

ESTIMATION OF ANNUAL ENERGY LOSS BY INDOOR AIR MIXING

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ABSTRACT

Excess energy consumption of HVAC system is not usually noticed without the dissatisfaction to thermal environment. Recently it has become easier to discover it in buildings with the aid of Building Energy Management System (BEMS). Indoor Air Mixing loss occurs in an office when the perimeter zone is heated by one HVAC system and the interior zone is cooled by the other system. The aim of this study is the quantitative analysis of the mixing energy loss in office buildings. Experiments are conducted in a full-size experiment room. The result shows that it is difficult to halt the mixing loss once the loss arises. After the reproduction of the simulation is confirmed by the comparison of results between experiment and simulation, several cases of simulation are conducted to quantify the annual energy loss in a model building. The comparison of results shows that preventing the mixing loss may reduce 9.4 % of the energy consumption of the whole building.

KEYWORDS

Indoor Air Mixing Loss, experiment, energy conservation, simulation

INTRODUCTION

Building services are often operated with hidden energy

waste that can be cut down. Excess energy consumption of HVAC system is not usually noticed without occupants' dissatisfaction with thermal environment. Recently it has become easier to discover it in buildings with the aid of Building Energy Management System (BEMS). Simultaneous energy consumption of heating and cooling in the same HVAC system is discovered by analyzing BEMS. Indoor Air Mixing loss occurs in an office when the perimeter zone is heated by one HVAC system and the interior zone is cooled by the other system. The internal heat gains such as lights and equipments increase in an office building, so interior zone need to be cooled by HVAC system even in winter. The qualitative and quantitative mechanism of mixing energy loss is clarified by experiments on steady state that indoor and outdoor temperature, and internal gains are constantly controlled (Nakahara, 1987). The aim of this study is the quantitative analysis of the mixing energy loss in office buildings. It is difficult to reproduce and quantify the mixing energy loss in office buildings. If there are two system of air conditioner in one zone, we need to evaluate the mixing energy loss of horizontal direction by experiments in order to quantify it. In this paper, experiments under unsteady state are performed in a climate chamber assumed as an office firstly, where outdoor temperature and internal heat gains are changed with daily cycle, and the features of mixing loss observed in operation of HVAC systems

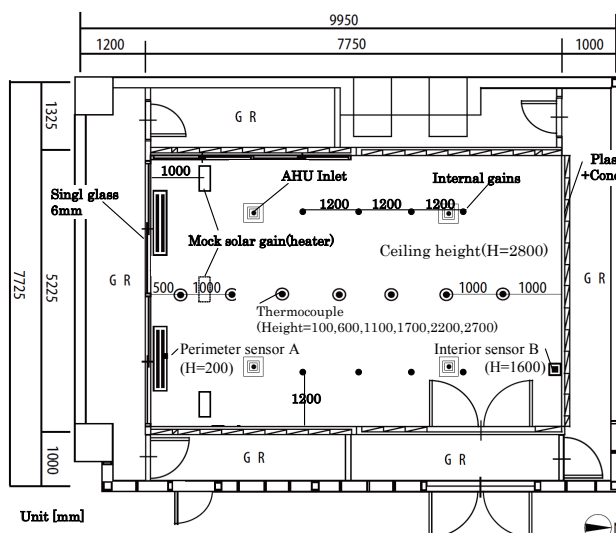


Figure 1 Plan and measurement points in the test room

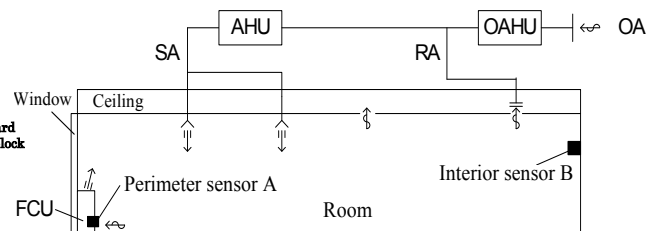


Figure 2 HVAC systems in the indoor room

Table 1 Capacity of HVAC system

Sign	Equipment	Capacity	Number
OAHU	Outdoor Air Handling Unit	Cooling Power:7.80kW Heating Power:3.94kW	1
AHU	Air Handling Unit	Cooling Power:10.4kW Heating Power:13.92kW	1
FCU	Fan Coil Unit	Cooling Power:3.69kW Heating Power:5.93kW	2

are made clear. Second, simulation results of a mathematical model for unsteady state heat analysis are compared with experimental results to confirm the reproduction of the simulation method. Finally, case studies are conducted by simulation to quantify the annual energy loss in a model building.

