Dynamic Simulation of Annual Energy Consumption in an Office Building by Thermal Resistance-Capacitance Method

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Key Words : Thermal Resistance-Capacitance Method, Office Building, Annual Energy Consumption

Abstract

The basic heat transfer process that occurs in a building can best be illustrated by an electrical circuit network. Present paper reports the dynamic simulation of annual energy consumption in an office building by the thermal resistance capacitance network method. Unsteady thermal behaviors and annual energy consumption in an office building were examined in detail by solving the simultaneous circuit equations of thermal network. The results are used to evaluate the accuracy of the modified BIN method for the energy consumption analysis of a large building.

Present thermal resistance-capacitance method predicts annual energy consumption of an office building with the same accuracy as that of response factor method. However, the modified BIN method gives 15% lower annual heating load and 25% lower cooling load than those from the present method.

Equipment annual energy consumptions for fan, boiler and chiller in the HVAC system are also calculated for various control systems as CAV, VAV, FCU+VAV and FCU+CAV. FCU+CAV system appears to consume minimum annual energy among them.

1. Introduction

Today, energy consumption for the heating and cooling of buildings have increased gradually with the construction of large and deluxe office buildings. Therefore, many scientific researches on the energy consumption of large office buildings have been performed to achieve the national energy saving. Computer control is widely used for the efficient operation of energy equipments in consideration of indoor and outdoor environmental conditions. But the more important factor for the annual energy saving of office buildings is to select

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the equipments for HVAC system cautiously to reflect the characteristics of the building at the design stage.

For this purpose, more accurate calculation of heating and cooling load are needed and the equipments for the HVAC system should be selected in accordance with the results. DOE, BLAST, NBSLD, TRANSYS, AIRCON, HASP /ALCD have been broadly used to calculate the annual energy consumption of office buil-dings. CEC and CDL-207 are also used widely as simple calculation methods. ASHRAE-4.7 which belongs to BIN method is used for a compact design procedure. In the present study, the dynamic simulation method is employed to estimate the annual energy consumption of a building and based on the results of the an-alysis, optimal control method and equipment types are selected. Unsteady heat transfer pro-cesses in the building elements are simulated using electrical networks in which each of wall ele-ment is transformed to a capacitance and two resistances. The results are compared with those from the modified BIN method.

The amount of energy consumption for va-rious air conditioning control methods (VAV, CAV, FCU+VAV, FCU+CAV) and HVAC sys-tem equipments, fan, chiller and boiler are also compared with each other.

2. Theoretical Analysis

Comfortable air conditioning of a building requires the precise control of indoor tempera-ture and humidity within a certain limit. But the continuous variation of surrounding con-ditions may induce the unsteady heat transfer through the walls and windows. It makes hea-ting and cooling loads calculation of the buil-ding very difficult. However, in the present study, the complex heat transfer process through the building elements can be calculated simply by using the thermal network method. The thermal network method will be explained in the following section.

2.1 Calculation of indoor heating load

2.1.1 Solar radiation

The total short-wave irradiance I_t reaching a terrestrial surface is the sum of the direct solar radiation I_D, and the sky and solar ra-diation reflected from surrounding surfaces I_d

$$I_t = I_D + I_d \tag{1}$$

Fig.1 shows the solar angles for the vertical and horizontal surfaces. The irradiance of the direct component is the product of the direct normal irradiation I_{DN} and the cosine of the angle of incidence θ between the incoming solar rays and a line normal to the surface.

$$I_D = I_{DN} \cos(\theta) \tag{2}$$

$$\cos(\theta) = \sin(h)\cos(\phi) + \cos(h)\cos(\phi)\cos(\psi - \beta) \tag{2a}$$

where θ is incidence angle of solar rays to

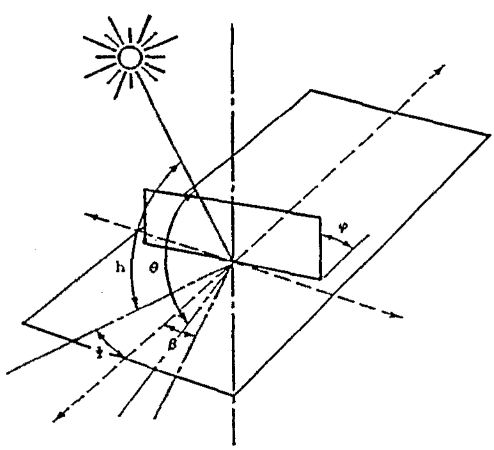


Fig.1 Solar angles for tilted and horizontal surfaces

the surface, h is the solar altitude above the horizontal, ψ is the azimuth measured from the south and β the positive surface azimuth. The diffusive radiation I_d is the sum of diffusive sky irradiations I_S and solar radiation reflected from surrounding surface I_R

$$I_d = I_S + I_R \tag{3}$$

$$I_S = F_S I_{sky} \tag{3a}$$

$$I_R = \rho_G F_G I_{HOL} \tag{3b}$$

where I_{sky} is the horizontal sky irradiations, $F_S(=1+\cos(\phi)/2)$ is the shape factor of inclined surface to sky radiation, ρ_G is the reflection rate to the ground solar radiation, $F_G =(1-F_S)$ is the shape factor of inclined surface to the ground and $I_{HOL}=(I_{DN}\sin(h)+I_{sky})$ is global irradiance.

