DESIGN WITH HEAT LOAD SIMULATION OF COMBINED PASSIVE-ACTIVE SYSTEMS

Mitsuhiro Udagawa[1], Hyunwoo Roh[2]
Department of Architecture, Kogakuin University, Tokyo, Japan
[1]udagawa@cc.kogakuin.ac.jp
[2]nohenuga@hotmail.com

ABSTRACT

Application of EESLISM for the energy and environmental control system design in the example building is described. Examples of the combined systems of building thermal performance and mechanical heating and cooling are the solar heating system, the heat storage type air conditioning system, the ventilation cooling systems, etc. The simulation models for a single family house and an office building are described. In addition, as an example simulation, the result of the fabric heat storage air conditioning system is described from the viewpoint of simulation algorithm and the application of the simulation results for optimum of total system design.

INTRODUCTION

Combination of thermal performance of building envelope and structure with mechanical heating and cooling systems become one of the commonly recognized useful ways for efficient environmental system in buildings. Examples of the combined system are solar thermal system, double facade glazing system, radiant heating/cooling system and ventilation cooling system (Udagawa and Roh, 2002; Udagawa and Nagata, 2003; Roh and Udagawa, 2004). Various types of heat storage system using heat storage tanks or air conditioning system using heat capacity of building fabric are also examples of the combined systems.

For designing such combined passive-active system, the heat loads of heating and/or cooling by the active or the mechanical system are necessary to examine the capacities and annual or seasonal heat loads to find the optimum selection of heating and cooling equipment and estimating the expected energy consumption. Designing the system based on the average heat load instead of the peak load has the possibility of energy efficient system. Because, when the capacity is determined based on the peak load, the heating and cooling equipment may be operated in the part load condition with lower efficiency in most period of a year. In such case, the heat load simulation considering the system scheme is effective to design the systems.

HEAT LOAD SIMULATION TOOL

The major simulation tools applicable for heat load simulation have been developed and used since 1960s (e.g. ASHRAE, 2001). DOE, BLAST, TRNSYS, ESP and HASPs are the examples of them. Recently, EnergyPlus (Crawley, 1999) has been developed and the extending works for HASP/ACLD have been started (e.g. SHASEJ, 2005). The Author has been developed EESLISM as an energy simulation tool since 1990, using C on UNIX and based on the overall heat balance room thermal model (e.g. Udagawa, 1986 and Udagawa, 1993). The algorithm is based on the whole building heat balance model with combination of room heat load calculation model and simulation model of mechanical heating and cooling systems. While a time step of one hour is usually used, simulation with a smaller time step is possible if necessary. One minute is a minimum time step.

For the simulation of the combined system, the use of EESLISM is suitable. Since EESLISM can be used for whole system simulation consisting of room heat load and heating and cooling systems including domestic hot water heating, even if the specifications of heating and cooling equipment have not been determined (Udagawa and Sato, 1999). Figure 1 shows the general idea of EESLISM as a generalized heat load simulation tool. While EESLISM uses the heat balance model of whole system components including building thermal model, the basic algorithm is flexible and simple. The system to be simulated is defined with the input data of each system components and the input data of system flow paths which describe the relationship among the system components, so that various systems based on innovative idea can be simulated. Control strategy should be also input for defining the controlled object and the sensor for simulating the heat load.

The heat loads components shown in Figure 1 are the imaginary system components which are defined by the heat load with fluid flow rate and input and output status such as the fluid temperatures. When the cooling coil is assumed in AC load or Coil load as shown in Figure 1, the constant relative humidity model of the outlet air is used (Udagawa and Nagata, 2003). When the system components without actively controlled feature such as heat exchangers, heat storage tanks are designed to use, the specification of each system components should be determined prior to the simulation, since the input and output status are determined as the heat balance of the whole system. Branch elements (Br) and converging elements (Cv) represent the flow paths within the system, but branch elements do not participate in the heat and moisture balances as branching elements only pass the input status to the lower components without any change. A set of simultaneous equations consisting of the heat balance equations of system components is set up and solved for the temperatures and/or humidity ratios at the outlets of each component and the heat loads of the heating and/or cooling load components in each time step.

EESLISM can be used various stages in the design process. The room heat load without supposing specific heating or cooling system can be simulated when the only building thermal data, room thermal environment set points and weather data are given. The room heat load is very useful as a most basic data of designing building thermal energy system. Room temperature variation in free floating condition is simulated when running EESLISM without the set point for room thermal environment.

