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TOTAL ENERGY PERFORMANCE OF THE AIR CONDITIONING SYSTEM USING OUTDOOR AIR COOLING POTENTIAL

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ABSTRACT

From the view point of designing the energy efficient air conditioning system with maximized ventilation, the possibility of an all fresh air system was discussed using the simulation results of a model building in Tokyo. The algorithm of single duct cooling system for simulating the cooling coil of air handling unit based on the heat balance model of whole system components is also described as a base for the simulation of cooling effect of ventilation fresh air. The simulation results show that the coil cooling load of the all fresh air system with an overall heat recovery system is the smallest among three Cases. The heat recovery in the midsummer contributes to decrease the increased cooling coil load of all fresh air system which is an energy efficient system in the middle season when the ambient air temperature is lower than the room set point temperature.

INTRODUCTION

For reducing the energy consumption of air conditioning system, the idea of minimized ventilation rate is widely accepted in designing the building energy systems. Whereas the minimum ventilation rate is very effective in heating season, it is less effective for cooling, as the temperature difference between the indoor and outdoor air temperatures in the cooling season is not so large in comparison with that in heating for the climatic condition of the large cities in Japan. Furthermore, there are many hours when the outdoor air temperature is lower than the design room air temperature in cooling season. Therefore, the idea of maximizing ventilation rate to minimized the cooling energy is worth evaluation using the building simulation tool.

ALL FRESH AIR SYSTEM

The simulation study for a standard model office building was described (Udagawa er al., 2002). In this study the potential of the air conditioning system with maximized ventilation rate in cooling season for the existing building in Tokyo is described in order to examine the possibility of improving the cooling system performance. The whole building energy simulation tool, EESLISM(Udagawa, 1990, Udagawa et al., 1999) based on the total heat balance model consisting of both building thermal system and mechanical heating and cooling system is used in this study. The simulation method related to this simulation case study is described as well as the simulation results.

Figure 1 shows an office cooling system used in this study. The constant air volume system(CAV) systems shown in Figure 1 were compared in this study. Case 0 shown in Figure 1a is considered as a base case with a re-circulation air path. The overall heat exchanger is provided in Cases 1 as well as Case 2. While Case 1 uses a re-circulation air path, Case 2 is an all fresh air system without re-circulation air path. It is assumed that the air temperature of office space is controlled by CAV system with a variable cooled air temperature. The cooling coil of air handling unit is controlled by a room set point temperature in the CAV system. While the cooling system can control a office space temperature, other space temperatures shown in Figure 1a can not be controlled to the set point and free floating conditions are observed.

Figure 2 shows the floor plan of a typical office floor of the model building. The above and below spaces are considered as same floor arrangement as the typical floor shown in Figure 2. Two air handling units for South zone and North zone respectively are provided. Adjacent to two office spaces two halls at the east and west sides and four isolated core spaces are assumed. The ceiling plenums are modeled as rooms which are used as the return air chambers.

COOLING COIL LOAD IN HEAT BALANCE MODEL

Cooling rates of a cooling coil is defined as Equation 1 for sensible heat. Whereas, Equation 1 shows the definition of the sensible cooling load, it can be also used to calculate the cooling coil loads combined with heat balance equation of building thermal simulation model and components of cool air distribution system. Usually, the cooling coil load is calculated as a sum of the cooling loads of the rooms supplied from the air handling unit and ventilation air cooling load of the air handling unit, the cooling coil load calculated from the heat balance models of system components can be simulated with consideration of control strategy and effects of system components on the coil load (Udagawa, 1999)

$$Qload_s = CaGc(Tc_{out} - Tc_{in}) \qquad (1)$$

In order to calculated the coil load using the heat balance model, the basic equations of each system component used in EESLISM are necessary. The components of air conditioning systems other than cooling or heating coil are room, branching and converging elements of duct system and heat exchanger for heat recovery.

