STUDY INTO OPTIMIZED CONTROL FOR AIR-CONDITIONING SYSTEM WITH FLOOR THERMAL STORAGE

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ABSTRACT

Air-conditioning systems with floor thermal storage can be used for cutting peak load and utilizing nighttime electric power. For the effective use of this system, however, thermal energy must be stored during the night in a way that does not waste energy.

In this paper, an optimal heat input to such a system is investigated under prescribed external climatic conditions by making use of the optimal control theory. An optimal heat input to a plenum chamber and an air-conditioned room is determined by minimizing a criterion function which requires little deviation in room temperature from a set-point and low energy consumption.

INTRODUCTION

Recently, various techniques in thermal storage have been extensively investigated in order to conserve energy and to save cost from a total point of view including air-conditioning systems. There are several kinds of media for thermal storage such as water, ice, phase change material and earth ground. An air-conditioning system with floor thermal storage can reduce initial costs by utilizing the floor slab as a storage material, and also provide the thermal comfort by making use of the long wave radiation from the floor. Furthermore, it is shown that this system is effective in cutting peak load and utilizing night electric power [1) 2) 3)].

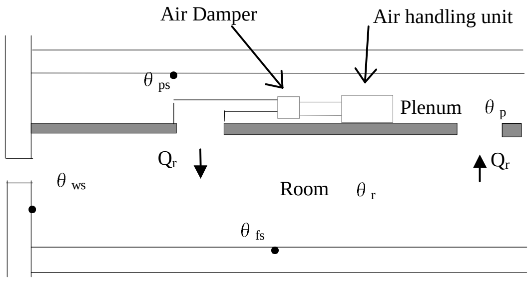

Fig.1. Room Model

For the effective use of this system, however, thermal energy must be stored during night in an optimal way without wasting energy. For that purpose, precise prediction of the outdoor temperature and also a stochastic optimal control of the system are essential [4)5)]. In this paper, an optimal heat input to the system is investigated under prescribed external climatic conditions by making use of the optimal control theory, as the first step to the stochastic problems.

1. FORMULATION OF OPTIMAL CONTROL PROBLEM FOR FLOOR THERMAL STORAGE

1.1 Optimal Control of Thermal Storage

Recently, experimental and analytical studies on the floor thermal storage have been investigated in order to reduce capacity of the system and the initial cost, to cut peak load during daytime by utilizing night electric power in summer and to reduce the running cost [1) 2) 3)]. These researches demonstrate the effectiveness of the air-conditioning system with the floor thermal storage. But the efficiency of the thermal storage depends strongly on the heat loss and the cost of electricity. Although the electric comsumption depending on supply air temperature and storage period [1)] has been examined, a constant storage rate is assumed there. Needless to say, the effective thermal storage taking into account heat loss and required cost depends on the starting time and duration of storage and the time profile of heat input. From such a point of view, objective of this paper is to determine an optimal control to floor thermal storage by making use of the optimal control theory.

1.2 Model of Room and Air-Conditioning System

An air-conditioning system with floor thermal storage controled by air dampers is shown in Fig. 1 schematically. In this air-conditioning system, the cold or hot air from an air-handling unit in the ceiling is blown into the room. The return air from the room is mixed with the air in the plenum chamber to be inhaled into the air-handling unit during daytime.

During nighttime, the air is blown against the concrete slab to store heat by changing the damper position to the plenum.

1.3 Fundamental Equations

With respect to an office building equipped with this system, the heat balance equations of the walls, floor, room air and the air in the plenum chamber are described below. For simplicity, the boundary between the room and the interior space, and the external wall of the plenum chamber are assumed as perfectly insulated.