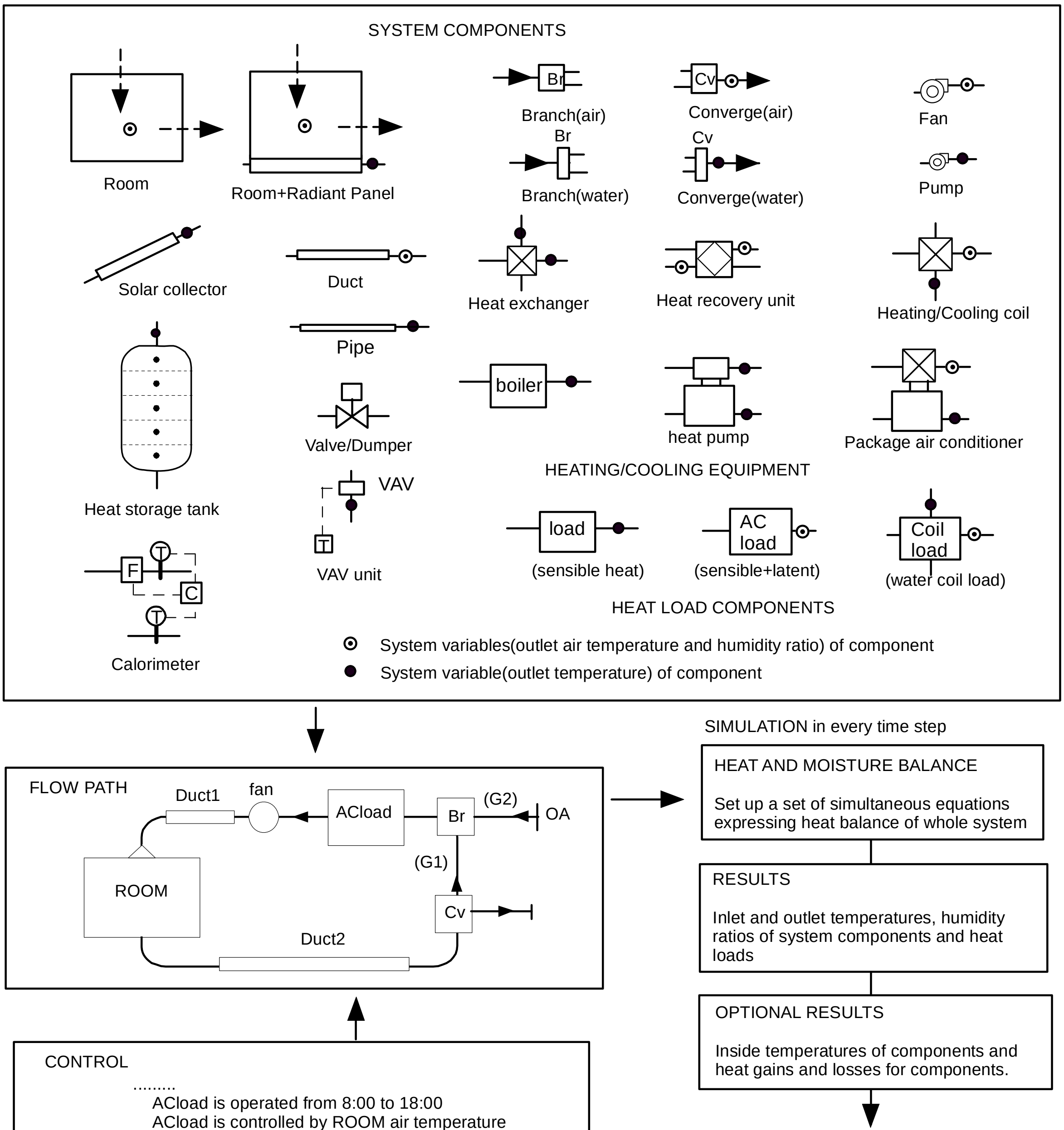


Figure 1 Simulation process in EESLISM

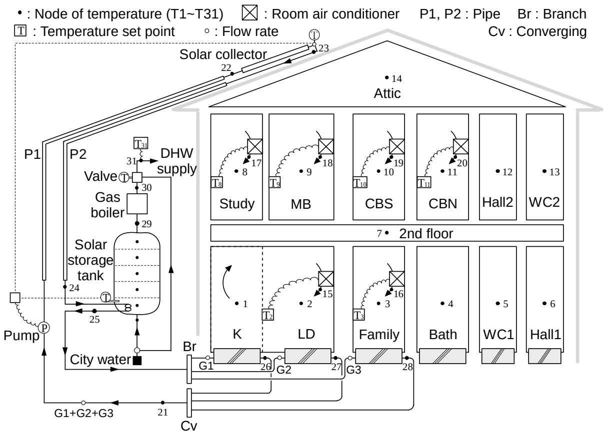


Figure 2 Simulation model for single solar family house

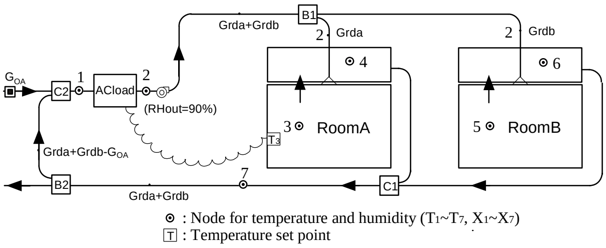


Figure 3 Simulation model for CAV system

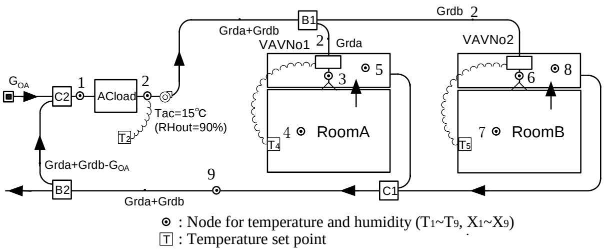


Figure 4 Simulation model for VAV system

After the system schematic was determined, the heat loads of heating cooling equipment, such as heating or cooling loads of air handling units and heating or cooling plants considering the schematic diagram can be simulated. The heat load caused by the required ventilation rate can be considered depending on the designed schematic and control strategy. The heat recovery system using heat exchanger can be included the simulation at this stage.

The running period may be heating or cooling season or one year or more depending on the system to be simulated. The statistical evaluation of the simulation results can be used for designing the capacity of the heating and cooling energy system.

EXAMPLES OF SIMULATION MODEL

Solar single family house

Figures 2 shows an example of space heating combined with solar heating system for a single family house. The solar thermal system is designed to reduce the heat loads for domestic hot water heating and space heating. A hot water heating boiler is used for the auxiliary domestic hot water heating. Room air conditioners are assumed for the space heating of the six rooms. The solar energy is used to heat the concrete floors of the kitchen, living room and family room on the first floor. The effect of solar energy is expected to reduce the space heating load due to raised floor surface temperature. The heating and cooling system shown in Figure 2 is a suitable example of the combined passive and active systems as the building thermal system using massive concrete floors are expected as heat storage component as well as radiant heating panel heated by the solar