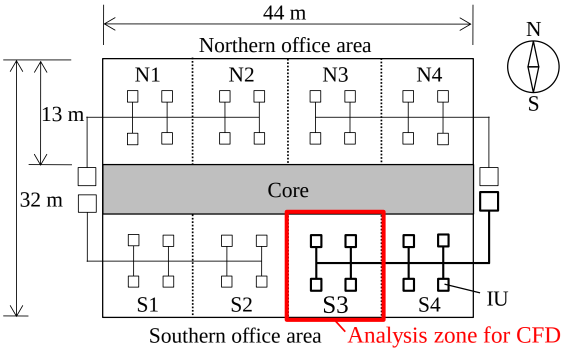


Figure 1*. Typical floor plan of the target office building*

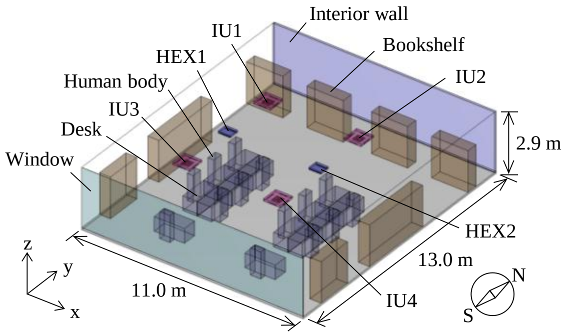


Figure 2*. Room model for CFD*

Table 1. *CFD analysis conditions*

Domain	11.0 m (x) × 13.0 m (y) × 2.9 m (z)	
Grid points	76 (x) × 122 (y) × 24 (z) = 222,528	
Scheme for convection terms	QUICK scheme for all governing equations	
Turbulence model	RNG k-ε model (high-Reynolds number type)	
Inlet boundary condition	**Indoor Unit (IU)** Velocity : $0.19\ m^3/s$ ($690\ m^3/h$) Temperature : P control (12°C (min.) - 18°C (max.)) Humidity : Results from HVAC system simulation k : $8.61 \times 10^{-2}\ m^2/s^2$ (NOTE) ε : $3.98 \times 10^{-2}\ m^2/s^3$	**Total Heat Exchanger Unit (HEX)** Velocity : $0.07\ m^3/s$ ($250\ m^3/h$) Temperature : Return air temperature × 0.5 + Outdoor air temperature × 0.5 Humidity : Return air humidity × 0.5 + Outdoor air humidity × 0.5 k : $1.09 \times 10^{-1}\ m^2/s^2$ ε : $4.36 \times 10^{-2}\ m^2/s^3$
Outlet boundary condition	Zero-gradient conditions for all variables	
Wall boundary condition	Velocity : A symmetry boundary condition was applied for the x direction. For other walls, a logarithmic law was used. Temperature : Overall heat transfer coefficients for window, interior wall and ceiling were 2.080 W/m^2K, 0.760 W/m^2K, and 0.314 W/m^2K, respectively. For other walls, adiabatic conditions were used. Humidity : Impermeable conditions without condensation	
Heat generation	Lighting : 15 W/m^2 or 7.5 W/m^2 Equipment : 10 W/m^2 or 7.0 W/m^2 Human body : 55 W/person (sensible heat) and 63 W/person (latent heat)	

HVAC system simulation

The HVAC system simulations were performed using the LCEM tool ver.3.03 developed under the supervision of Ministry of Land, Infrastructure, Transport and Tourism (MLIT) of Japan (cf. Technical description of the LCEM tool, 2011). Table 2 shows the specifications of the outdoor unit, indoor unit, and total heat exchanger unit. The specifications of the indoor unit were determined based on the results of heat load calculations in the target area. Figure 3 shows the performance characteristics of the outdoor unit. The characteristic curves of capacity and input ratios to the rated values were obtained with the LCEM tool.