1) wall

$$c_w \gamma_w \frac{\partial \theta_w}{\partial t} = \lambda_w \frac{\partial^2 \theta_w}{\partial x^2} \qquad (1)$$

2) floor

$$c_f \gamma_f \frac{\partial \theta_f}{\partial t} = \lambda_f \frac{\partial^2 \theta_f}{\partial x^2} \qquad (2)$$

3) room air

$$c\gamma V_r \frac{d\theta_r}{dt} = S_{rw}\alpha_{ir}(\theta_{ws} - \theta_r) + (S_g k_g + c\gamma V_r n) \times (\theta_o - \theta_r) + S_f \alpha_{ir}(\theta_{fs} - \theta_r) + S_c k_c (\theta_p - \theta_r) + q_{sol} + q_{in} + c\gamma Q_r (\theta_p - \theta_r) + g_r u \qquad (3)$$

4) air in the plenum chamber

$$c\gamma V_p \frac{d\theta_p}{dt} = S_f \alpha_{ip}(\theta_{ps} - \theta_p) + S_c k_c (\theta_r - \theta_p) + c\gamma Q_r (\theta_r - \theta_p) + g_p u \qquad (4)$$

where, $c_w \gamma_w$, $c_f \gamma_f$: volumetric heat capacity of wall and floor, respectively, λ_w, λ_f : thermal conductivities of wall and floor, respectivly, θ_w, θ_f : temperature of wall, floor, $c\gamma$: volumetric heat capacity of air, V_r : room volume, θ_r : room air temperature, S_{rw} : wall area enclosing room, α_{ir} : inside heat transfer coefficient in room, θ_{ws} : wall surface temperature on room side, S_g : window area, k_g : overall heat transfer coefficient of window, n : air exchange rate, θ_o : outdoor temperature, S_f : floor area, θ_{fs} : floor surface temperature on room side, S_c : ceiling area, k_c : overall heat transfer coefficient of the ceiling, θ_p : air temperature in plenum chamber, Q_r : volume rate of supply air to room, q_{sol} : solar radiation, q_{in} : internal heat generation , g_r : unit function related to heat supply to room, u : heat input, V_p : volume of plenum chamber, α_{ip} : inside heat transfer coefficient in plenum, θ_{ps} : ceiling surface temperature on plenum side, g_p : unit function related to heat supply to plenum

1.4 Operation of System and Air Flow Rate

As a control variable, heat input u is adopted. The air flow rates to the room, Q_r, and to the plenum chamber, Q_p, are assumed as constant. From the optimal heat input obtained as a solution to the optimal control problem, the temperature difference between the plenum chamber and supply air is calculated based on the following equations. Heat input u at each mode of operation, supply air volume to the room Q_r, and the unit step functions expressing on-off of heat input g_r, g_p are given as follows.

1) During thermal storage:

$$Q_r = 0,\quad g_r = 0,\quad g_p = 1$$

$$u = c\gamma Q_p (\theta_{\text{supply}} - \theta_p)$$

2) During air-conditioning:

$$Q_r = Q_{r0},\quad g_r = 1,\quad g_p = 0$$

$$u = c\gamma Q_r (\theta_{\text{supply}} - \theta_p)$$

3) During no running:

$$Q_r = 0,\quad g_r = 0,\quad g_p = 0$$

2. FORMULATION AS OPTIMAL CONTROL PROBLEM

2.1 Procedures

The optimal control theory is applied to the present problem. In the following, the optimal control theory is described first, next the present system is expressed as a set of state equations. Finally, by determining a criterion function for optimization, the optimal control problem is formulated.

2.2 Optimal Control Theory

Consider a system described by the following state equation,

$$\dot{\mathbf{x}}(t) = \mathbf{A}(t)\mathbf{x}(t) + \mathbf{B}(t)\mathbf{u}(t),\quad \mathbf{x}(0) = \mathbf{x}_0 \qquad (5)$$

where, $\mathbf{x}(t)$ is an M dimensional vector corresponding to the temperatures of the walls, floor, room air and plenum chamber, etc. $\mathbf{A}(t)$, $\mathbf{B}(t)$ are M×M dimensional matrix and an M dimensional vector, respectively. $\mathbf{u}(t)$, $\mathbf{x}_0$ and " $\cdot$ " denote the heat input, initial value and time derivative, respectively.