energy incident of the solar collector as a free floating heat source. The heat load of DHW boiler and heat pump room air conditioners are affected by the solar collector system. Therefore, whole system simulation consisting of both building thermal system and mechanical control system including solar collector is useful to examine the heat capacity of the DHW boiler and the room air conditioners. Annual purchased energy by the boiler and air conditioners as well as saved energy by the solar collector are also estimated as the results of the simulation.

According to the pre-determined schedule, switching the flow paths and the air and water flow rates of all the paths in Figure 2 should be given before the simulation process in every time step. The number of system components of Figure 2 is 31. The 31 temperature nodes called system variables correspond to the system components as shown in Figure 2. Basically, the node of the air represents two variables, an air temperature and a humidity ratio. However, the humidity can be neglected if latent heat load and humidity variation of room air are not necessary in the simulation as assumed in Figure 2. When the DHW is assumed to be used and space heating for the rooms where the air conditioners are scheduled to be provided, the DHW tap temperature, T31 and the room set point temperatures for LD, Family, MB, CBS, CBN and Study which are T2, T3, T8, T9, T10 and T11, respectively should be given. The heat loads of the DHW boiler controlled by T31 and the six sensible heat loads controlled by each of the room temperatures of K, LD, Family, MB, CBS, CBN and Study are unknown values to be determined by the simulation. As the total number of the components and the unknowns consisting of the seven heat loads and the 15 temperatures of uncontrolled components are the same, the simultaneous equations of the whole system thermal balance can be solved and all the heat loads and room temperatures are determined.