EXPERIMENTS

Outline of the test room

The experiments are performed with actual sized equipment (Figure 1). The chamber consists of a test room and surrounding 'guard rooms' (hereafter denoted by GRs) whose temperatures are controlled as outdoor and adjacent rooms. In GRs Fan Coil Units (hereafter denoted by FCUs) are installed to control temperature. Two FCUs are installed for the perimeter zone of the test room and Air Handling Unit (hereafter denoted by AHU) for the interior zone. There is Outdoor Air Handling Unit (hereafter denoted by OAHU) to control air temperature and humidity taken as outdoor fresh air for AHU. (Figure 2). FCUs in GRs and OAHU in the interior can simulate the outdoor condition. HVAC system is AHU that air supply on the ceiling in the interior and FCUs are established on the floor in perimeter. Table 1 shows the capacity established in the experiment room. Floor area of the test room is 40.5 m², the volume is 113.4 m³.

Measurement method

Figure 1 shows the measurement points of temperature sensors in the experiments. Air temperature is measured by Cu-Co thermocouples with tin aluminum foil to

prevent thermal radiation. Measurement points are located at the center line of the room. There are 6 locations along the line and there are 7 measurement points with different height in a location. Supply and exhaust air temperature of HVAC system is also measured by thermocouples.

Table 2 shows experimental conditions. The test room is considered as a one span of a large office room. The occurrence of mixing loss is significantly influenced by the measurement point of thermal sensors to control indoor temperature (Nakahara, 1987). In this experiment thermal sensors are established in the measurement point where mixing loss occur easily (Figure 1 and 2). The climatic condition is for heating system design in Tokyo (JABMEE, 1987) with excess probability of 20%. Internal gains are assumed as persons and lights in the office. Solar heat gains are given by three heaters operated with schedule. Solar gains are calculated by flux of solar radiation facing south in Tokyo with excess probability of 20% in winter as the shading coefficient of window is assumed to be 0.96. In order to measure thermal load of AHU and FCU, outdoor air condition is made by OAHU and is mixed with return air to AHU. Table 3 shows experimental cases. Case 1 is the condition of design value (Heating mode: 22 °C, Cooling mode: 26 °C) used as preset temperature. Case 2 is the same preset temperature (24 °C) for heating and cooling, where mixing loss will be expected to occur easily. Case 3 changes the number of FCU in order to compare the influences by the differences of instrument capacity.

Table 2 Experimental conditions

Parameter	Condition
Climatic condition	Tokyo with excess probability of 20% in heating system design
Preset temperature of non air conditioning room	South(Tokyo with excess probability of 20% in heating system design). Other: 20°C constant
HVAC system	Operation: 8:00 ~ 18:00 Air flow rate: AHU:680m ³ /h FCU: Middle mode(475m ³ /h)
Outdoor air flow rate	282m ³ /h
Internal heat gains	Persons: 12.3W/ m ² (9:00 ~ 18:00) Artificial Lighting: 18.7W/ m ² (9:00 ~ 18:00)
Solar gain	600W(6:00- 9:00, 15:00- 18:00), 1800W(9:00- 15:00) *Calculated from weather data

Table 3 Experimental cases

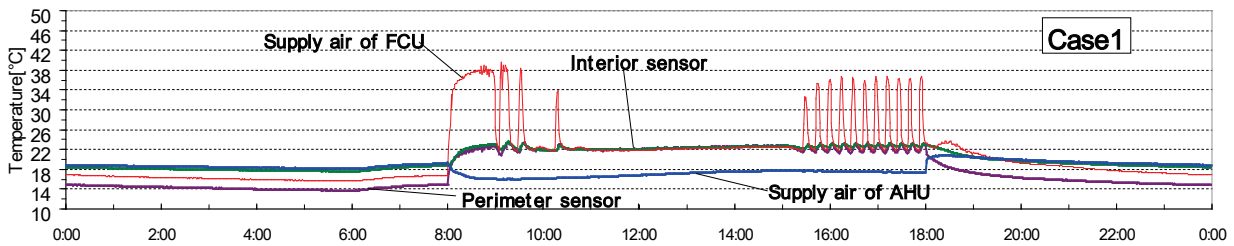
Case	1	2	3
P preset temperature	Heating mode:22°C	Heating mode:24°C	Heating mode:24°C
I preset temperature	Cooling mode:26°C	Cooling mode:24°C	Cooling mode:24°C
Air Conditioning System	AHU by ceiling diffusers FCU established floor × 2		AHU by ceiling diffusers FCU established floor × 1