2.1.2 Convective heat transfer

Convective heat transfer rate from the wall or window surfaces to the surrounding air can be given by

$$q_c = h(T_\infty - T_s) \tag{4}$$

where T_∞ is the surrounding air temperature and T_s is the surface temperature of building elements. Heat transfer coefficient of indoor wall surfaces for various air flow conditions is shown in Table 1.

The convective heat transfer coefficient on the exterior wall of a building is significantly affected by wind velocity. The wind velocity can be related to heat transfer coefficient as follows :

$$h_0 = aV^2 + bV + c \tag{5}$$

where V is the wind velocity and a, b, c are the experimental constants.

2.1.3 Radiative heat transfer

The interior surfaces of a building, walls, floors, ceilings, windows, furnitures, equipments and human bodies may exchange heat through the radiation arisen from the temperature differences between them. Net radiative energy absorbed by a surface i can be given by

$$q_{R,i} = \sum_{j=1}^{N} \sigma F_{i,j}(T_j^{*4} - T_i^{*4}) + q_{r,int} + q_{r,sol} \tag{6}$$

where N is the total number of surfaces, σ $(=5.699\times10^{-8}\ W/m^2K^4)$ is the Stefan-Boltzmann constant, $F_{i,j}$ is the radiative heat exchange coefficient, T^* is the absolute temperature of a surface, $q_{r,int}$ is the radiative energy absorbed from the internal heating elements and $q_{r,sol}$ is the absorbed energy from the transmitted solar rays.

Net radiative energy per unit area absorbed by the exterior surface of the building is

$$q_r = a_S I_T - \varepsilon F_S RN \tag{7}$$

where a_S is the absorptivity of the surface and RN is the night radiation. Difference between the radiation of the ground surface R_{GRD} and the sky radiation R_{sky} is defined as the night radiation RN such as

$$RN = R_{GRD} - R_{sky} \tag{8}$$

Sky radiation which means the long wave

Table 1 Convective heat transfer coefficients of inner wall surfaces

Ventilation rate	Convective heat transfer coefficient(W/m²·℃)
Low	2.3
Medium	3.4
High	4.5
Very high	6.8

length radiation can be calculated by the following equation.

$$R_{sky}=\left(\left(1-0.62\frac{cc}{10}\right)B_r+0.62\frac{cc}{10}\right)$$
$$\sigma(T_a+273.15)^4 \qquad (9)$$

where cc is the cloud amount, $B_r=(0.51+0.209\sqrt{P_u}$ is the radiation rate and P_u is the water vapor pressure.

Assuming the earth surface is black, the night radiation rate is given by

$$RN=\sigma(T_a+273.15)^4-R_{sky} \qquad (10)$$
$$=\left(1-0.62\frac{cc}{10}\right)(1-B_r)\sigma(T_a+273.15)^4$$

2.1.4 Thermal resistance-capacitance method

In the thermal resistance-capacitance method, one approximates a building as being composed of a finite number of parts, N, called nodes, each of which is assumed to be isothermal. To model the heat exchange between the elements, each node is depicted by resistance and capacitance, thus forming a thermal network. Unsteady heat transfer for an intermediate nodal point of wall is given by

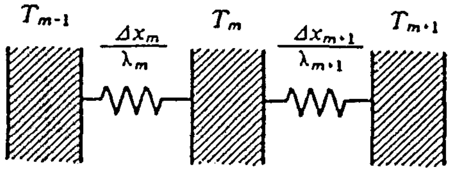
$$\frac{\Delta E}{\Delta t}=q_e-q_w \qquad (11)$$

If a simple node is treated as one thermal capacitance and two resistances as shown in Fig.2, internal energy change for an arbitrary nodal point m is expressed as

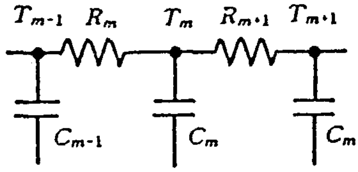
$$\frac{\Delta E}{\Delta t}=C_m\frac{T_m-T_m^*}{\Delta t} \qquad (12)$$

and the heat transfer rates between the nodes are

$$q_e=\frac{1}{R_{m+1}}(T_{m+1}-T_m),$$
$$q_w=\frac{1}{R_m}(T_m-T_{m-1}) \qquad (13)$$