The heat balance of a room air is expressed with Equation 2. Where Equation 3 shows a cooling or heating rate due to the air flow from an air conditioning system or adjacent spaces. The mathematical operation on Equation 2 combined with the equations of unsteady state heat conduction of walls, the thermal model of windows and radiation interchange between room surfaces Equation 4 can be conducted using the coefficients, $RMT_{i,j}$ and RMC_i conducted from the heat balance equations of whole building elements. Equation 4 is used as a component model of room thermal calculation model for room air temperature or heat load (Udagawa 1986, 1993).

$$RM_i \frac{dTr_i}{dt} = \sum A_{i,n} h_{c\,i,n} (Tr_i - Ts_{i,n}) + Qrmc_i + CaGo(Ta - Tr_i) + \sum Qd_{i,k} \qquad (2)$$

$$Qd_{i,k} = CaGd_{i,k} \, (Td_{i,k} - Tr_i) \qquad (3)$$

$$\sum_{j}^{Nroom} RMT_{i,j} Tr_j - CaGo(Ta - Tr_i) - \sum_{k}^{Ndi} Qd_{i,k} = RMC_i \qquad (4)$$

Branching and converging elements of duct system are important to define flow path of the conditioned air. For a converging duct, heat balance is expressed by Equation 5, while a branching duct the heat balance equation is not necessary as a system component model, since fluid temperatures at the inlet and outlets are the same for branching element.

$$\sum G_i Tin_i = \left(\sum G_i\right) Tout \qquad (5)$$

Sensible heat recovery rate at a heat exchanger used in air conditioning to heat exchange between the exhausting air and the fresh air is expressed using Equations (6a) and (6b), respectively.

$$Qhex_s = CaGex(Tex_{out} - Tex_{in}) = \varepsilon_{exs} (CaG)_{\min} (Ta - Tex_{in}) \qquad (6a)$$

$$Qhex_s = CaGo(Toa_{out} - Ta) = \varepsilon_{exs} (CaG)_{\min} (Tex_{in} - Ta) \qquad (6b)$$

Heat gain of duct system or fan can be included in the simulation model. For the calculation of heat gain of a duct element or a fan, the steady state heat balance model is used. Heat gains from the supply fan and return fan are considered in this simulation as shown in Figure 1.

MOISTURE BALANCE MODEL AND HUMIDITY SIMULATION

Room humidity in air conditioned space is necessary for the simulation of cooling system as the cooling load the space consists of both sensible and latent heat. Usually room heat load calculation as well as cooling coil calculation, the room humidity of conditioned space is assumed to be kept at a set point. However, when chilled cooling coil is used without reheat coil, it is impossible to meet the assumption of a constant room humidity. The humidity of the cooled space should be considered as a free floating value as the result of temperature control strategy of the air conditioning system.

The humidity is simulated using the moisture balance models which correspond to the sensible heat balance models of each system components. The moisture balance of a cooling coil corresponding Equation 1 is expressed by Equation 7. Whereas the outlet temperature Tc_{out} is controlled to keep the set point either CAV system, the outlet humidity can not be controlled in a chilled cooling coil. However, a simple assumption of the constant outlet relative humidity is useful for the simulation model of the dehumidifying cooling coil. The outlet relative humidity is kept at a constant value of 90-95% for the normally operated air conditioning systems. Using this assumption and linear approximation of the relationship between temperature and humidity ratio of moist air as shown in Equation 8, Equation 9 can be used for the component model of the latent heat load of a cooling coil.

$$Qload_l = rGc(Xc_{out} - Xc_{in}) \qquad (7)$$

$$Xc_{out} = f_0 + f_1 Tc_{out} \qquad (8)$$

$$Qload_l = rGc(f_0 + f_1 Tc_{out} - Xc_{in}) = rGc(Xc_{out} - Xc_{in}) \qquad (9)$$

As far as moisture absorption at a room surface is not considered, the moisture balance of room air is expressed with Equation 10. Using the finite difference approximation, Equation 12 can be used as the calculation model to be combined with total system simulation model. The coefficients, RMX_i and $RMXC_i$ can be conducted from the finite difference approximation of Equation 10. Equation 11 corresponds to Equation 3.

$$RMW_i \frac{dXr_i}{dt} = Wrmc_i + Go(Xa - Xr_i) + \sum Wd_{i,k} \qquad (10)$$

$$Wd_{i,k} = Gd_{i,k}\,(Xd_{i,k} - Xr_i) \qquad (11)$$

$$RMX_i Xr_i - Go_i (Xo - Xr_i) - \sum_{k}^{Ndi} Wd_{i,k} = RMXC_i \qquad (12)$$

(j=1,2,..,*Nroom*)

While for the converging duct Equation 13 corresponds to Equation 5, the calculation models for duct and fan are not necessary as the outlet humidity ratio is same as the inlet humidity ratio.