* P: Perimeter, I: Interior

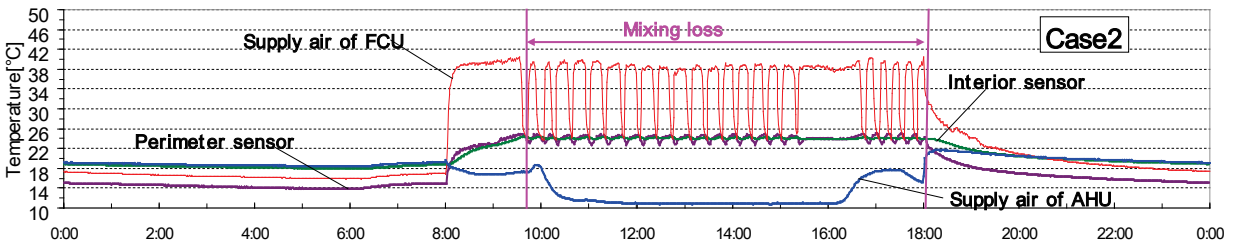
Experimental results

Figure 3 shows the temperature of the supply air of AHU, FCU and thermal sensors of interior and perimeter. Figure 4 shows thermal load for cooling, heating, and internal gains and the valve travel. Originally the mixing loss means energy loss in the room caused by air mixing between heating and cooling equipment. In this study increase of the thermal load of HVAC system caused by original mixing loss is estimated from the view point of the potential with energy conservation. Case 1: FCU for perimeter begin to operate in the heating mode at 8:00 (see Figure 3 (a)), and stop around 9:00 by the increase of internal gains (see the valve travel of FCU in Figure 4(a)). FCU begin to operate again around 15:00 because of the decrease of internal gains intermittently, and stop at 18:00. AHU for the interior keep operating in the ventilation mode without heat supply (see the valve travel of AHU in Figure 4(a)). In Case 1 mixing loss do not occur. Case 2 (the preset temperature change): FCU for the perimeter begin to operate in the heating mode at 8:00 (see Figure 3(b)), change to intermittent operation around 9:30 because the temperature of the perimeter sensor reaches

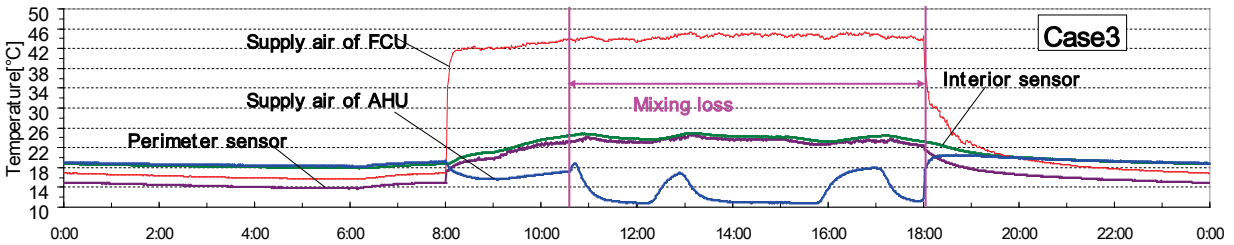
at the preset temperature of 24 °C, and stop at 18:00. AHU for the interior begin to operate in the cooling mode at 10:00 (see Figure 3(b)), stop around 16:30 because of decreasing internal gains, begin to operate again around 17:30, and stop at 18:00. AHU keep operating with maximum power to its capacity from 10:00 to 16:30. In Case 2 mixing loss is occurred from around 10:00, and mixing loss continues regardless of the outdoor conditions. Case 3 (the number of operating FCU change): Perimeter FCU keeps operating with maximum power to its capacity from 8:00 to 18:00 (see the valve travel of FCU in Figure 4(b)). Interior AHU begin to operate around 10:30, repeat ON/OFF every one or two hours, stop at 18:00. In Case 3 mixing loss also occur like Case 2, continue regardless of the outdoor conditions. Mixing loss occur in the cases where FCU begin to operate in the heating mode and indoor thermal environment reach to steady condition. The result shows that it is difficult to halt the mixing loss once the loss arose. As the feature observes in the equipment when the mixing loss occurs, both cooling and heating valves are opened and the valve of the equipment with smaller capacity was fully open.