Office building

Figures 3 and 4 show the example air conditioning systems for typical office rooms. While four spaces are assumed in each example, two spaces are office room and the other two spaces above the office ceiling are the plenums as the plenum can be considered as a space in the multi room model used in EESLISM. More spaces can be simulated as shown in the single family case and also in the simulated examples in the later section. However, for explaining the algorithm for simulating heat load of the office building case rather simple examples are shown. The cooling system with an air conditioning unit for two office rooms is assumed in each case. The ceiling plenum is considered as a return air space where the return air from the office space is flow into each plenum. The radiant cooling effect caused by the ceiling temperature with the return air in the plenum can be considered in the simulation. The simulation also takes into account the temperature variation of plenum air before returning to the air conditioner.

Two major control ways, constant air volume (CAV) system and variable air volume system (VAV) are often used for the cooling system. The difference of room thermal environment and latent heat load in the two systems can be examined with the idea of heat load simulation of air conditioning unit considering the control strategy.

CAV system

Figure 3 shows an example of CAV system for two rooms. There are seven system components in Figure 3 with a heat balance equation and a moisture balance equation for each component. Cooling system invariably perform a dehumidification function as well as extracting sensible heat from the air. In this example the air temperature of Room A, T3 is selected for the set point to control the cooling coil. Therefore, the air temperature of Room B, T5 is free floating value even the room is considered as an air conditioned space. However, the room cooling load of Room B can be calculated as well as the room cooling load of room A. The cooling coil load is calculated simultaneously including cooling load for ventilation air. The thermal effects of the ceiling plenums are also included. The heat gain from the fan and duct system can be included if necessary, while they are ignored in Figure 3.

The system component model for cooling coil load is simply expressed as the definition of the cooling load using the temperature difference at the inlet and outlet of the coil. For the cooling load, the difference of humidity ratio at the inlet and outlet is also necessary. According to the CAV system control scheme, the coil outlet air temperature controlled by the air temperature of Room A. On the other hand, the coil outlet air relative humidity is constant as the constant relative humidity is assumed in the model. Therefore, all the humidity ratios in Figure 3 are free floating values including office rooms, Rooms A and B. The room temperature of Room B is also free floating. Such characteristic of room thermal environment in the CAV system can be simulated.

A set of the heat balance equation consisting of 14 equations of seven system components expressed with a couple of heat and moisture balance equations describes the whole system to be simulated. The number of unknown values are 14 which are six temperatures, T1, T2, T4, T5, T6, T7 and seven humidity ratios, w1 to w7, in addition the heat load of air conditioning unit, ACload.

VAV system

Figure 4 shows an example of VAV system. Even though conditioned air is supplied from an air conditioning unit, the VAV system can control the room air temperature of each room to the set point when the VAV unit is installed for each room. As the air flow rate supplied to the controlled space is determined to maintain at the set point temperature, the flow rate can not be determined before the simulation process at every time step in this case. Therefore, the algorithm for VAV system tends to be complicated comparing to the algorithm for CAV system. Whereas the algorithm in EESLISM for the VAV system is based on the algorithm for CAV system, the step to determine the supply airflow rate is necessary.

Step 1. Assuming CAV system with a scheduled constant air supply rate and a reheat coil instead of a VAV unit calculate the room heat load. A set point temperature of the supplied air from an air conditioning unit must be given to run this step.

Step 2. An airflow rate is determined from the room heat load simulated in Step 1. Once the airflow rate through each VAV unit is determined, all the flow rates can be determined. Then, the simulation process is same as that for CAV system with the air flow rate obtained by this process.

The simulation process is repeated mainly for confirming the latent heat load and the humidity values which are affected by the air flow rate.

SIMULATION OF AIR CONDITIONING WITH FABRIC HEAT STORAGE

Fabric heat storage system

Using thermal mass of building structure for heat storage operation of the cooling system is one of the possible ways to reduce the peak load without complicated equipment. This type of system is combined with the midnight power service to reduce the peak power demand and the energy cost by heat storing operation.

The cooling system with packaged air conditioners installed in the ceiling plenum above office space is widely used in Japan, therefore, it becomes a commonly recognized idea to use such type of cooling system for heat storage operation using concrete floor slab during midnight as well as usual cooling operation during office hours. It is expected that the cooled floor slab contribute to reduce the cooling load during daytime.