(a) Thermal circuit



(b) Electrical equivalent circuit

Fig.2 Transformation of wall element in electrical circuit

Then, equation (11) becomes

$$C_m\frac{T_m-T_m^*}{\Delta t}=\frac{1}{R_{m+1}}(T_{m+1}-T_m)$$
$$-\frac{q}{R_m}(T_m-T_{m-1}) \qquad (14)$$

where C_m, R_m, R_{m+1} are given by

$$C_m=\frac{1}{2}C_m\rho_m\Delta x_m+C_{m+1}\rho_{m+1}\Delta x_{m+1} \qquad (15)$$

$$R_m=\frac{\Delta x_m}{\lambda_m},\quad R_{m+1}=\frac{\Delta x_{m+1}}{\lambda_{m+1}} \qquad (16)$$

Transforming the exterior and interior surfaces of a wall by above-mentioned thermal network, the heat balance equations for the nodal points can be expressed as

$$C_m\frac{T_m-T_m^*}{\Delta t}=h_0(T_a-T_m)+(a_S I_T$$
$$-\varepsilon F_S RN)-\frac{1}{R_M}(T_m-T_{m-1}) \qquad (17)$$

$$C_1\frac{T_1-T_1^*}{\Delta t}=h_0(T_2-T_1)-h_{ic}(T_1-T_R)$$
$$+h_{ir}(T_1-T_f)+q_r) \qquad (18)$$

Solving simultaneously the differential equations (14), (17) and (18), unsteady temperature variation of each nodal point can be calculated.

2.1.5 Heat balance between a window and its surrounding

If solar rays radiate to a window, the temperature of the window rises up above the surrounding air temperature. The heat diffuses to the surrounding space and air through radiation and convection. Heat transfer rate from the interior surface of a window to indoor space is expressed as

$$q_{Rci} = N_i(\alpha I_T) + U(T_0 - T_i) \tag{19}$$

where N_i is the direction coefficient of absorbed radiant energy. Heat transfer rate from the two glazing glass to indoor space is obtained from

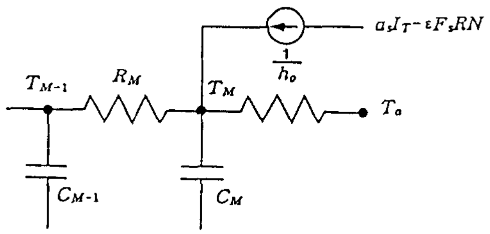


Fig.3 Electrical equivalent circuit of a wall element edjacent to outer air

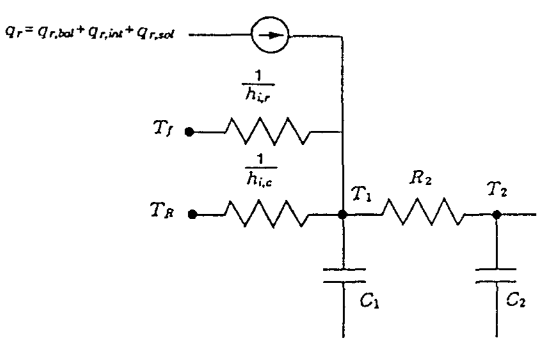


Fig.4 Electrical equivalent circuit of a wall element adjacent to inside room air

$$q_{Rci} = N_{i0}(\alpha_0 I_T) + N_{ii}(\alpha_i I_T) + U(T_0 - T_i) \tag{20}$$

In the above expression N_{i0} is the direction coefficient of outside glass and N_{ii} is the direction coefficient of inside glass. Following relations are used to calculate N_{i0} and N_{ii}

$$N_{i0} = U/h_0, \quad N_{ii} = U/h_0 + U/h_s \tag{21}$$

where h_s is the heat transfer coefficient of intermediate air layer. Thus the heat influx through a window is given by

$$q_{Rci} = U\left[\frac{\alpha_0 I_T}{h_0} + \alpha_i I_T\left(\frac{1}{h_0} + \frac{1}{h_s}\right) + T_a - \frac{\varepsilon F_s RN}{h_0} - \frac{h_{ic}T_R + h_{ir}T_f + q_r}{h_{ic} + h_{ir}}\right] \tag{22}$$

Then, heat balance between a window and its surrounding may becomes

$$q_{Rci} + h_{ic}(T_R - T_S) + h_{ir}(T_f - T_S) + q_r = 0 \tag{23}$$

2.1.6 Heat gain from the internal heating elements

(1) Heat gain from human body

Heat gain from human body is composed of sensible heat gain through convective and radiative heat transfer and latent heat gain through breathing. It depends upon the room temperature and occupants' activity. For any working conditions, sensible and latent heats may be influenced by the surrounding air temperature, but the total amount of heat is not. If we denote the number of occupants, occupation rate and sensible and latent heats per an occupant to N_H, f_H, H_S and H_L, the sensible heat Q_{HS} and latent heat Q_{HL} of total occupants can be given by

$$Q_{HS} = f_H N_H H_S \tag{24}$$

$$Q_{HL} = f_H N_H H_L \tag{25}$$

$$H_S = H_{S24} - d(T_r - 24) \tag{26}$$

$$H_L = H_T - H_S \tag{27}$$

where H_{S24} is the sensible heat for the room temperature of 24℃ and H_T is total heat.