(a) Case1

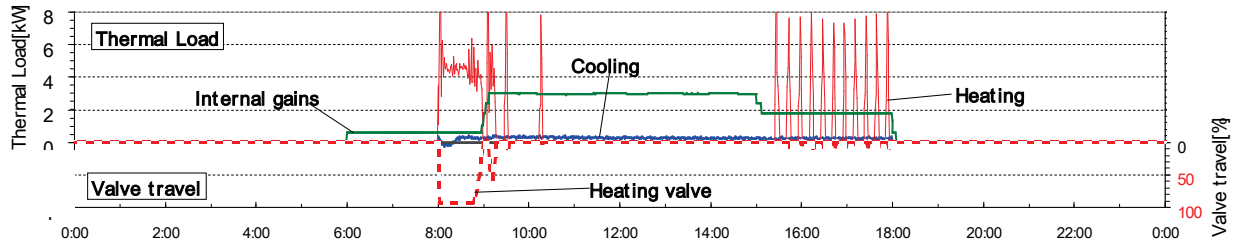


(b) Case2

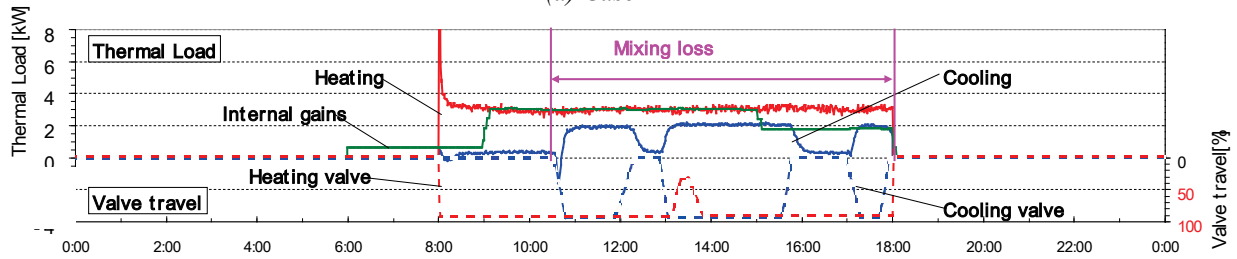


(c) Case3

Figure 3 Temperature of supply air of AHU & FCU, thermal sensor



(a) Case1



(b) Case3

Figure 4 Thermal load heat and valve travel

SIMULATIONS

Outline of simulation

In this study a simplified model for predicting vertical temperature distribution installed a coil model of HVAC system by HASP/ACSS is used (Togari, 1993). This model consists of three parts (Figure 5). The first is the “wall surface current model” for evaluating the descending (or ascending) current flowing along the vertical wall surface. The second is the “primary airstream evaluation model,” which handles the airstreams discharged from outlets as non-isothermal free jets to evaluate their influence on vertical temperature distribution. Last is the “heat transfer factor C_B ” for evaluating the heat transfer caused by the temperature difference between vertically adjacent zones. The value of the heat transfer factor, C_B $W/m^2 \text{ } ^\circ C$ is determined based on the reproducibility of the

experimental results. In order to estimate the mixing energy loss between horizontal zones, it is necessary to clarify the nature and numeric value of the heat transfer factor. The value of C_B between vertical zones decides $C_B = 2.3 \text{ } W/m^2 \text{ } ^\circ C$ ($2 \text{ } kcal/hm^2 \text{ } ^\circ C$) (Togari, 1993). According to Togari and Takemasa, 1995, the basic cause of the mixing loss is different vertical temperature distribution for both zones, which leads to the large horizontal temperature difference and then the intense horizontal mixing to reduce the temperature difference. The simulation model is constructed to reproduce the mechanism and heat transfers between horizontal zones are evaluated with natural ventilation by temperature difference. The height used to evaluate the amount of mixing loss is taken as each zone height. The model of the test room is shown in Figure 6. We divide the perimeter, interior, and ceiling plenum vertically into 15 zones. Calculation conditions are the

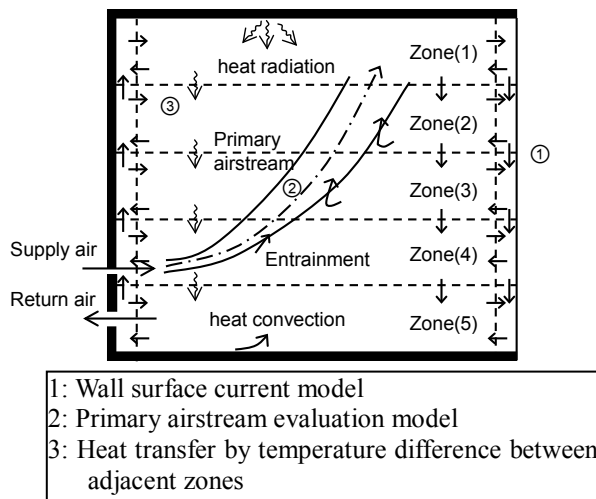
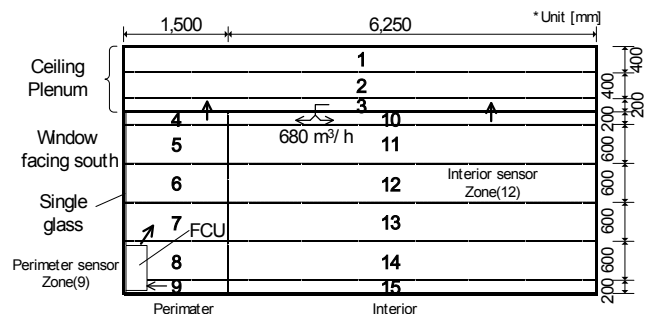


Figure 5 Outline of simplified prediction model



Internal heat gains: Lighting $18.7 W/m^2$, Persons $12.3 W/m^2$ (zone(13)(14)), Solar gain schedule in Table 2 (zone(8)(9)), Calculation interval 120sec, Outer wall: urethane foam 20mm + Concrete block 120mm + Flexible board 12mm, Window: single glass

Figure 6 Simplified prediction model

same as the experimental conditions (Table 2). The point of the perimeter sensor is zone (9), and the interior sensor is zone (12). Calculation cases are same as the experimental cases (Table 3). Lighting heat gains are given with the diving ratio of the room and the ceiling plenum 7 to 3. Lighting gains is given to zone (3), (4) and (10) as convection, and to the floor and the ceiling as radiation.