Simulation model

Using the weather condition of Tokyo, the simulation was carried out for an office building with reinforced concrete structure as shown in Figure 5. The typical floor spaces consisting of two east office zones (EN and ES), two west office zones (WN and WS) and a core zone on the standard floor are modeled. The above and below floors are the same space arrangement as the typical floor. The thickness of the concrete floor slab is 150 mm. The exterior wall is 150 mm thick concrete with insulation 25 mm thick foam polystyrene. The single glazed window is used. The ceiling is composed of 9 mm thick gypsum board and 12 mm thick rock fiberboard. The office space is divided into four zones of 151 m^2 floor area as shown in Figure 5. Five package air conditioners with VAV control systems were provided for corresponding to four office zones and a core space.

Figure 6 shows the simulation model of EESLISM. Each zone consists of five system components for the heat and moisture balance of the each. They are an office zone, a ceiling plenum, a converging element CSV, an air conditioner CC, a fan and a VAV unit in the daytime. As shown in Figure 6a, the simulation model of the space cooling system on the floor consists of 30 components including core in the daytime. As shown in Figure 6b, the simulation model of the cooling storage operation consists of 18 components which are four offices zones for the cooling operation of the four ceiling plenums and the core zone with the ceiling plenum in the nighttime. The ordinary space cooling system without the nighttime cooling storage operation named the base system is also simulated for the comparison.

The internal heat sources and the space cooling operation hours are shown in Table 1. For both cases, the operative temperature (defined as mean of room air temperature and mean room surface temperature) is used for the set point in the daytime. The light color blinds are used for the windows faced to east from 8:00 to 11:00, west from 13:00 to 18:00, south from10:00 to 14:00 and all windows from 18:00 to 8:00. In this simulation, convective heat transfer coefficient for the cooling storage operation is 5.12 W/m^{2}K at the floor slab below surface and for all other indoor surfaces 4.5 W/m^{2}K during cooling operation.

Using the weather data of Tokyo the hour by hour simulation was carried out for four months from June

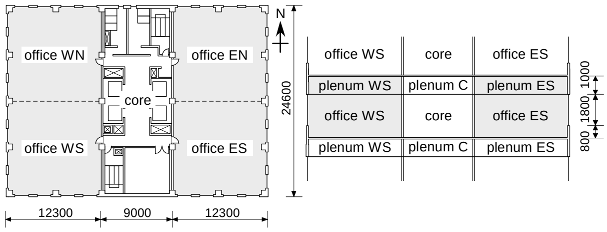
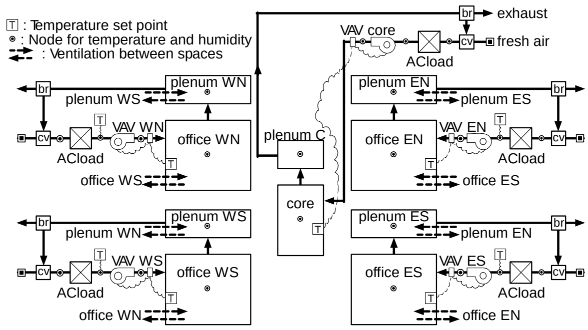
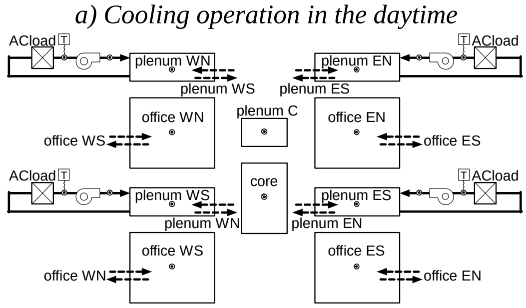


Figure 5 Floor plan and cross section of model building



a) Cooling operation in the daytime



b) Cooling storage operation in the nighttime

Figure 6 System simulation model of fabric heat storage system

Table 1
Schedules of internal heat sources and cooling

	Office		Core	
	office space	plenum	core space	plenum
Internal heat source (8:00-18:00)				
Light [W/m^2]	15	5	7.5	2.5
Appliance* [W/m^2]	10	-	-	-
Occupant* [person/m^2]	0.2	-	-	-
Air conditioning				
Cooling set point for the space** (8:00-18:00)	OT 26 °C VAV	-	OT 26 °C VAV	-
Heat storage cooling*** (3:00-8:00)	-	Outlet air 15 °C	-	-