Table 2. *Specifications of outdoor unit, indoor unit, and total heat exchanger unit*

Outdoor Unit (Package type)	**Indoor Unit (Package type)**	**Total Heat Exchanger Unit**
Cooling capacity : 22.4 kW	Supply air volume : $0.19\ m^3/s$	Supply air volume : $0.07\ m^3/s$
Heating capacity : 25.0 kW	Cooling capacity : 2.80 kW	Power consumption : 0.097 kW
Cooling power consumption : 5.73 kW	Heating capacity : 3.10 kW	Sensible heat exchange efficiency : 50 %
Heating power consumption : 6.05 kW	Cooling power consumption : 0.05 kW	Enthalpy exchange efficiency : 50 %
	Heating power consumption : 0.05 kW	

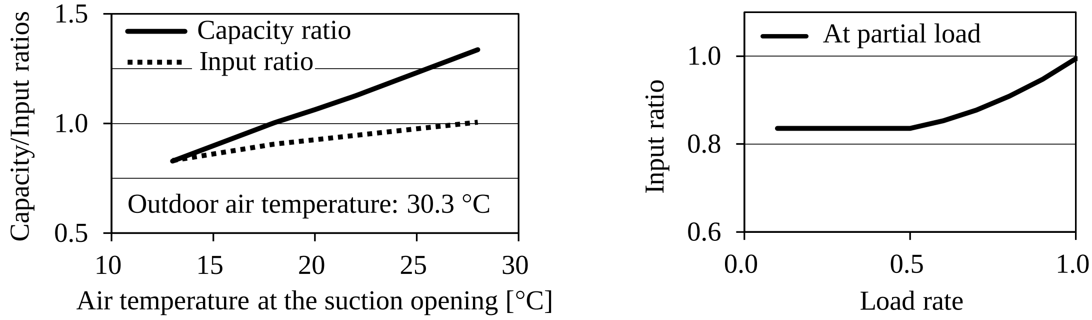


Figure 3. *Performance characteristics of the outdoor unit*

Coupling method of CFD and HVAC simulations

In the coupled simulations conducted here, the air temperature and humidity at the suction openings of IUs, which were attached to each IU, calculated by CFD were given to the HVAC system simulation. The supply air temperature of each IU was provided by a proportional (P) control according to the temperature at the reference point that was set at the suction opening of each IU. The humidity at the supply inlet of each IU obtained with the HVAC system simulation, which was calculated based on the coil surface air temperature and humidity, the coil outlet air temperature, the supply air temperature, and the return air temperature and humidity (cf. Technical description of the LCEM tool, 2011), was given to the CFD simulation as the boundary condition. These processes were performed every 30 seconds. Here, note that, in this study, the minimum and maximum temperatures of the supply air of IUs were set at 12 °C and 18 °C, respectively. The setting temperature in the room was 26 °C. Furthermore, for IUs, two operational modes (cooling/ventilation) were introduced by an On/Off control. The On/Off control used in this study switched to the cooling mode when the temperature at the reference point reached 28 °C and above and to the ventilation mode when it reached 24 °C and below.

For HEXs, the temperature/humidity at the supply inlet was provided by (the air temperature/humidity at the suction opening) $\times 0.5 +$ (the outdoor air temperature/humidity) $\times$ 0.5. The values were calculated every one second.

Simulated cases

Table 3 shows the simulated cases in this study (6 cases). Case 1 was a standard case. In Cases 2 and 3, the situation of power saving was assumed, in which heat generation from the lighting and equipment was reduced by 50 % and 30 %, respectively. Furthermore, a forced/rotational On/Off control was used in Case 3. For the rotational control, one of the four IUs was forced to switch from the cooling mode to the ventilation mode at a regular interval (15 minutes) in order of IU1 (northwest), IU2 (northeast), IU3 (southwest), and IU4 (southeast). In Cases 1-3, the reference temperature points were set at the suction opening of each IU.

Cases 4, 5, and 6 corresponded to the conditions of Cases 1, 2, and 3, respectively; however, the locations of the reference temperature points were different. In Cases 4-6, the reference temperature points were the same and were located in the vicinity of the walls with a height of FL + 1.5 m (for IU1: (x, y) = (0.01 m, 11.0 m), for IU2: (x, y) = (10.99 m, 11.0 m), for IU3: (x, y) = (0.01 m, 0.6 m), for IU4: (x, y) = (10.99 m, 0.6 m)).