In this study, the optimal control $\mathbf{u}^0(t)$ minimizing the following criterion function is sought.

$$J(\mathbf{u}) = \int_0^T [(\mathbf{x}(t)-\mathbf{s}(t))'\mathbf{R}_1(t)(\mathbf{x}(t)-\mathbf{s}(t))$$

$$+\mathbf{u}'(t)\mathbf{R}_2(t)\mathbf{u}(t)]dt \qquad (6)$$

where, J is the criterion function and $\mathbf{s}(t)$ is the set point value of the state vector $\mathbf{x}(t)$. $\mathbf{R}_1$, $\mathbf{R}_2$ are the weights to each term, " ' " denotes transpose. T represents a terminal time of control. The optimized solution $\mathbf{u}^0(t)$ of equations (5) and (6) is given as follows.

$$\mathbf{u}^0(t) = -\mathbf{F}(t)\mathbf{x}(t) + \mathbf{d}(t) \qquad (7)$$

where,

$$\mathbf{F}(t) = \mathbf{R}_2^{-1}(t)\mathbf{B}'(t)\mathbf{P}(t) \qquad (8)$$

$$\mathbf{d}(t) = \mathbf{R}_2^{-1}(t)\mathbf{B}'(t)\mathbf{q}(t) \qquad (9)$$

An M×M dimensional matrix $\mathbf{P}(t)$ and an M dimensional vector $\mathbf{q}(t)$ are the solutions of the following Riccati equations, respectively.

$$-\dot{\mathbf{P}}(t) = \mathbf{A}'(t)\mathbf{P}(t) + \mathbf{P}(t)\mathbf{A}(t) + \mathbf{R}_1(t)$$

$$-\mathbf{P}(t)\mathbf{B}(t)\mathbf{R}_2^{-1}(t)\mathbf{B}'(t)\mathbf{P}(t),\ \mathbf{P}(T) = \mathbf{0} \qquad (10)$$

$$-\dot{\mathbf{q}}(t) = \left[\mathbf{A}(t) - \mathbf{B}(t)\mathbf{R}_2^{-1}(t)\mathbf{B}'(t)\mathbf{P}(t)\right]'\mathbf{q}(t)$$

$$+\mathbf{R}_1(t)\mathbf{r}(t),\ \mathbf{q}(T) = \mathbf{0} \qquad (11)$$

Since the terminal conditions $\mathbf{P}(T)$ and $\mathbf{q}(T)$ are given, these equations must be solved backward against time from $t = T$.

2.3 Discretization

The wall and floor temperatures are expressed as a set of state variables by discretizing the basic equations. For simplicity, the walls and the floor are assumed as single layers made of concrete, and discretized into three grid points (thermal masses).

1) Discretized equations of wall

$$\dot{\theta}_{w1} = 2b_w\left[\theta_{w2} - \theta_{w1} + \frac{\alpha_o \Delta x}{\lambda_w}(\theta_{sat} - \theta_{w1})\right] \qquad (12)$$

$$\dot{\theta}_{w2} = b_w(\theta_{w3} - 2\theta_{w2} + \theta_{w1}) \qquad (13)$$

$$\dot{\theta}_{w3} = 2b_w\left[\theta_{w2} - \theta_{w3} - \frac{\alpha_{ir} \Delta x}{\lambda_w}(\theta_{w3} - \theta_r)\right] \qquad (14)$$

2) Discretized equations of floor

$$\dot{\theta}_{f1} = 2b_f\left[\theta_{f2} - \theta_{f1} + \frac{\alpha_{ip} \Delta x}{\lambda_f}(\theta_p - \theta_{f1})\right] \qquad (15)$$

$$\dot{\theta}_{f2} = b_f(\theta_{f3} - 2\theta_{f2} + \theta_{f1}) \qquad (16)$$

$$\dot{\theta}_{f3} = 2b_f\left[\theta_{f2} - \theta_{f3} - \frac{\alpha_{ir} \Delta x}{\lambda_f}(\theta_{f3} - \theta_r)\right] \qquad (17)$$