OT: operative temperature
* 50 % reduction from 12:00-13:00
** Fresh air rate 25m^3/h/person for office and core spaces from 8:00-18:00
*** Not applicable for base case

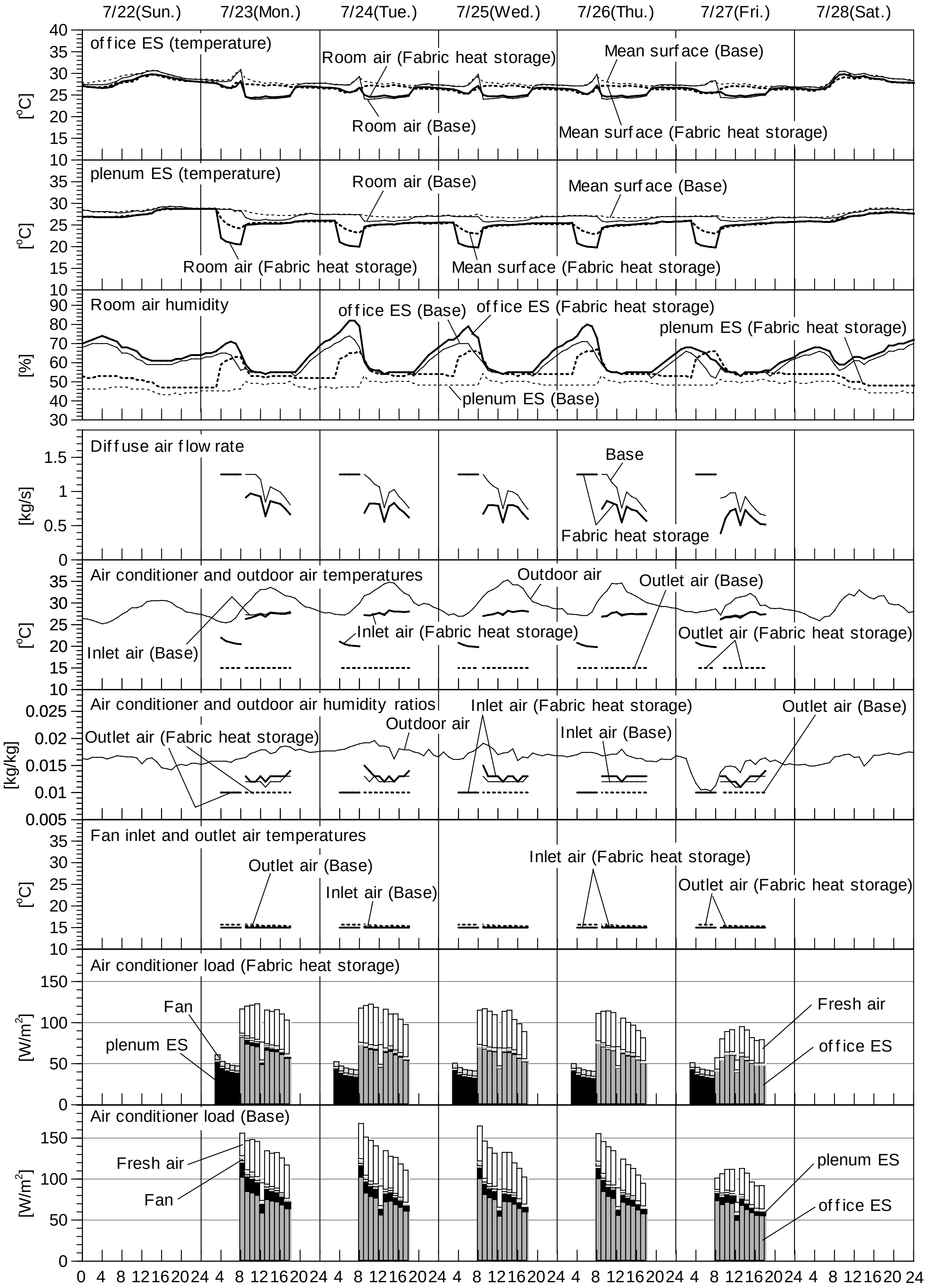


Figure 7 Comparison of hourly variation with standard and fabric heat storage system (office ES)