All the coupled simulations were conducted for 7,200 seconds (2 hours).

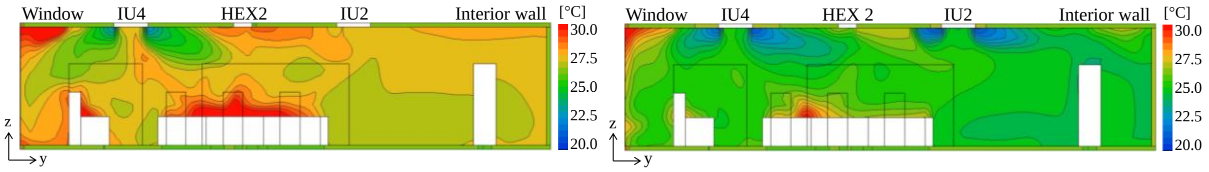
__Table 3.__ Simulated cases

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Internal heat sources: Lighting Equipment	 15 W/m^2 10 W/m^2	 7.5 W/m^2 7.0 W/m^2	 7.5 W/m^2 7.0 W/m^2	 15 W/m^2 10 W/m^2	 7.5 W/m^2 7.0 W/m^2	 7.5 W/m^2 7.0 W/m^2
Rotational control	Without	Without	With	Without	Without	With
Reference temperature point	Suction opening of each IU			Vicinity of wall (FL + 1.5 m)		

RESULTS AND DISCUSSION

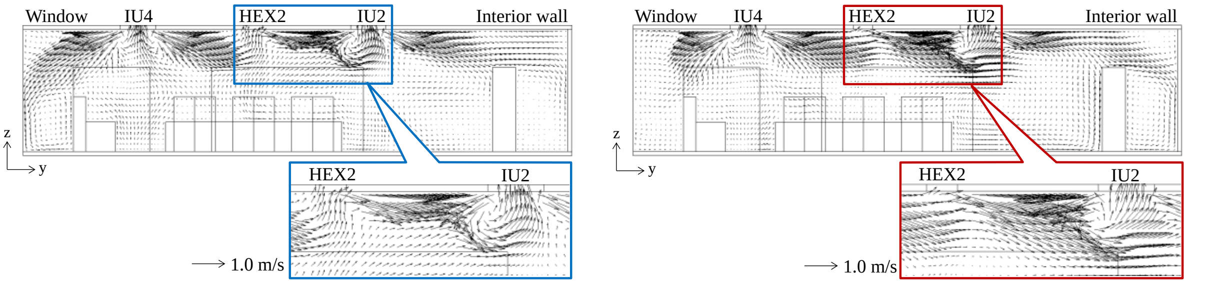
Thermal and airflow environments

Figures 4 and 5 illustrate examples of the results of the indoor environment. Figure 4 shows the vertical distributions of the air temperature at x = 7.75 m in Case 1, including IUs 2 and 4, and HEX2, at 1,800 seconds (Figure 4(1)) and 7,200 seconds (Figure 4(2)) from the start of the coupled simulation. At 1,800 seconds (Figure 4(1)), as a whole, the air temperature increases due to the supply air which is not cold from IU2 (ventilation mode). On the other hand, at 7,200 seconds (Figure 4(2)), both IUs 2 and 4 switch to the cooling mode and the air in the entire room is well mixed due to the cold supply air from IUs 2 and 4. The air temperature in the occupied zone (z ≤ 1.7 m) generally satisfies the setting temperature (26 °C) at 7,200 seconds.