$$\sum G_i Xin_i = \left(\sum G_i\right) Xout \qquad (13)$$

For the overall heat exchanger used to heat recovery both sensible and latent heat, Equations 14a for an exhaust side and 14b for a fresh air intake side are used for the simulation model. Equations 14a and 14b are based on enthalpy, they are used in the simulation algorithm to express the heat balance of a overall heat exchanger using temperatures and humidity ratios instead of enthalpy as shown in APPENDIX.

$$\begin{aligned} Qhex_t &= Gex(hex_{out} - hex_{in}) \\ &= \varepsilon_{ext} (G)_{\min} (ha - hex_{in}) \end{aligned} \qquad (14a)$$

$$\begin{aligned} Qhex_t &= Goa(hoa_{out} - ha) \\ &= \varepsilon_{ext} (G)_{\min} (hex_{in} - ha) \end{aligned} \qquad (14b)$$

VENTILATION RATE AND COOLING COIL LOAD

The simulation was carried out for the model building from April to October when considerable amount of room cooling load is expected. The internal heat generations caused by lighting equipment and office equipment are assumed as shown in Figure 2. The minimum ventilation rate of 20m^3/hour/person is assumed for the Cases 0 and 1 for all seasons. The weather data of Tokyo based on AMeDas weather data set were used.

The cooling operation of the air conditioning system is assumed daily from 9:00 to 21:00 except on Sunday. The room set point for the office room is 26 C in the summer season from July 1 to September 15 and 24 C in the middle season from April 1 to November 30 except for the summer season. Other space temperature and all the humidity are free floating even in the cooling time.

The ventilation rates for three Cases in Table 1 corresponding to Figure 1 were compared. As the office space is divided into two zones provided with each air conditioning system shown in Figure 1, the simulation model of two sets of air conditioning systems were combined with building thermal model in the simulation.

All the system components shown in Figure 1 are described with Equations 1 to 14. For Case 0 the system variables which are shown in Figure 1a as the nodes for the air temperatures and humidities are basically the unknown values. Assuming the CAV system with a set point of the room temperature of the office room, all other system variables are the unknown values to be calculated by solving a set of simultaneous equations consisting to the system components in Figure 1. The flow rates of air at each path of system should be determined at constant rates or scheduled rates. During the free floating hours without air conditioning at night time or on holidays, all room air temperatures and humidities are unknown values, while all other system components related to the air conditioning system are extracted from the simultaneous equations describing the whole system. This process is repeated at each time step during the simulation period.

The simulation results are shown in Figures 3 to 6. Figure 3 shows the daily total cooling load of two office spaces calculated defined by Equations 1 and 7. The room heat loads for three Cases are the same as room heat load is not influenced by the ventilation rate in air conditioning system. The room heat load is always cooling from April to November due to internal heat generation from office spaces , while the coil loads are different depending on the ventilation rate or the heat recovery. As shown in Figure 6 there are only small difference between the cooling coil loads of the North and South zones, the cooling coil loads are discussed for the total of two zones.

Figure 4 show the examples of hour by hour results in a hot typical week in July . While the temperature is kept at a constant of 26 C during cooling period, the relative humidity varies around 40-50%　The results of a typical week in the middle season are shown in Figure 5. Whereas on October 15 and 16 the room temperature as well as humidity vary as the outdoor air temperature is lower.

The seasonal heat loads of the cooling coils are compared as shown in Figure 6. Figure 6 shows that the Case 2 using the all fresh air system shows the smallest seasonal coil cooling load because of the cooling effect of ventilation air in the middle season. Cases 0 and 1 show the similar results, whereas the total cooling load of Case 0 is slightly greater than that of Case1. The effect of the total heat recovery system is not so large for the seasonal cooling load.

CONCLUSION

Algorithm of a single duct cooling system for simulating the cooling coil load is described as a base for the simulation of cooling effect of ventilation fresh air. The algorithm is useful to simulate the cooling coil load considering system control strategy and total thermal performance of building thermal model. In addition, the presented algorithm can be used to simulate the room thermal environment considering free floating behavior of the room humidity of air conditioned space with single duct cooling system
.