(2) Heat gain from light

Heat gain from the light takes most part of interior heating of an office building and it produces large cooling load. Since the heat gain from the light of indoor and ceiling spaces is all from the sensible heat, cooling load of a building due to light can be calculated by

$$Q_{LR} = (1-P)Q_{Light} \tag{28}$$

$$Q_{LP} = PQ_{Light} \tag{29}$$

$$Q_{Light} = f_L bE_{Light} \tag{30}$$

Where P is the radiation rate in the inner space of ceiling, f_L is lighting rate, b is ballaster loss rate and E_{Light} is heat generation rate from lamps.

(3) Heat gain from heat generating equipments

If we denote the sensible and latent heats of heat generating equipments and their using rate to W_{AS}, W_{AL} and f_A, the amount of sensible and latent heats are written as

$$Q_{AS} = f_A W_{AS} \tag{31}$$

$$Q_{AL} = f_A W_{AL} \tag{32}$$

2.1.7 Infiltration heat gain

Infiltration is the uncontrolled air flow though all the little cracks and openings in a building. Outside air infiltration causes both sensible and latent heat loss. The energy required to raise the infiltration air temperature to that of indoor air corresponds to the sensible heat loss and the energy associated with the net loss of moisture from the space comes from the latent heat.

Sensible and latent heat gains Q_{IS}, Q_{IL} through infiltration are given by

$$Q_{IS} = C_a \rho V_0 (T_a - T_R) \tag{33}$$

$$Q_{IL} = \gamma \rho V_0 (T_a - T_R) \tag{34}$$

where V_0 is the infiltration air velocity. The number of infiltration air exchange is generally calculated by the air exchange method[11]

$$\textit{number of air exchange} = a + bV + c(T_0 - T_i) \tag{35}$$

where a, b, c are experimental constants and V is wind velocity.

2.1.8 Heat balance equation for indoor air

If indoor air and wall surface temperatures are denoted to T_R and T_S, one may arrive at the following heat balance equation for the indoor air.

$$C_R \frac{\Delta T_R}{\Delta t} = \sum_{N=1}^{NW} A_n h_{ic,n} (T_{s,n} - T_R) + C_a C_0 (T_a - T_R) + Q_{c,int} - H_{ES} \tag{36}$$

where NW is the total number of interior surfaces,

$$C_R = C_a G_R + C_{furniture} \tag{37}$$

is the heat capacity of a furniture, $Q_{c,int}$ is the internal heat generation rate and H_{ES} is heat removal rate by a ventilation system.

2.1.9 Calculation of mass balance

Mass balance equation of indoor air is given as follows :

$$\gamma G_R \frac{\Delta x_R}{\Delta t} = \gamma G_0 (x_a - x_R) + Q_{L,int} - H_{EL} \tag{38}$$

$$Q_{L,int} = Q_{HC} + Q_{AL} \tag{39}$$

2.2 HVAC system load

Heat gains from the HVAC system equipments are called HVAC system load. HVAC system load includes heat gain and loss from humidifier, hot and cool air, ducts and fans.

Dehumidification and heating process are calculated by use of phychrometric chart. Many methods are used to control HVAC system. The schematic diagram of constant air volume system(CAV) and variable air volume system (VAV) are shown in Fig.5 and Fig.6. VAV, CAV, FCU+VAV and FCU+CAV methods are usually applied to the office building modeling and the annual energy consumption due to the application of those control systems are investigated by the present dynamic simulation method.

2.3 Calculation of indoor temperature and humidity

Indoor temperature and humidity variation of air conditioning space are calculated by the following energy and mass balance equations.

$$C_R\frac{\Delta T_R}{\Delta t} = \sum_{n=1}^{NW} A_n h_{ic,n}(T_{s,n} - T_R) + C_a C_0(T_a - T_R) + Q_{c,int} - C_a G_{DZ}(T_R - T_{RD}) \qquad (40)$$

$$\gamma G_R\frac{\Delta x_R}{\Delta t} = \gamma G_0(x_a - x_R) + Q_{L,int} - \gamma G_{DZ}(x_R - x_{RD}) \qquad (41)$$

2.4 Outline of the model building

The model building is located in Seoul area of latitude 37.6° north and exposed to the north. The building has 5 floors underground and 15 above ground. Plan view of standard floor is shown in Fig.7 and wall areas of each floor is shown in Table 2.

Each floor is divided into two region, air conditioning area and non-air conditioning area. Lobby, elevator hall and stair case hall are divided to non-air conditioning area. Air conditioning area is once more divided into 3 outer spaces which face with the south, west and east and an inner space. Floor area of this model building is 730m^2 and total floor area is 9,490m^2.

(1) Air conditioning state

Indoor temperature of the model building is set up to different values according to the seasonal change. Occupied and air conditioning hours of the building are also differently set up for the weekdays, Saturday and holiday.