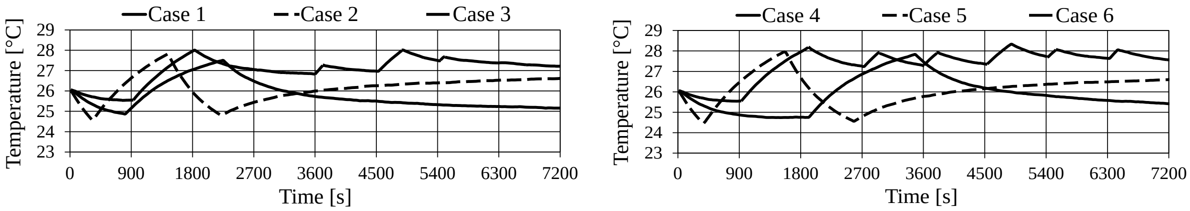
(1) 1,800 seconds *(2) 7,200 seconds*

__Figure 4.__ Vertical distributions of the air temperature in Case 1 (x = 7.75 m)



(1) 1,800 seconds *(2) 7,200 seconds*

__Figure 5.__ Vertical distributions of the velocity vectors in Case 1 (x = 7.75 m)



(1) Cases 1, 2, and 3 *(2) Cases 4, 5, and 6*

__Figure 6.__ Time histories of the space-averaged air temperatures of the entire room

Figure 5 depicts the vertical distributions of the velocity vectors at x = 7.75 m in Case 1 at 1,800 seconds (Figure 5(1)) and 7,200 seconds (Figure 5(2)) from the start of the coupled simulation. At both moments, interference of the supply air from IU2 and that of HEX2 can be observed. A short circuit, in which the supply air from HEX2 flows directly into the suction opening of IU2, occurs at both moments (cf. enlargements in Figure 5). At 1,800 seconds, another short circuit, in which the supply air from IU4 flows directly into the suction opening of HEX2, can also be seen. Furthermore, we can observe that the supply air from IU2 to the opposite side of HEX2 flows along the ceiling at both moments (the Coanda effect).
Figure 6 shows the time histories of the space-averaged air temperatures of the entire room. In all cases, the space-averaged air temperatures fluctuate in the range of about 25 °C to 28 °C according to the operational modes (cooling/ventilation).

Time history of the operational mode

Figure 7 shows the time history of the operational mode for each IU (IUs 1-4) in all cases. In the figure, colors are used to indicate the cooling mode, and white is used for the ventilation mode. In Case 2 (reduction of the internal heat sources and no rotational control), the number of IU in the ventilation mode is larger than that in Case 1. In Case 3 (reduction of the internal heat sources and rotational control), the number of IU and the amount of time in the ventilation mode are larger than those in Cases 1 and 2. The same tendency is observed in the comparison among Cases 4-6 (the difference between Cases 1-3 and Cases 4-6 is the location of reference temperature point). Such temporal variations of the operational mode are due to the unsteadiness of the indoor environment analyzed by CFD.
Table 4 shows the amount of time in the cooling mode, the amount of supplied heat, and the mean return air temperature averaged over the amounts of time in the cooling mode in each IU for 7,200 seconds. In Case 2, the amount of time in the cooling mode for IU1 is much smaller than that in Case 1. After switching the ventilation mode for IU1 at 2,100 seconds from the start of the coupled simulation in Case 2, the return air temperature (the reference temperature) of IU1 remains under 28 °C (the temperature to switch the cooling mode) due to the effect of the reduction in the internal heat sources and that of the cold supply air from IU2, and, as a result, IU1 continues to operate in the ventilation mode from 2,100 seconds to 7,200 seconds. As shown in Table 4, in Case 2, the reduction in the internal heat sources causes a significant difference in the heat loads for IU1 and IU2. The total amounts of supplied heat from all IUs are 19.8 kWh and 15.6 kWh in Cases 1 and 2, respectively. The amount in Case 2 is 4.2 kWh smaller than that in Case 1.
On the other hand, in Case 3, the amount of time in the cooling mode for each IU is not significantly different because the cooling/ventilation modes were switched rotationally. The total amount of supplied heat from all IUs in Case 3 is the smallest (14.5 kWh) among Cases 1-3 and is 1.1 kWh smaller than that in Case 2.
In Cases 4, 5, and 6, the total amounts of supplied heat from all IUs are 19.7 kWh, 15.7 kWh, and 14.8 kWh and are almost identical to those in Cases 1, 2, and 3, respectively. This suggests that the location of the reference temperature point has a small effect on the total amount of supplied heat from IUs.