The simulation results of the model building in Tokyo shows that the energy requirement for the cooling shows the smallest seasonal value when the all fresh air system is used combined with an overall heat recovery system among three Cases. The heat recovery in the midsummer summer season is effective to reduce the increased cooling coil load by the increased ventilation rate compared to the minimum ventilation rate.

This result also suggests that the importance of dehumidification in the cooling system with large ventilation rate, while the decreasing of the sensible cooling load is expected. Whereas the combination of all fresh air system in the middle season and the minimum ventilation rate in the midsummer season may be the most suitable for the seasonal cooling load reduction, the all fresh air system has the advantage considering the simple control strategy since the return path and the switching control is not necessary and better indoor air quality throughout entire cooling season.

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NOMENCLATURE

A: surface area, m^2
Ca: specific heat of air, J/kgK
Cs: specific heat of moist air, J/kgK
Cv: specific heat of water vapor, J/kgK
G_i: flow rate of converging flow path *i*, J/kgK
Gc: cooling coil flow rate of air, kg/s
Go: flow rate of infiltration air, kg/s
Gd: flow rate of diffuse air or inlet air, kg/s
Goa: flow rate of fresh air, kg/s
Gex: flow rate of exhaust air, kg/s
hc: convective heat transfer coefficient at room surface, W/K
hex,hoa: specific enthalpy of exhaust air and fresh air of overall heat exchanger, respectively, J/kg
Qd: heat supply rate by the air flow into a room, W
Qrmc: convective heat from internal heat generation sources, W
Qload: heat load, W
Qhex: rate of heat exchange, W
RM: heat capacity of room air, J/K
RMW: moisture capacity of room air, kg/(kg/kg)
t: time, s
Tr; room air temperature, C
Ts: room surface temperature, C
Ta: ambient air temperature, C
Td: diffuse air or inlet air temperature to a room, C
Tex, Toa: temperatures of exhaust air and fresh air of overall heat exchanger, respectively, C
Tc: coil air temperature, C
Tin, Tout: air temperatures of converging duct, C
Xr: room air humidity ratio, kg/kg
Xa: ambient air humidity ration, kg/kg
Xd: diffuse air or inlet air humidity ratio to a room,
Xex, Xoa: humidity ratios of exhaust air and fresh air of overall heat exchanger, respectively, kg/kg
Xc: coil air humidity ratio, kg/kg
Xin, Xout: air humidity ratio of converging duct, kg/kg
r : latent heat of vaporization of water, J/kg
Wd: moisture supply rate by the air flow into a room, kg/s
Wrmc: moisture from internal heat generation sources, kg/s
ε_{exs} , ε_{ext} :effectiveness of overall heat exchanger for sensible heat and total heat, respectively, -
$(\ )_{min}$: smaller value of two flow rates of a heat exchanger

Subscripts

i or *j*: room or flow path connected to converging duct
n: surface of room
k: air flow path of room *i*
in: inlet air
out: outlet air
s: sensible heat
l: latent heat
t: overall heat

APPENDIX

Simulation model of over all heat exchanger:

Using the definition of specific enthalpy shown in Equation A1, Equations 14a and 14b can be expressed with air temperature and humidity ratio instead of enthalpy. As the variation of specific heat of moist air *Cs* is small, *Cs* can be estimated using a humidity ratio calculated at the previous time step.

$$h = CsT + rX = (Ca + CvX)T + rX \qquad \text{(A1)}$$

$$\begin{aligned} Qhex_t &= Gex(CsTex_{out} + rXex_{out} - CsTex_{in} - rXex_{in}) \\ &= \varepsilon_{ext}(G)_{\min}(CsTa + rXa - CsTex_{in} - rXex_{in}) \end{aligned} \qquad \text{(A2)}$$

$$\begin{aligned} Qhex_t &= Goa(CsToa_{out} + rXoa_{out} - CsTa - rXa) \\ &= \varepsilon_{ext}(G)_{\min}(CsTex_{in} + rXex_{in} - CsTa - rXa) \end{aligned} \qquad \text{(A3)}$$

Together with Equations A2 and A3, sensible heat balance equations for corresponding to the exhaust side and the fresh air side are necessary.

$$\begin{aligned} Qhex_s &= CaGex(Tex_{out} - Tex_{in}) \\ &= \varepsilon_{exs}(G)_{\min}(Ta - Tex_{in}) \end{aligned} \qquad \text{(A4)}$$

$$\begin{aligned} Qhex_s &= CaGoa(Toa_{out} - Ta) \\ &= \varepsilon_{ext}(CaG)_{\min}(Tex_{in\,in} - Ta) \end{aligned} \qquad \text{(A5)}$$