3) Room air temperature

$$\dot{\theta}_r = \frac{1}{c\gamma V_r}\{S_{rw}\alpha_{ir}(\theta_{ws} - \theta_r) + (S_g k_g + c\gamma V_r n)$$
$$\times(\theta_o - \theta_r) + S_f\alpha_{ir}(\theta_{fs} - \theta_r) + S_c k_c(\theta_p - \theta_r)$$
$$+ q_{sol} + q_{in} + c\gamma Q_r(\theta_p - \theta_r) + g_r u\} \qquad (18)$$

4) Air temperature in plenum chamber

$$\dot{\theta}_p = \frac{1}{c\gamma V_p}\{S_f\alpha_{ip}(\theta_{ps} - \theta_p) + S_c k_c(\theta_r - \theta_p)$$
$$+ c\gamma Q_r(\theta_r - \theta_p) + g_p u\} \qquad (19)$$

2.4 Criterion Function

The objective of the present optimization is to determine the outlet air temperature that minimizes a criterion function. Two kinds of criterion functions are studied.

The first one requires small deviation in the room temperature from the set-point value during working hours and low energy consumption. Therefore, the criterion function J_1 is given as an integral of the sum of the two squared terms.

$$J_1(u) = \int_0^T [r_1(t)(\theta_r - \theta_s)^2 + r_2(t)u(t)^2]dt \qquad (20)$$

The first term in the integral means the square of the deviation in the room temperature from the set-point value, and the second the square of the heat input. r_1 and r_2 represent the weights to these terms. The optimal heat input to the plenum chamber and the air-conditioned room is determined by minimizing this function.

As the second case, the following criterion function J_2 is adopted that takes into account a deviation of the operative temperature from the set-point temperature and energy consumption.

$$J_2(u) = \int_0^T [r_1(t)(\frac{\theta_r h_c + \theta_{mrt} h_r}{h_c + h_r} - \theta_S)^2 + r_2(t)u(t)^2]dt \tag{21}$$

The first term in the integral means the deviation of the operative temperature from the set-point value θ_s, and the second heat input. The θ_{mrt} is approximately given as follows.

$$\theta_{mrt} = \frac{\theta_{ws} S_{rw} + \theta_{fs} S_f + \theta_g S_g + \theta_c S_c}{S_{rw} + S_f + S_g + S_c} \tag{22}$$

2.5 State Equations

By introducing a vector $\mathbf{x}(t)$,

$$\mathbf{x}(t) = [x_1,\ x_2,\ x_3,\ x_4,\ x_5,\ x_6,\ x_7,\ x_8]'$$
$$= [\theta_{w1},\ \theta_{w2},\ \theta_{w3},\ \theta_{f1},\ \theta_{f2},\ \theta_{f3},\ \theta_r,\ \theta_p]' \tag{23}$$

the above equations (12) to (19) can be written by a state equation as follows.

$$\dot{\mathbf{x}}(t) = \mathbf{A}(t)\mathbf{x}(t) + \mathbf{B}(t)\mathbf{u}(t) + \mathbf{q}_o(t) + \mathbf{q}_{sol}(t) + \mathbf{q}_{in}(t)$$
$$\mathbf{x}(0) = \mathbf{x}_0 \tag{24}$$

where, $\mathbf{A}$ is an 8×8 dimensional matrix and $\mathbf{B}$ is an 8 dimensional vector. The $\mathbf{q}_o(t)$, $\mathbf{q}_{sol}(t)$ and $\mathbf{q}_{in}(t)$ are 8 dimensional vectors related to the outdoor temperature, solar radiation and internal heat generation, respectively.

2.6 Application of Optimal Control Theory to Floor Thermal Storage

The present problem differs from the standard optimal control theory only by the terms $\mathbf{q}_o(t) + \mathbf{q}_{sol}(t) + \mathbf{q}_{in}(t)$ on the right hand side of the state equation (24). Therefore, the original problem is transformed into the form to which the optimal control theory is applicable.