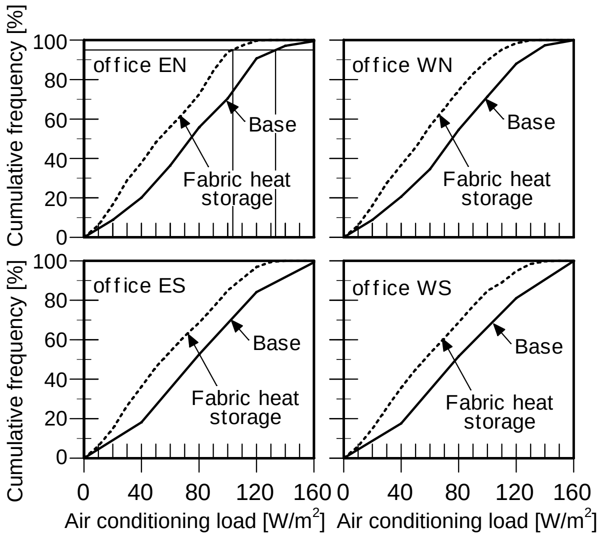


Figure 8 Comparison of cumulative frequency distribution of hourly air conditioner load in daytime

Table 2
Cooling capacities of the air conditioners
(Based on 95 % occurrence)

Zone	Base	Fabric heat storage
EN	131 W/m^2	103 W/m^2
ES	145 W/m^2	117 W/m^2
WN	134 W/m^2	110 W/m^2
WS	143 W/m^2	120 W/m^2

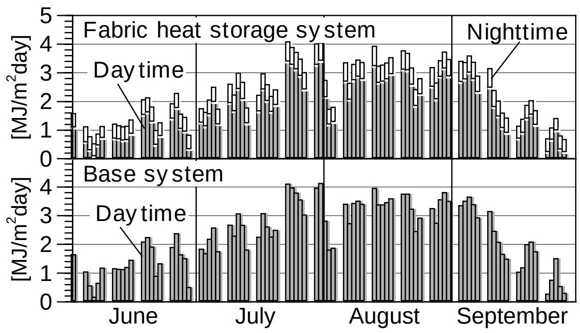


Figure 9 Comparison of daily air conditioner loads

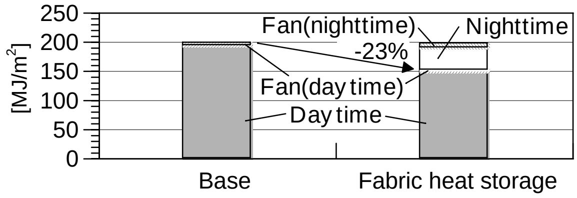


Figure 10 Comparison of seasonal air conditioner loads (June – September)

to September. The weather data used in the simulation was obtained from the Extended AMeDAS (AIJ, 2000) weather data set prepared for the building simulation purpose from the observed weather data by AMeDAS observation system in Japan.

Simulation results

Figure 7 shows the hourly simulation results of a zone, the office ES in hot clear days in July. The room thermal environment and the operating status and heat load of the air conditioning unit are shown for the fabric storage case and the base case. In the daytime, the room thermal environments in both cases are considered as the same, since the operative temperature is used for the set point and the relative humidity is 50-60% in both cases. The cooling load of the fabric heat storage system is smaller than that of the base. Especially, the large difference of 54-20 W/m^2 (49-16 % reduction based of the base case) is observed from morning to the noon. The room air temperature of the fabric storage case in the daytime is higher than the base case by 0.1-1.3 ^{o}C, for the same operative temperature. This implies the effect of the building fabric cooling during night. The higher room temperature also contributes to reduce the ventilation cooling load. The variation of air flow rates show also the variation of cooling loads as the VAV system is assumed for both cases. The set point temperature of the coil outlet is fixed at 15 ^{o}C in both case of operation, the office cooling operation and the fabric storage operation. The coil outlet relative humidity is 90% as the constant relative humidity model of ACload component l is used. For the fabric heat storage case the nighttime cooling is operated at the maximum airflow rate The cooling load in the nighttime is 40-50 W/m^2 and slightly larger than the difference between the cooling load of the fabric storage case and the base case in the daytime.

Figure 8 shows the cumulative frequency distributions of the hourly air conditioning cooling load for four office zones. The total hour is 860 hours corresponding to the daytime cooling operation (10 hours/day) in the four months. When the capacity of the air conditioner is determined by the heat load occurrence of 95 %, 117 W/m^2, the capacity is smaller than the base case by 28 W/m^2 for the office ES. The results of comparison of the capacities are shown in Table 2.