Combining Equations A2 to A5 with the heat and mass balance equations of all other system components and solving the simultaneous equations describing the whole system, the leaving conditions, Tex_{out}, Toa_{out}, Xex_{out} and Xoa_{out} can be obtained

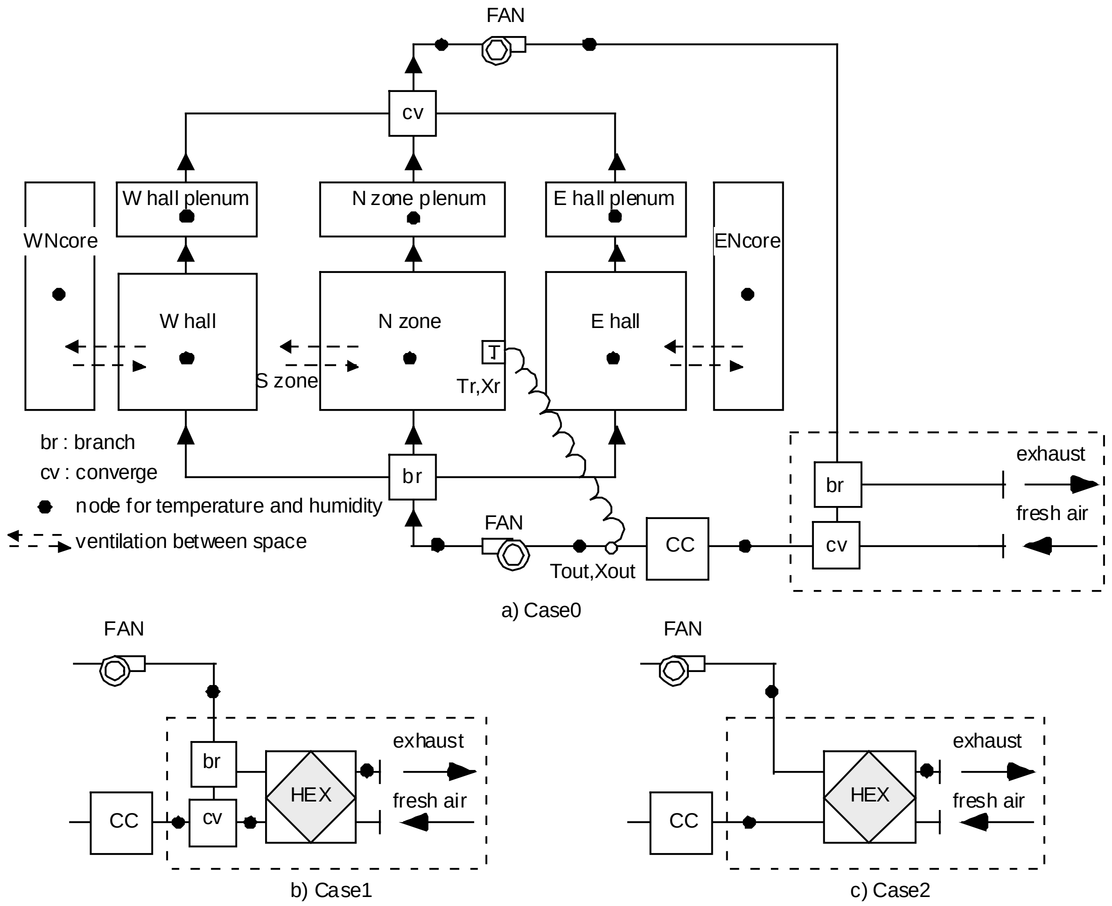
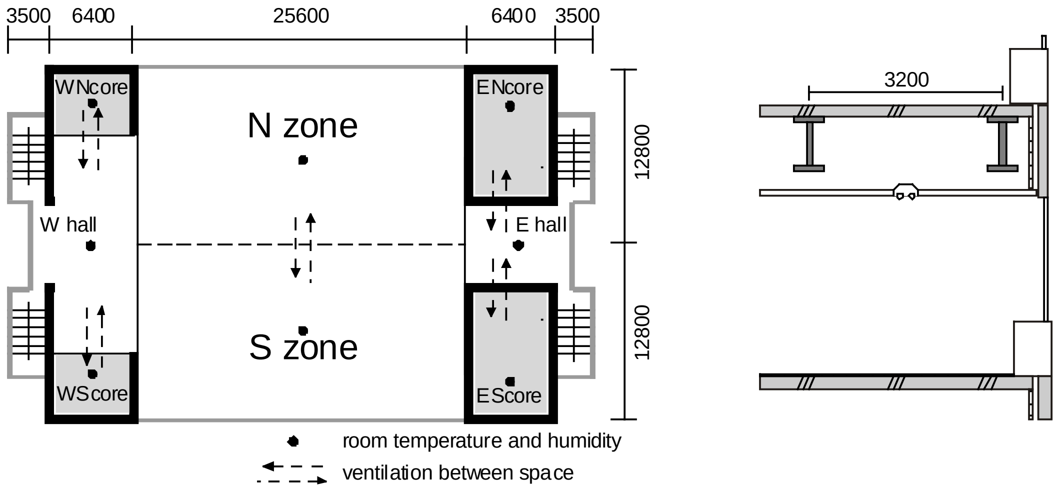