Comparison between calculated and measured values

The comparison between calculated and measured results is mainly performed in Case 3 in which mixing loss occurred. In simulation heat transfer between the zones can be adjusted by changing methods for evaluating heat transfer between horizontal zones. The comparison between calculated and measured results is performed in two cases of changing heat transfer between horizontal zones. In Case 1 the amount of heat transfer is calculated to represent heat transfer by air movement between horizontal zones due to the temperature difference. In Case 2 the heat transfer is assumed to be 100 times of that for Case 1. Calculated results of vertical temperature distribution in the perimeter and interior are shown compared with the measured results (the measured values for the perimeter is the temperature of 500mm from window, and the values of interior is averaged from 2500mm to 6500mm) in Figure 7 at 6:00 as the representative values before operating HVAC system, at 9:00 as the representative after operating HVAC system, and at 15:00 when supply air temperature of AHU and FCU become steady. In Case 1, before HVAC system operated, calculated values are slightly lower than the

measured values in the perimeter and the interior. In the perimeter the calculated values are slightly higher than the measured values when HVAC system operated. But the characteristic of vertical temperature distribution is well represented. The differences between calculated and measured values are caused by three reasons: firstly thermal capacity of air is not assumed in calculation, secondly the interior heat gains is directly given as thermal load to zone in calculation and finally the responsiveness of the thermal sensors in the experiments is ignored (see Figure 7(a)). Although the calculated values near the ceiling are slight lower than the measured values in the interior, the characteristic of vertical temperature distribution is well represented (see Figure 7(b)). The reason for the lower temperature near the ceiling in calculation is that all the cooled air from AHU is supplied to zone (10). In Case 2, in the perimeter the calculated values near the ceiling are lower than the measured values when HVAC system operated (see Figure 7(c)). The reason for the lower temperature near the ceiling in calculation is that heat transfer between interior and perimeter are promoted. This model doesn't evaluate the influence of jets from the HVAC system that transcends the boundary of horizontal zones. However, in the perimeter and interior the characteristic of vertical temperature distribution is well represented in both cases. The following study used Case 1 that is heat transfer between horizontal zones due to the temperature difference.

Figure 8 shows the comparison of the thermal load and the temperatures of thermal sensors. The calculated values are slightly higher than the measured values when HVAC system begins to operate. The differences between calculated and measured values are caused by