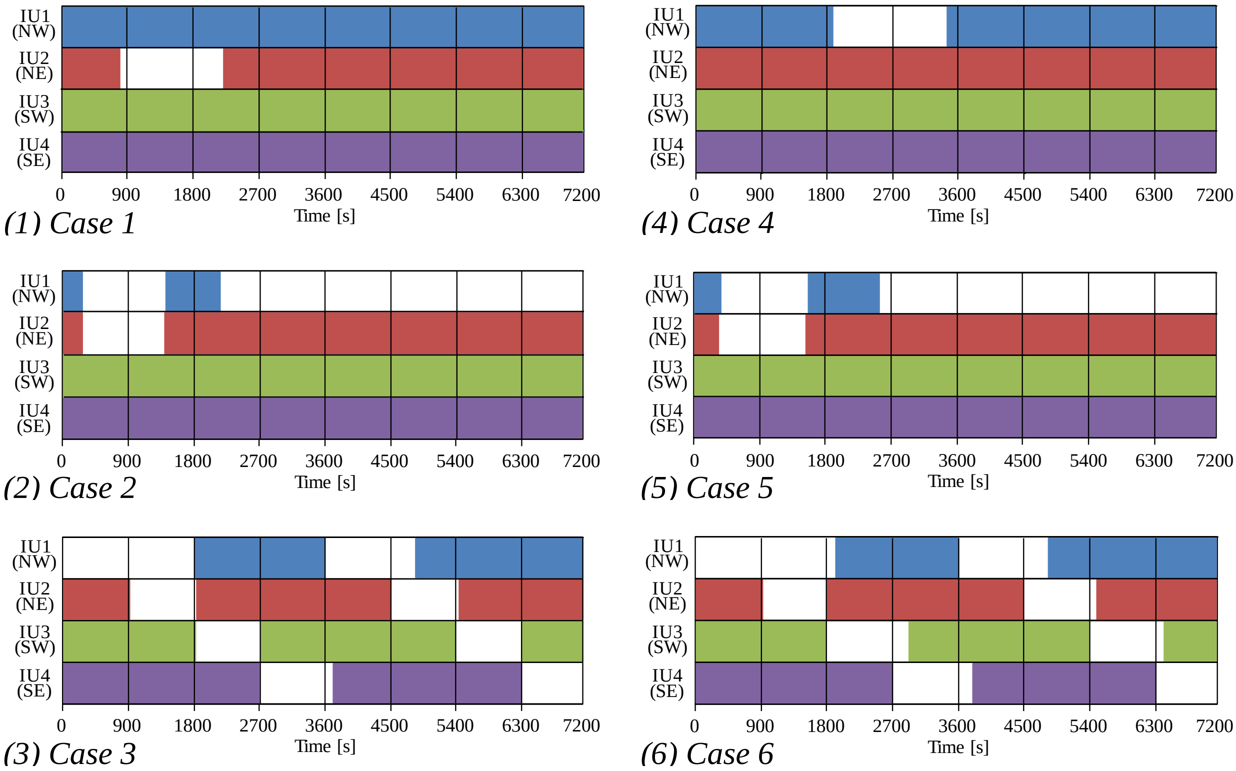


Figure 7. *Time histories of operational mode (colors: cooling mode; white: ventilation mode)*

Table 4. *Amount of time in the cooling mode, amount of supplied heat, and mean return air temperature*

	Cooling mode [min]				Heat supply [kWh]				Mean return air temperature [°C]			
	IU1	IU2	IU3	IU4	IU1	IU2	IU3	IU4	IU1	IU2	IU3	IU4
Case 1	120	96	120	120	5.21	4.17	5.21	5.21	25.3	25.1	25.8	25.9
Case 2	18	102	120	120	0.78	4.41	5.21	5.21	25.3	25.5	26.2	25.9
Case 3	78	86	90	80	3.38	3.73	3.90	3.45	26.9	26.8	27.1	27.2
Case 4	94	120	120	120	4.08	5.21	5.21	5.21	25.2	25.4	26.1	26.0
Case 5	22.5	100	120	120	0.98	4.34	5.21	5.21	25.2	25.4	26.1	25.9
Case 6	67.5	88.5	84.5	87.5	2.92	3.84	3.66	3.79	27.4	26.9	27.2	27.1