Indoor temperature is set up to 26℃ for the summer season and 27℃ for the intermediate

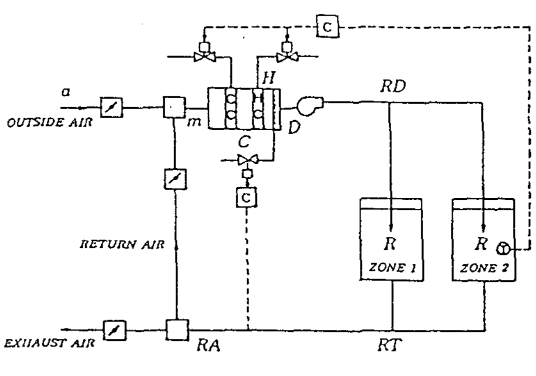


Fig.5 Constant air volume system

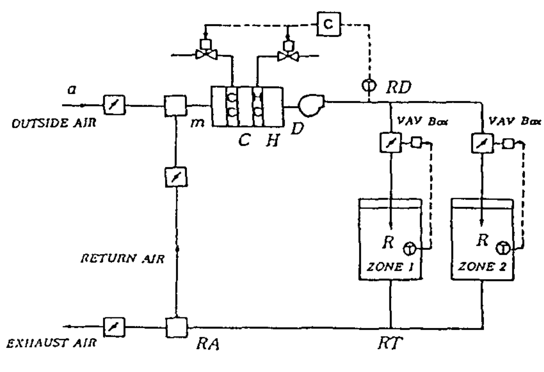


Fig.6 Variable air volume system

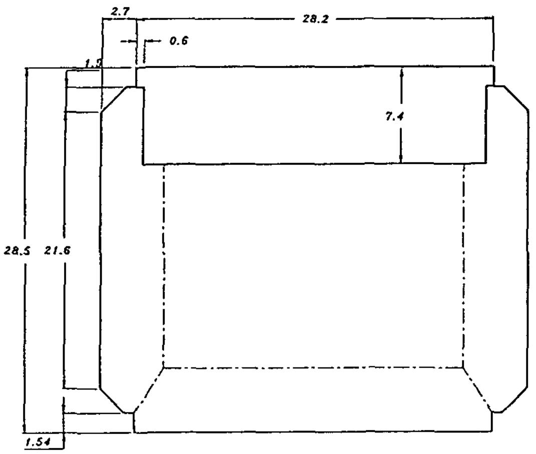


Fig.7 Plan view of standard floor

Table 2 Wall areas of model building

Zone	Area per a floor	Exposed wall area per a floor	Window area per a floor
Inner zone	377.6		
East zone	109.34	66.75	39
South zone	133.04	89.74	26
West zone	109.34	66.75	39

season and 22℃ for the winter season, but the humidity is set up to same value, 50% all through the year.

Air conditioning hour is assumed to be from 7 to 18 o'clock during weekdays but from 7 to 13 o'clock on Saturday but no air conditioning on Sunday.

High heating load occurring at about 8 o'clock is due to the temperature drop of building elements during the non-air conditioning hours.

(2) Internal heat generation

Occupied area per a person is assumed to $6m^2$, the density of light to 12 W/m^2 and the heat generation density to 1 W/m^2.

(3) HVAC system data

HVAC system data for the model building are shown in Table 3~Table 5.

Table 3 HVAC system data of model building

HVAC system	Central and Zonal
HVAC system type	VAV, CAV, FCU+ VAV, FCU+CAV
Minumum cold supply temperature	15℃
Maximum supply hot temperature	40℃
Heat exchanger efficiency	60%

Table 4 Fan data of model building

Supply fan size	41.6㎥	
Return fan size	41.6㎥	
Fan type	Forward curved	
Fan volume control	Speed, Cycling, Vane, Damp	
Supply total pressure	VAV	15kPa
	CAV	0.7kPa
	FCU	0.4kPa
Return total pressure	VAV	0.7kPa
	CAV	0.4kPa
Supply fan/motor efficiency	51%	
Return fan/motor efficiency	51%	

Table 5 Plant data of model building

Cooling lockout temperature	15℃
Cooling equipment type	Centrifugal, Screw, DX(water cooling)
Cooling equipment size	763kW
Condenser water temperature	29.4℃
Heating lockout temperature	21.1℃
Heating equipment type	Steam boiler
Heating equipment size	763kW

3. Results and Discussions

3.1 Heating load of indoor space

In the present study, each element of the model building is transformed into thermal circuit by using the thermal resistances and capacitances method to analyze unsteady heat transfer process. The heat balance equations for the nodal points are solved with variation of influencing parameters.

3.1.1 Calculation of internal heating load

Temperature variation of each element of model building is simulated to obtain the heat removal rate for maintaining the indoor temperature to a setting value. If the amount of heat removal has a positive value, it is regarded as cooling load but a negative value, the heating load. Fig.8 shows the variation of heat removal rate from the indoor area per a day during the winter season for various wall insulation thicknesses.

For a thin insulation thickness, heating load continuously occurs all through the day time in the winter season but the heating load de-creases gradually in the afternoon. However, if the insulation thickness is very thick, hea-ting load occur only in the morning but it changes to cooling load in the afternoon.