The state variable $\mathbf{x}(t)$ is divided into two parts as follows.

$$\mathbf{x}(t) = \mathbf{x}^a(t) + \mathbf{x}^b(t) \tag{25}$$

where, $\mathbf{x}^a(t)$ and $\mathbf{x}^b(t)$ satisfy the following equations, respectively.

$$\dot{\mathbf{x}}^a(t) = \mathbf{A}(t)\mathbf{x}^a(t) + \mathbf{q}_o(t) + \mathbf{q}_{sol}(t) + \mathbf{q}_{in}(t) \tag{26}$$
$$\mathbf{x}^a(0) = \mathbf{x}_0$$

$$\dot{\mathbf{x}}^b(t) = \mathbf{A}(t)\mathbf{x}^b(t) + \mathbf{B}(t)\mathbf{u}(t) \tag{27}$$
$$\mathbf{x}^b(0) = \mathbf{0}$$

The $\mathbf{x}(t)$, the sum of $\mathbf{x}^a(t)$ and $\mathbf{x}^b(t)$, satisfies the state equation (24). The $\mathbf{x}^a(t)$ can be calculated independently to the optimal control problem becacuse it is not related to control. The criterion function (20) becomes as follows.

$$J_1(u) = \int_0^T [r_1(t)(x_7^b + x_7^a - \theta_s)^2 + r_2(t)u(t)^2]dt \tag{28}$$

Thus, by setting as

$$\mathbf{s}(t) = \theta_s - \mathbf{x}_7^a(t), \tag{29}$$

the equation (28) becomes a criterion function in the same form as equation (6). Therefore, by regarding $\mathbf{x}^b(t)$ as $\mathbf{x}(t)$ in the equations (7) to (11) and solving the optimal control problem, the final solution can be obtained by the addition of $\mathbf{x}^a(t)$ to $\mathbf{x}^b(t)$.

3. OPTIMAL CONTROL OF FLOOR THERMAL STORAGE

The method described in the preceding section is applied to a simple example. First, the calculation conditions are given. Next, the difference of optimal control strategies depending on whether a discount rate of electric power during night is utilized or not is examined. Finally the result is discussed when the second criterion taking into account the operative temperature is adopted.

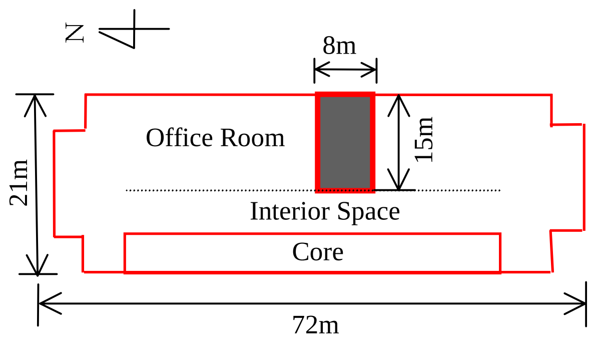

Fig.2. Plan of typical floor in calculated building

3.1 Room Calculated

The plan of a typical floor in the calculated building is shown in Fig. 2. A calculated room is located at the center of the standard floor (shaded area), and it is surrounded by the same type of neighboring rooms except for the east side. The room size is 8m×15m. The height of the ceiling, plenum and windows are 2.6m, 1.1m and 1.3m, respectively. The glazing area on the east side is 10.4m^2.

3.2 Computational Conditions

The computational conditions are as follows.

$c\gamma =$ 0.31 [kcal/m^3℃], $c_w\gamma_w =$ 462 [kcal/m^3℃], $c_f\gamma_f =$ 340 [kcal/m^3℃], $V_r =$ 2.6×8×15 [m^3], $V_p =$ 1.1×8×15 [m^3], $\lambda_w =$ 1.3 [kcal/mh ℃], $\lambda_f = 1.3$ [kcal/mh℃], $\Delta x =$ 0.09 [m], $\alpha_o =$ 20 [kcal/m^{2}h ℃], $\alpha_{ir} =$ 8 [kcal/m^{2}h ℃], $\alpha_{ip} = 13$ [kcal/m^{2}h℃], $n =$ 1 [1/h], $S_{rw} =$ 10.6 [m^2], $S_g =$ 10.2 [m^2], $S_c = S_f =$ 120 [m^2], $k_g =$ 5.27 [kcal/m^{2}h℃], $k_c = 2.82$ [kcal/m^{2}h℃]