Figure 1 System diagrams of simulation Cases.

Table 1
Air flow rates the air conditioning system for each zone

	Coil air flow rate	Fresh air rate	Heat recovery
Case 0	4600m^3/h (1.27m^3/s)	1150m^3/h (0.32m^3/s)	no
Case 1	4600m^3/h (1.27m^3/s)	1150m^3/h (0.32m^3/s)	yes (July 1 – Sept. 15)
Case 2	4600m^3/h (1.27m^3/s)	4600m^3/h (1.27m^3/s)	yes (July 1 – Sept. 15)

* Above flow rates for each zone(South and North zone, respectively)
** 10 % of the coil flow rate is supplied to the east and west halls.



Office floor area/floor: 655m^2 (25.6mx25.6m)
Internal heat generation:
Occupants: 115 persons(5m^2/person), 20.9W/m^2
Lighting: 16W/m^2 Appliances: 5.8W/m^2
Minimum ventilation rate
20m3/h person(5.5L/s person) x 115=2300m^3/h

Figure 2 Floor plan and cross section of model building.

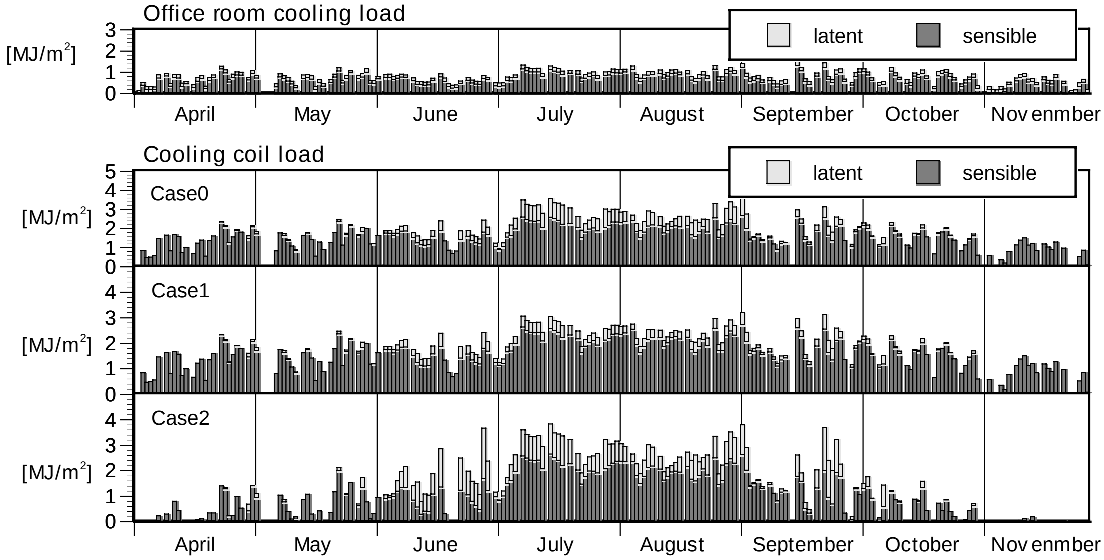


Figure 3 Seasonal variation of daily coil heat loads for total floor area.