3.1.2 Heating and cooling loads variation of indoor space

Fig.9~Fig.11 show the comparison of monthly heating and cooling loads and total load with variation of insulation thickness. Thick insulation may decrease the heating load in the winter season but induce the cooling load in the summer season. Annual heating and cooling loads and total load variation with insulation thickness is shown in Fig.12. Total load appears to decrease with the increase in insulation thickness.

Fig.13 shows the comparison of annual cooling loads predicted by using dynamic simulation methods (present thermal resistance-capacitance method and response factor method) and the modified BIN method. Two dynamic methods give same accuracy in cooling load predictions and show gradual increase of cooling load with insulation thickness. Modified BIN method, however, predict cooling load variation which does not show any insulation

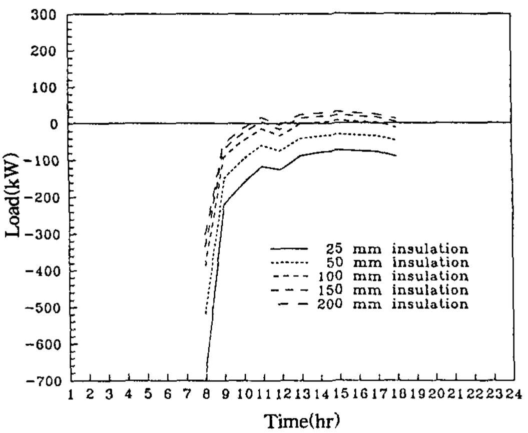


Fig.8 Variation of instantaneous sensible heat load of indoor space with respect to wall insulation thickness during a day in January

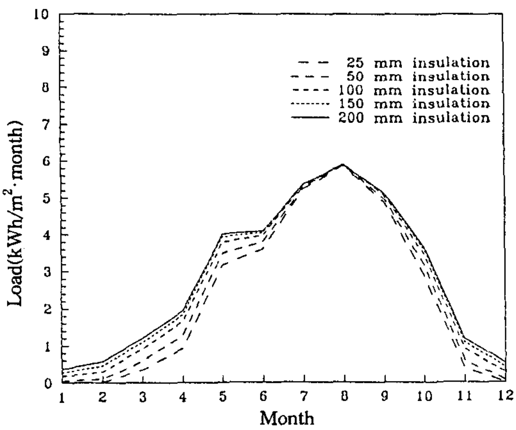


Fig.9 Monthly cooling load with respect to wall insulation thickness

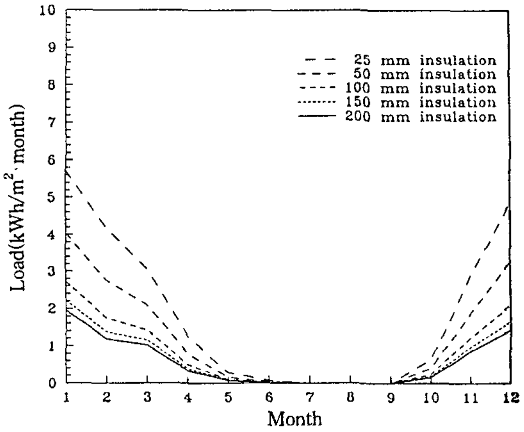


Fig.10 Monthly heating load with respect to wall insulation thickness

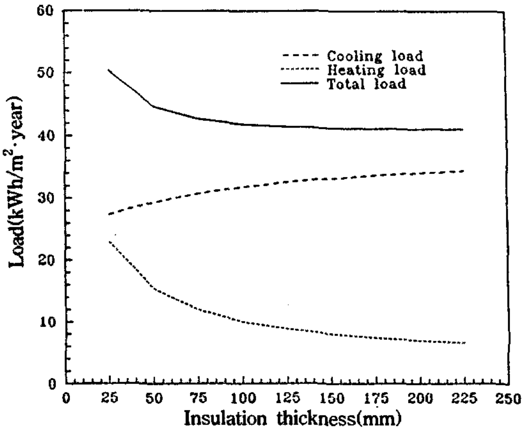


Fig.12 Annual heating and cooling loads with respect to wall insulation thickness