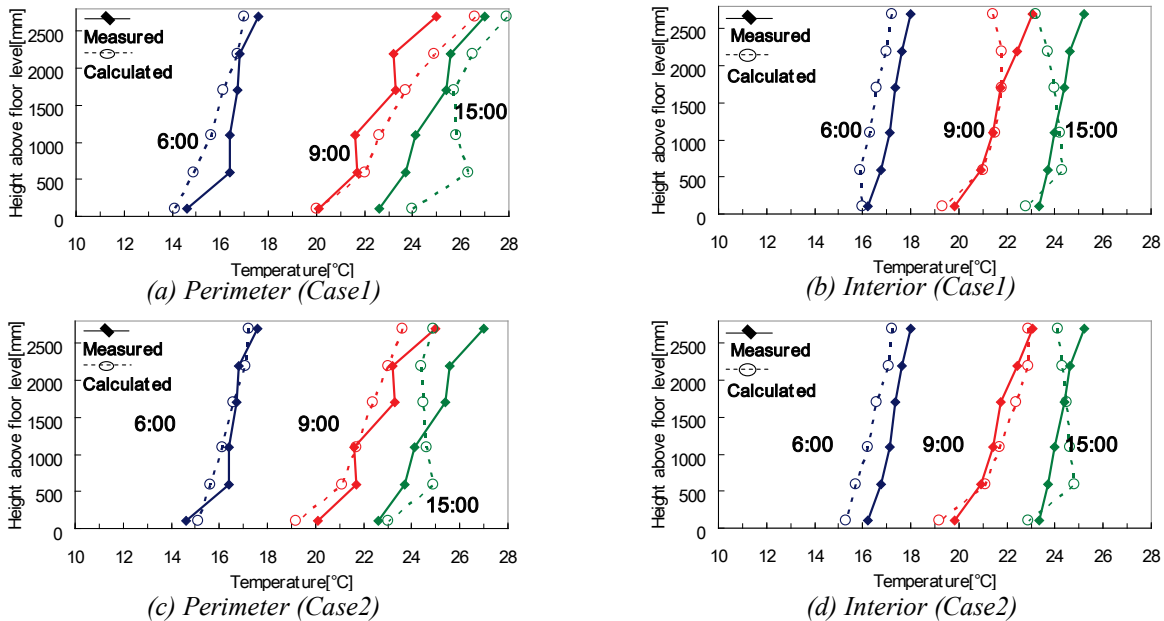


Figure 7 Comparison of vertical temperature distribution in Case 3

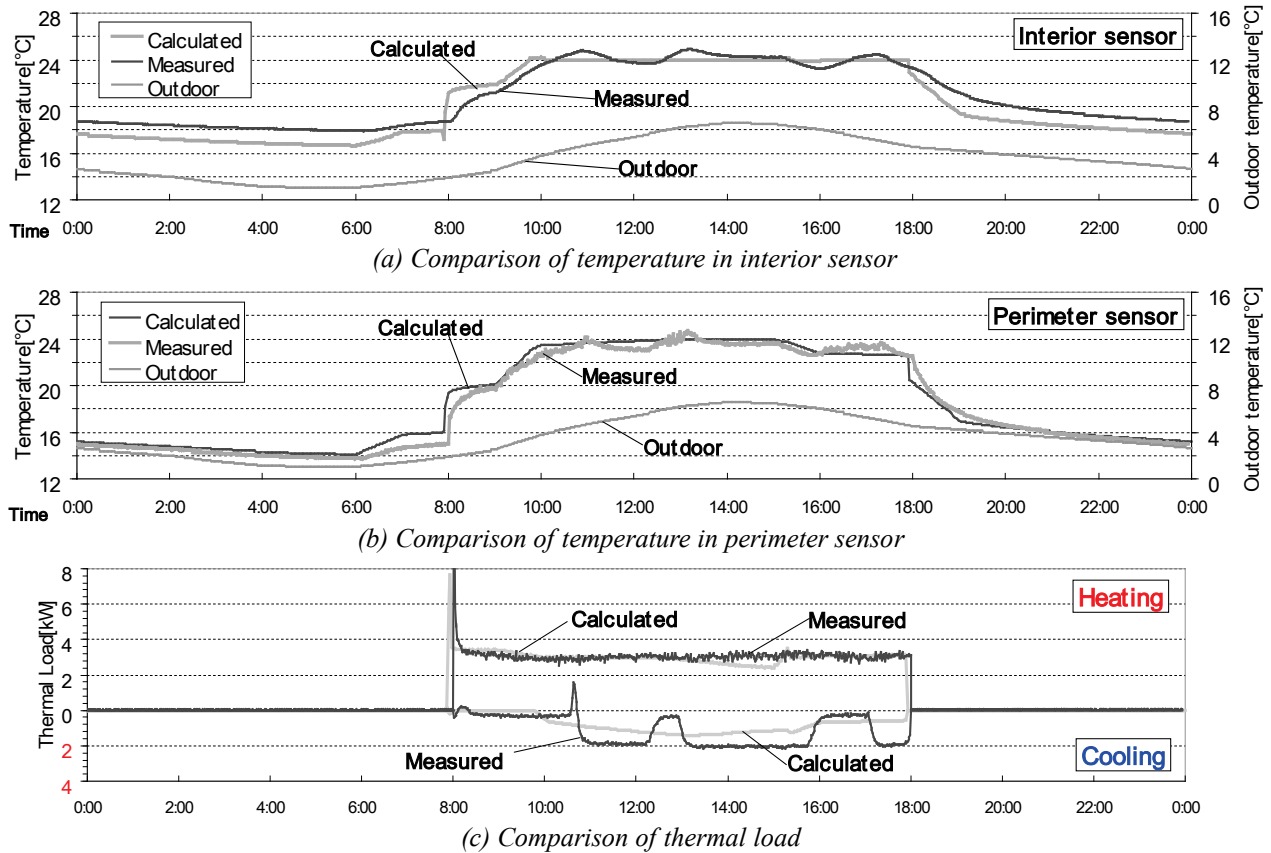


Figure 8 Comparison between calculated and measured in Case3

the same reasons former paragraph (see Figure 8(a)). The calculated temperatures coincide well with the measured. There is small difference on the heating load of FCU between the calculated and measured, but the change of the cooling load of interior AHU in simulation is different from the experiment (see Figure 8(c)) because PID control for the valve do not work well in experiment and the valve travel changes much. The period of occurrence for mixing loss is well reproduced because AHU began to operate in the cooling mode past 10:00 in experiment, around 10:00 in calculation. Figure 9 shows the comparison of the daily cumulative thermal load in each case. In Case 3 the daily cumulative cooling load coincides well between simulation and experiment within the difference of 5%. The calculated values for the heating load are about 30% higher than the measured values in Case 1, but the values for the simulation and for the experiment in other cases coincide well within the difference of 10%.