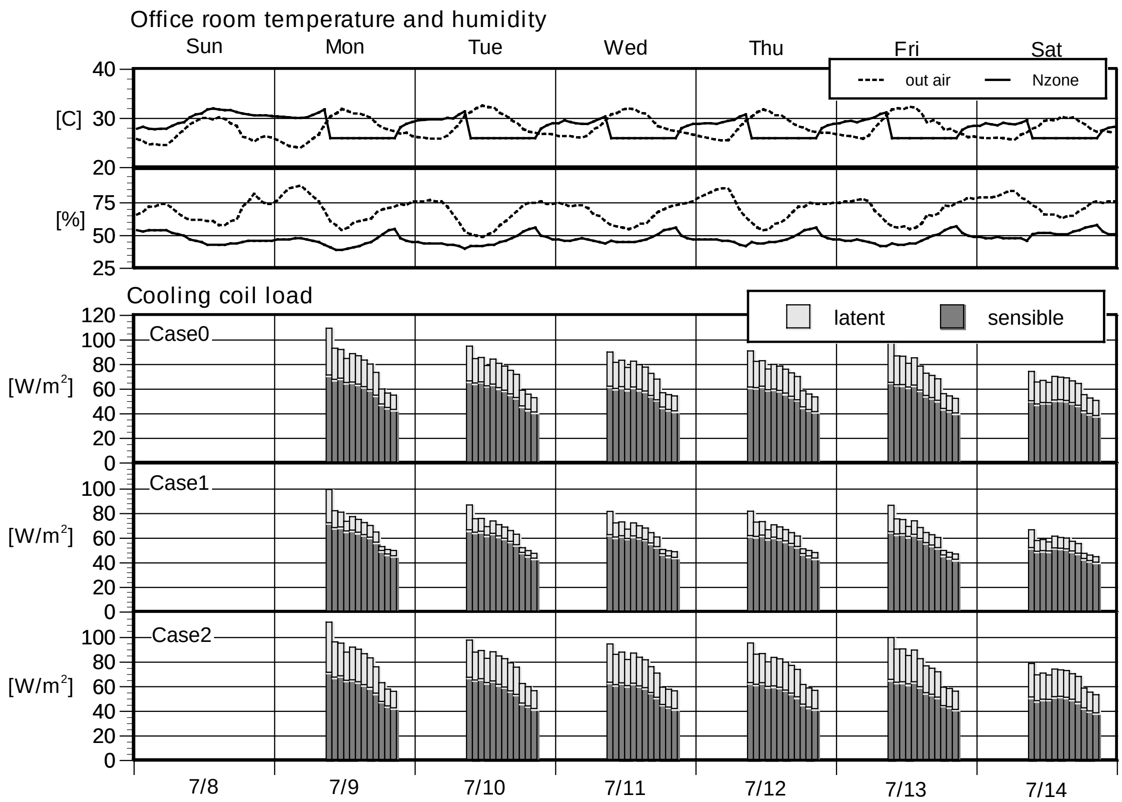


Figure 4 Daily variation of hourly cooling coil loads and room thermal environment in July