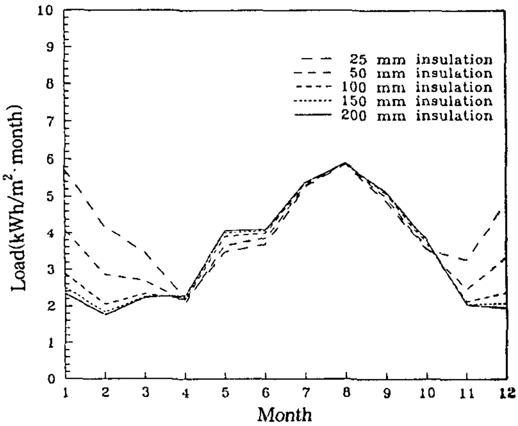


Fig.11 Monthly total load with respect to wall insulation thickness

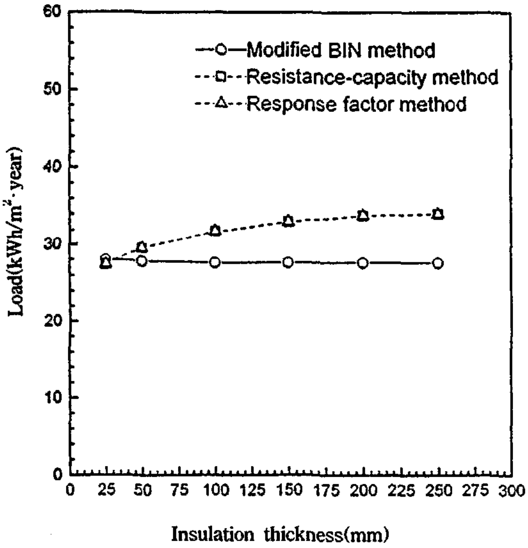


Fig.13 Comparison of annual cooling loads for three different calculation methods with respect to wall insulation thickness

thickness effect.

The comparison of annual heating load for three different calculation methods is shown in Fig.14. Results by the present thermal resistance-capacitance method agree well with that of the response factor method. However, the modified BIN method gives somewhat smaller value than that of dynamic simulation methods. The differences between predictions by the modified BIN method and present methods increase gradually with the increase in insulation thickness.

Annual total loads for five different types of window glasses are compared in Fig.15. The modified BIN method predicts the annual total load of model building a little smaller than that of the present method for all types of glasses.

Table 6 compared the total computing time for annual total load calculation of the three

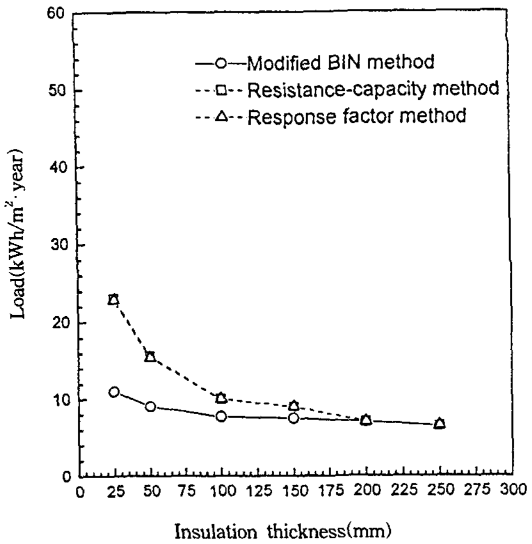


Fig.14 Comparison of annual heating load for three different calculation methods with respect to wall insulation thickness

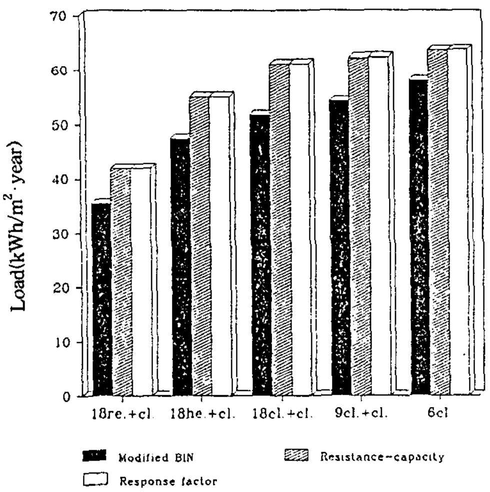


Fig.15 Comparison of annual total load for five different types of window glasses

simulation methods. IBM PC486 DX-2 was used. The response factor method requires CPU time of 3.1 times more than that of the present resistance-capacitance method.

We conclude from the above mentioned results that correction factor of 1.15 should be

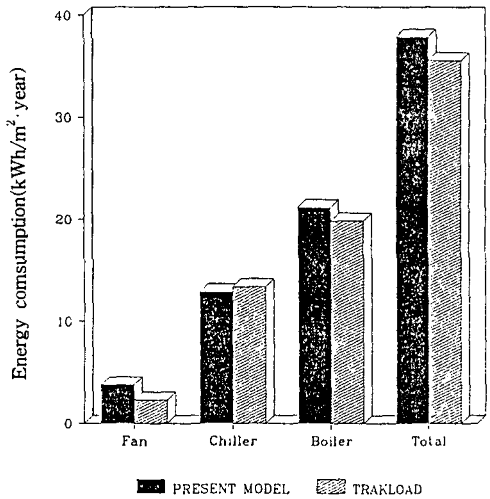


Fig.16 Comparison of annual equipment energy consumption in VAV system for two different calculation methods

Table 6 CPU time for annual total load computation(sec)

Resistance-capacitance method	23.52
Response factor method	72.91

multiplied to the cooling load value by the modified BIN method and 1.25 to the heating load value in order to apply the modified BIN method to practical engineering problem.

3.2 Equipment energy consumption

Present prediction of annual equipment energy consumption for the HVAC system does not show a large difference with that from the TRACLOAD method.