ENERGY CONSERVATION POTENTIAL WITH INDOOR AIR MIXING LOSS IN MODEL BUILDING

Outline of model building

Figure 10 shows the outline a model of building and the plan of typical floor. The total floor area is about

54000m² and used as an office. The outer walls are structured by the reinforced concrete 150mm and urethane foam 20mm. the windows are multi-layered glass and blind.

Outline of simulation

Calculation is conducted for one span in office room (see surrounding part of a dotted line in Figure 10). We divide the perimeter, interior, and roof-space vertically into 18 zones (see Figure 11). Table 4 shows calculation conditions. Internal heat gains are given referring to Togari and Takemasa 1995. Table 5 shows calculation cases. Energy consumption with different preset temperatures between perimeter and interior in the whole building is estimated for each case and the comparison is done among them.

Calculated results

Figure 12 shows the relation between outdoor

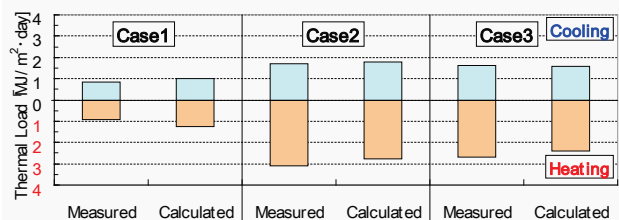
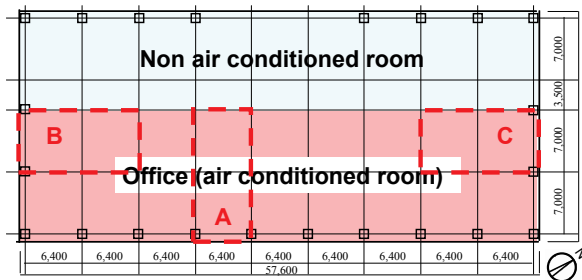
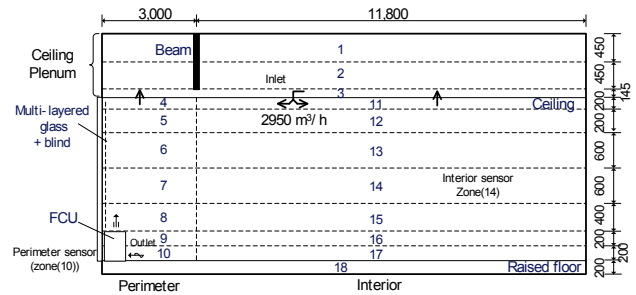


Figure 9 Comparison of daily cumulative thermal load (Cooling; AHU, Heating; FCU)



Use: office, Number of stories: lower ground floor 4 and 28-storey building, Total floor area: 54,000m², Floor height: 3.845m, Ceiling height: 2.80m

Figure 10 Outline of building and plan of typical floor



Internal heat gains: Persons and Equipments (zone(8)(9)(15)(16)), Distribution of Lighting is same as Fig 5, Calculation interval 120sec, Outer wall: RC 150mm + urethane foam 20mm, Window: multi-layered glass + blind

Figure 11 Simplified prediction model

Table 4 Calculation conditions

Parameter	Condition
Climatic condition	Reference weather year data (Tokyo)
Calculation duration	year (January ~ December) (approach duration: one month)
Temperature of non air-conditioned room	Toax: 0.15 + Tzone14 × 0.85
HVAC system	Interior: AHU, Perimeter: FCU × 2
Operation of HVAC	Weekday: 8:00-21:00, Weekend: stop, National Holiday: stop
Airflow rate of HVAC	AHU: 2950m ³ /h, FCU: 475m ³ /h
Capacity of HVAC	AHU: Cooling: 12kW, Heating: 8kW FCU: Cooling: 3kW, Heating: 3.4kW
Outdoor flow rate	570m ³ /h
Internal gains	Persons: 0.1person/m ² , Equipments: 12.8W/m ² , Lighting: 13W/m ²
Leakage	Air change rate: 0.11/h
Gains schedule	Weekday: 9:00-18:00 (100%), 18:00-21:00 (50%)
Internal furniture	assumed as flat plate: 12.6 kJ/m ³ K (Ishino, 1987)

Table 5 Calculation cases

Case	1	2	3
P preset temperature	Cooling mode: 26°C Heating mode: 22°C	Cooling mode: 26°C Heating mode: 23°C	Cooling mode: 26°C Heating mode: 24°C
I preset temperature	Cooling mode: 26°C Heating mode: 22°C	Cooling mode: 25°C Heating mode: 22°C	Cooling mode: 24°C Heating mode: 22°C