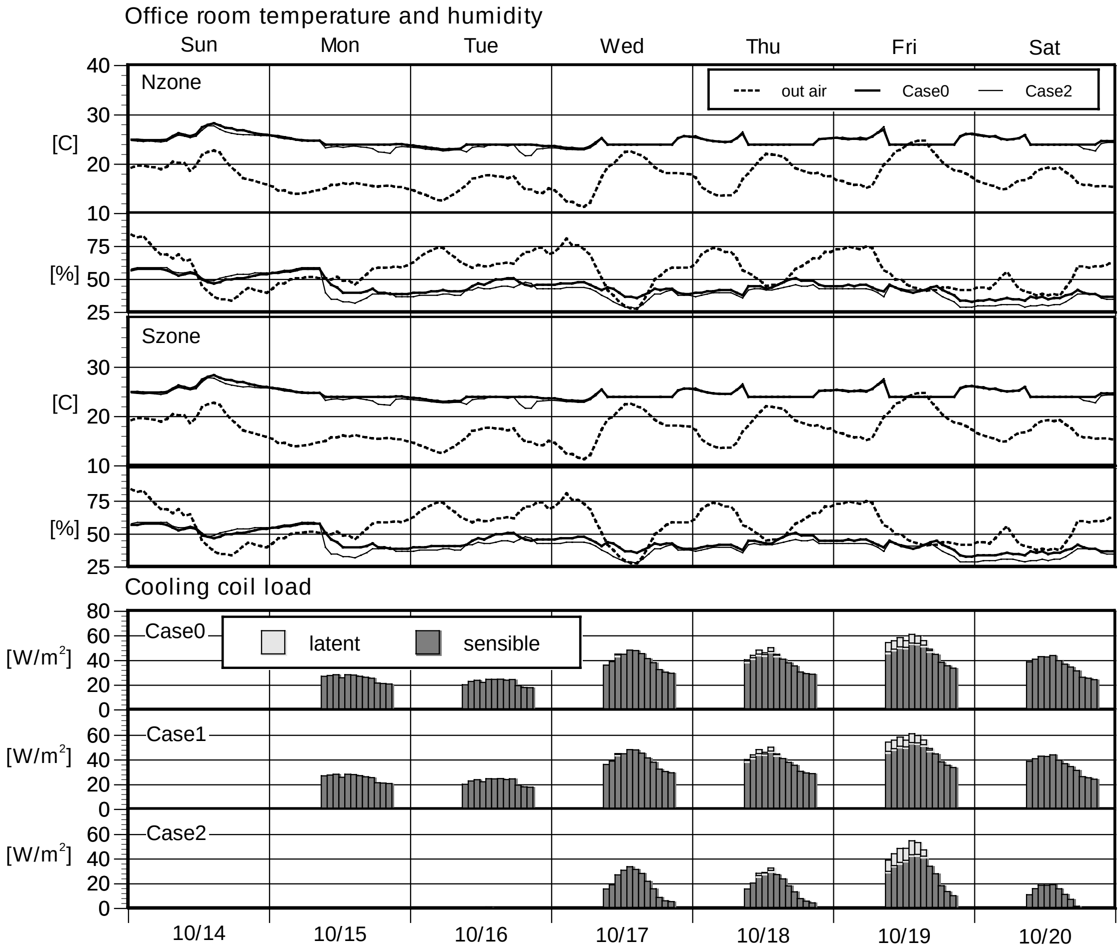


Figure 5 Daily variation of hourly cooling coil loads and room thermal environment in October

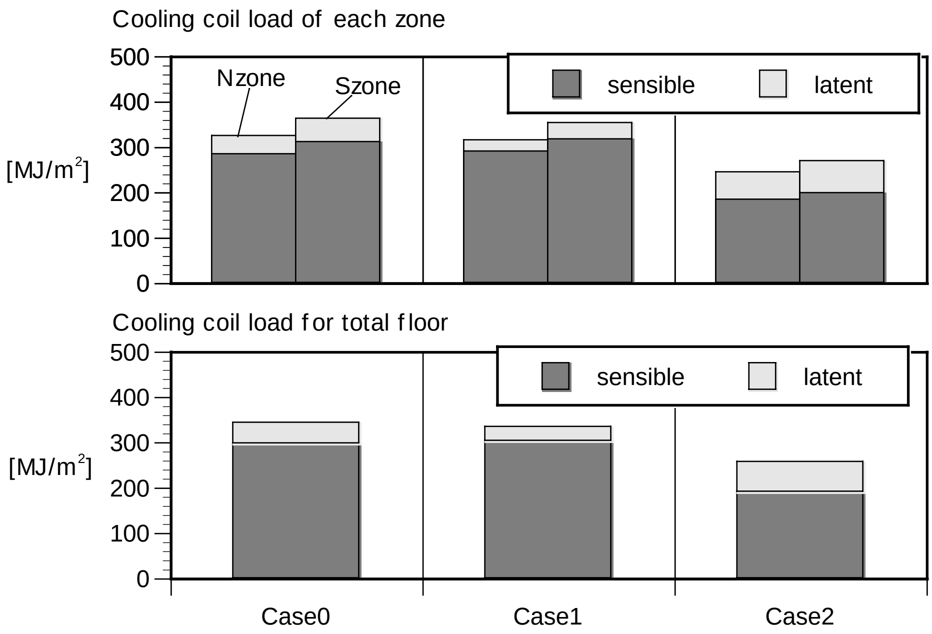


Figure 6 Comparison of seasonal cooling loads of air conditioned floor area(April - November)