Energy consumptions for the different types of equipments and control methods for the HVAC system are shown in Fig.17~Fig.19. Fig.20 compared the annual equipment energy consumption for different control methods. Energy consumption for the CAV system appears to be much higher than that of the VAV system and fan cooling unit in combination with a duct system gives minimums energy consumption value.

For fan cooling units, the CAV system gives lower energy consumption than that of the VAV system. Smaller variation of the HVAC system load in the CAV system requires lower fan power so that it reduce the total energy consumption of the fan cooling unit.

For a chiller, the CAV system requires larger energy consumption than that of the CAV system but for boiler, the total air system(CAV or VAV) gives a little larger energy consumption than that of the air-water system. Comparing the total energy consumption, the CAV system requires the largest energy consumption but the FCU+CAV requires the least value.

4. Conclusions

In order to calculate the seasonal heating

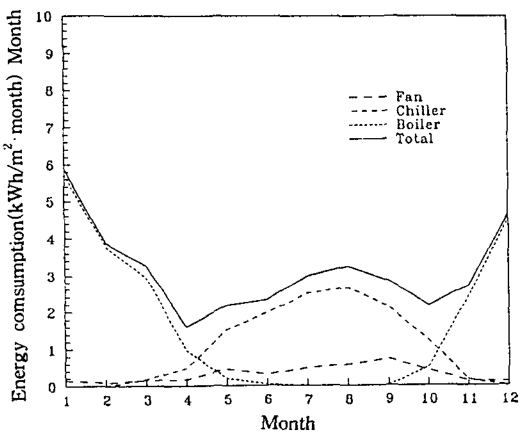


Fig.17 Monthly equipment energy consumption for VAV system

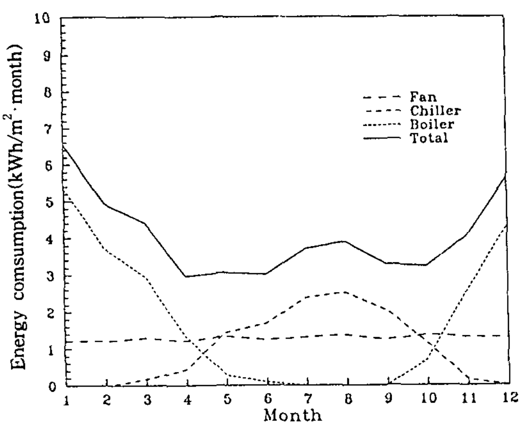


Fig.18 Monthly equipment energy consumption for CAV system

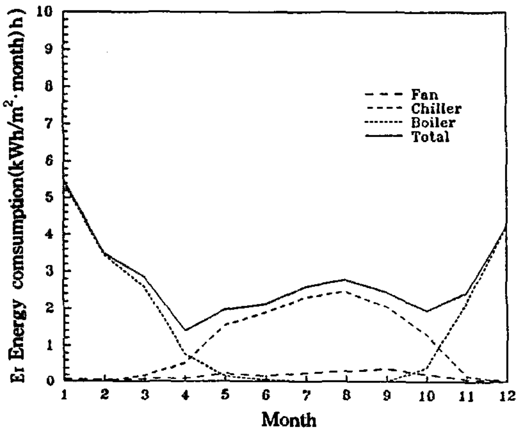


Fig.19 Monthly equipment energy consumption for FCU+VAV system

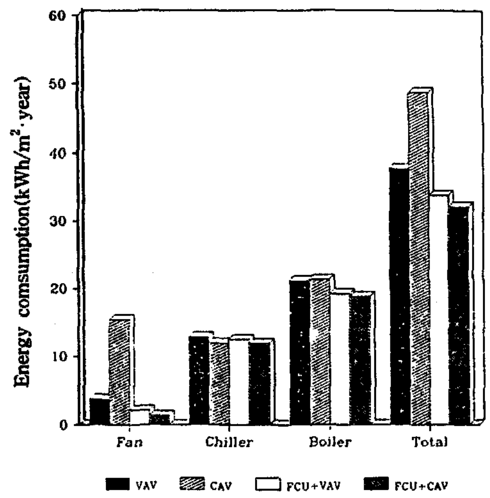


Fig.20 Annual equipment energy consumption for four different air conditioning systems

and cooling loads and energy consumptions for the model office building, the thermal resistance-capacitance method was introduced and the monthly and annual equipment load were calculated. From the results we obtain the fallowing conclusions.

1) The thermal resistance-capacitance method requires only one third of computational time than that of the response factor method to calculate the annual energy consumption of an office building.

2) The modified BIN method predicts annual heating and cooling load calculation much different from that of the dynamic simulation methods. Therefore correction factor of 1.15 should be multiplied to the predicted cooling load value and 1.25 to the heating load value in order to apply it to the practical engineering problem.

3) The total air system require larger annual energy consumption that that of the air-water system. For the total air system, the CAV system requires somewhat larger annual energy consumption that that of the VAV system but for the air water system, the VAV system requires a little larger consumption than that of the CAV system.

Acknowledgments

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