* P: Perimeter, I: Interior

temperature and the thermal load for AHU and FCU. Figure 13 shows the comparison of the annual cumulative load for HVAC system in three cases. Hourly thermal load of typical floor add thermal load of each direction (Southeast (A) by 8 zones is added to Southwest (B) by 1.5 zones and Northwest (C) by 1.5 zones in Figure 10). The values for cooling and heating in Case 1 are lower than that in Case 3 when outdoor temperature is low. In this temperature range mixing loss is occurred (Figure 12). The annual cumulative heating load for HVAC system in Case 1, in which mixing loss is prevented by preset temperature, is 41% lower than that in Case 3, which occurred mixing loss, and 9% lower than that in Case 2. Supposing that the energy consumption of air conditioning and heating equipment is 22.8% of the whole energy consumption in office building similar to Case 3 (The Building-Energy Manager's Association of Japan, 2005), the comparison of results between Case 1 and

Case 3 shows that preventing mixing loss might reduce 9.4%. To prevent the mixing energy loss, the preset temperature of heating mode in the perimeter should be lower than that of cooling mode in the interior. In case of constructing new building, we could improve thermal insulation performance of outer skin, and omit the perimeter zone.

CONCLUSIONS

The aim of this study is the quantitative analysis of the mixing energy loss in office buildings. The following method is adopted to reproduce the mixing energy loss in office buildings by experiments, analyzing the cause where mixing loss is generated, extracting items at HVAC equipment to detect mixing loss, and estimating the mixing loss of the energy consumption.

1) Mixing loss occur in the cases where FCU begin to operate in the heating mode and indoor thermal environment reach to steady condition. The result

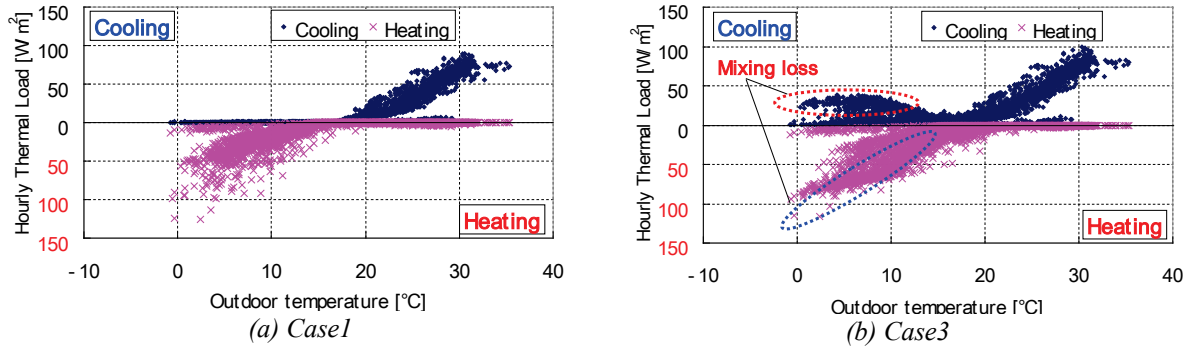


Figure 12 Relation between outdoor temperature and thermal load for HVAC system

shows that it is difficult to halt the mixing loss once the loss arises. As the feature observes in operation of HVAC systems when mixing loss occurs, both cooling and heating valves become open and the valve of the equipment with smaller capacity is fully open.

2) The reproduction of the simulation is confirmed by the comparison of the result between experiment and simulation. The differences between calculated and measured values are caused because thermal capacity of air is not assumed in calculation, because the interior heat gains is directly given as thermal load to zone in calculation and because the responsiveness of the thermal sensors in the experiments is ignored.

3) Several cases of simulation are conducted to quantify the annual energy loss in a model building. The comparison of results shows that preventing the mixing loss might reduce 9.4 % of the energy consumption of the whole building. To prevent the mixing energy loss, the preset temperature of heating mode in the perimeter should be lower than that of cooling mode in the interior. In case of constructing new building, we could improve thermal insulation performance of outer skin, and omit the perimeter zone.

ACKNOWLEDGMENTS

This study was carried out as part of the research work by New Energy and Industrial Technology Development Organization (Grant #06002572-0). For the preparation of this paper, the authors received very valuable advice from Associate Prof. Tatsuo Nagai of Tokyo University of Science and Prof. Noriyasu Sagara of Kitakyushu University.

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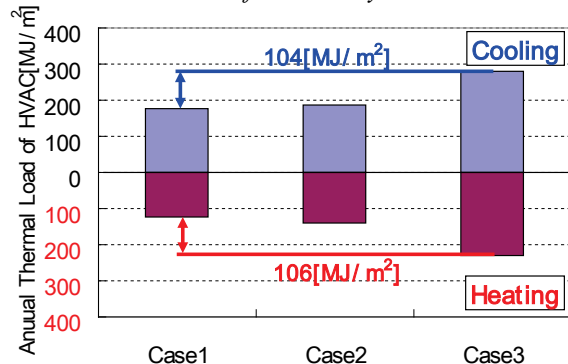


Fig 13 Comparison of annual cumulative thermal load for HVAC system in three cases

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