Figures 9 and 10 show the total air conditioner cooling load of four office spaces and a core space. Figure 9 shows the comparison of the daily variation during four months. The average daytime daily air conditioner load of the fabric heat storage system is smaller by 0.54 MJ/m^2/day than that of the standard system. The daily total cooling loads including the nighttime cooling for the fabric storage system are almost the same. The seasonal air conditioner cooling loads are compared as shown in Figure 10. The seasonal total cooling loads are the same and the cooling load of the daytime for the fabric heat storage system is smaller than the base case by 23 %.

Therefore, it was found that the fabric heat storage system is effective for the peak reduction of the cooling load without increasing the seasonal total cooling load.

DISCUSSION

Heat load of air conditioning unit or heating and cooling plant can be obtained by summing up the heat loads of the rooms to which heat is supplied from the heating and cooling equipment. Whereas this may be a usual way of obtaining the heat load of heating and cooling equipment, to develop a generalized algorithm for summing up the room heat load depending on the system schematic diagram is a complicated problem. The algorithm used in EESLISM to obtain the heat load of the heating and cooling equipment is based on the method with the system components and the flow paths as shown in Figures 1 to 4. While the flow paths and flow rates are necessary for running the simulation program, the combined effects of building thermal behavior and mechanical system are automatically considered. The effect of heat recovery system for ventilation air is easily considered. Radiant heating and cooling system are already included in the algorithm (Udagawa, 1993). Double façade system can be calculated. Extending the system application is easy as the component model is used. New or innovative system model can be constructed with connecting the components and define the control strategy. Therefore, this method is very useful for designing any kind of system including ordinary systems. In addition, energy consumption can be simulated using heat load with a maximum capacity and efficiency of heating and cooling equipment.

CONCLUSION

The features and the algorithm of the simulation of heating and cooling loads using system components model and the information related to the system flow paths and control strategy are described using the examples of a single family solar house and the cooling system for office buildings with CAV and VAV systems.

As an example of heat load analysis useful for system design of the room thermal environment and energy system, the fabric heat storage system of the office building using packaged air conditioner was simulated. The heat loads of air conditioning units for four office zones were examined using the simulation results.

Using the cumulative frequency distribution of the heat load the capacity of the air conditioning unit is reduced by 23-28 W/m^2 or 16-21 % for each of the four zones. While nighttime cooling for the heat storage operation necessary, total daily cooling load is almost the same comparing to the base case. The heat storage operation of the nighttime did not increase the daily total cooling load. Therefore, the fabric heat storage system is considered as effective system, since the cooling load of air conditioning unit in the daytime is reduced without increasing the daily total cooling load as well as the seasonal cooling load.

REFERENCES

Architecture Institute of Japan (AIJ), 2000. Expended AMeDAS weather data. (in Japanese, English version 2003)

ASHRAE, 2001. ASHARE Fundamental Handbook, Chapter 29, ASHRAE

Crawley, D. B., Lawrie, L. K., et. al. 1999. Energplus, a new-generation building energy simulation program, Proceedings of Building Simulation ’99, Vol.1, 81-88, September 13-15, Kyoto, Japan.

Roh, H., Udagawa, M. 2004. Simulation study on solar house with floor and hot water heating system, Journal of Asian Architecture and Building Engineering, Vol.3, No.2, 239-246.

The Society of Heating, Air-Conditioning and Sanitary Engineers of Japan (SHASEJ) 2005. Present State and Future of Heat Load and Air Conditioning System Software, Chapter 4, SHASEJ, 2005

Udagawa, M, Nagata, T. 2003. Total energy performance of the air conditioning system using outdoor air cooling potential, Proceedings of Building Simulation 2003, 1309-1316, August 11-14, Eindhoven, Netherlands.

Udagawa, M, Roh, H. 2002. Maximizing the ventilation rate for cooling energy reduction, Advances in Building Technology, Vol.2, 1367-1374, December 4-6, Honking, China.

Udagawa, M., Sato, M. 1999. Energy simulation of residential houses using EESLISM, Proceedings of Building Simulation ’99, Vol.1, 91-98, September 13-15, Kyoto, Japan.

Udagawa, M. 1993, Simulation of panel cooling systems with linear subsystem model, ASHRAE Transactions, Vol. 99, Part 2, 534-547.

Udagawa, M. 1986, Calculation Methods of Air-Conditioning Systems with Personal Computers, Ohm-sha (Tokyo). (in Japanese)