System energy performance

Table 5 shows the operational status of the outdoor unit in all cases. In Cases 1, 2, and 3, the outputs of the outdoor unit are 41.4 kWh, 32.6kWh, and 30.2 kWh, respectively. By reducing the internal heat sources, the amount of power consumption is decreased by 2.2 kWh (in the comparison of Cases 1 and 2). Furthermore, by introducing the rotational control, the amount of power consumption is decreased by 0.6 kWh (in the comparison of Cases 2 and 3). The results of the operational status of the outdoor unit in Cases 4, 5, and 6 are almost the same as those in Cases 1, 2, and 3, respectively. As for the COP of the outdoor unit, Case 3 has the highest value of 4.99, mainly due to the higher air temperature at the suction openings of IUs. Figure 8 shows the total amounts of power consumption in the target area (S3 and S4 in Figure 1) for all cases. In Cases 1, 2 and 3, the total amounts are 24.8 kWh, 16.9 kWh, and 16.3 kWh, respectively. The energy consumption was decreased by 32 % by reducing the internal heat sources (cf. Table 3). In addition to the reduction in the internal heat sources, by introducing the rotational control, the reduction ratio in energy consumption becomes 34 %.

In this study, a shutdown mode for IUs is not introduced; however, a reduction in the power consumption of IUs is also expected if the shutdown mode, rather than the ventilation mode, is introduced.

Table 5. *Operational status of outdoor unit*

	Output [kWh]	Power consumption [kWh]	COP
Case 1	41.4	8.9	4.65
Case 2	32.6	6.7	4.88
Case 3	30.2	6.1	4.99
Case 4	41.2	8.8	4.67
Case 5	32.9	6.8	4.86
Case 6	29.7	6.0	4.95

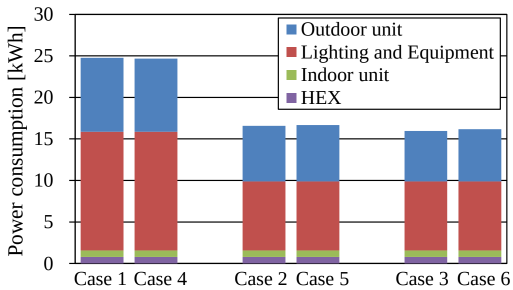


Figure 8. *Total amounts of power consumption in the target area*

CONCLUSION

The coupled analysis system of CFD and HVAC system simulations was applied for a multi-split air-conditioning system in the operation phase in this study. In particular, the effects of (1) the difference in internal heat sources, (2) that in operational modes (cooling /ventilation) for indoor units, and (3) that in reference temperature points on the indoor environment and the energy performance of the system were investigated.

By introducing CFD analysis, operational problems in the air-conditioning system, such as interferences of the supply air between an indoor unit (IU) and a total heat exchanger unit (HEX) and short circuits of the airflow between IU and HEX, were confirmed. Furthermore, the effects of energy saving by reducing internal heat sources and introducing a forced/rotational control of IUs were quantitatively clarified.

ACKNOWLEDGEMENTS

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NOTE

The value of the turbulent kinetic energy at the inlet boundary condition for each indoor unit was determined based on experimental data conducted by Daikin Industries, Ltd.

REFERENCES

Building Equipment and Environment Division, Government Buildings Department, Minister’s Secretariat Ministry of Land, Infrastructure and Transport. 2011. The technical description of the LCEM tool ver.3.03 (in Japanese).

Iizuka S., Sasaki M., Yoon G., Okumiya M., Kondo J., Sakai Y. 2011. Coupling strategy of HVAC system simulation and CFD, Part 2, Study on mixing energy loss in an air-conditioned room, Proceedings of Building Simulation 2011, pp.2096-2101.

Yoon G., Kondo J., Sakai Y., Watanabe T., Iizuka S., Okumiya M. 2011. Coupling strategy of HVAC system simulation and CFD, Part 1, Study on air conditioning system using OA floor in design phase, Proceedings of Building Simulation 2011, pp.1997-2003.