The weight function to the room air temperature in the criterion functions, $r_1(t)$, is given a large value during air-conditioning time from 8:00 to 18:00 as shown in Fig. 3, where g1' is a weight during the non-air-conditioning period. It is introduced to include situations where the room temperature during night time should be also evaluated by changing a relative ratio of g1' to g1, although only the case of g1'=0.0 is discussed in the following examples. With respect to the weight function to the heat input, $r_2(t)$, two cases are computed and compared, that is, one with a constant value through a day and the other case with weight by one-third in order to take into account a discount rate of electricity during night time from 22:00 to 8:00. A set point value of the room temperature, θ_s, is set at 26℃. The air volume during the thermal storage is 2000[m^3/h], while 1000[m^3/h] during the air-conditioning time.

The outdoor temperature θ_o is assumed to be given by the following equation.

$$\theta_o = 30 + 5\cos\left[2\pi / 24(t - 14)\right] \qquad (30)$$

Solar radiation is given as the sum of the sky radiation and the direct solar radiation on the east wall on a sunny day. The outdoor temperature and the solar radiation are shown in Fig. 4. An internal heat gain is set at 3,500[kcal/h] during working hours from 8:00 to 18:00.

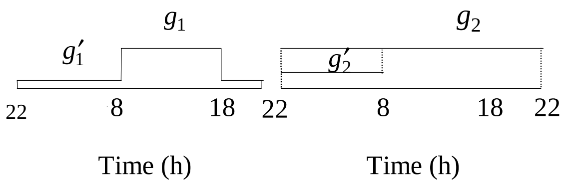

Fig.3. Weight functions $r_1(t)$ and $r_2(t)$

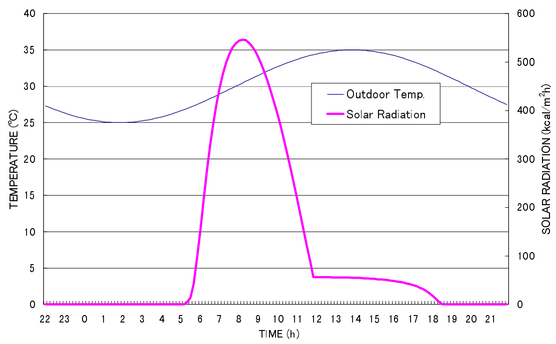

Fig.4. Outdoor temperature and solar radiation

3.3 Optimal Control When Using Cheap Electricity During Night

1) In case of using room air temperature in criterion function (J_1)

A simulation for 3 days is carried out, where the same weight to the heat input is assumed during day and night, that is, $g_1 = 10.0$, $g'_1 = 0.0$, $g_2 = g'_2 = 1.0\times10^{-6}$. As an initial condition, a temperature of 26℃ is given to the room, plenum, walls and floor. The calculated heat input (cooling) and room air, plenum chamber and supply air temperatures are shown in Fig.5. Although the room air temperature starts to rise from about 6 a.m. with the outdoor air temperature and the solar radiation, it is controled close to the set-point value during the air-conditioning time, since the value of g_1, the weight of deviation in the room temperature from the set-point value during the air-conditioning time is large. As the time approaches 18:00, the end of the air-conditioning time, the supply air temperature rises gradually. On the second day, it becomes about 18℃ just before 18:00. As a result of this, the room air temperature increases to about 27℃, and then it begins to drop gradually with the outdoor temperature. It decreases from 22:00 due to a storage operation.

The heat input (cooling) increases gradually from the starting time of the thermal storage, and reaches its peak in the morning due to the solar radiation since the building faces to the east. But, the influence of the solar radiation decreases near 12:00 o'clock, while that of outdoor air temperature becomes larger. The results show that the optimal control is to store heat through the whole storage time and to increase storage rate gradually with time.

The result is shown in Fig. 6 in the case where the heat input term in the criterion function is weighed by one-third during night in order to take into account a discount rate of electricity. The temperature of the plenum chamber becomes lower by about 3 degrees than the case where weight to the heat input term is set at one. At the same time, the peak cooling load of 4,100[kcal/h] (at about 16:00 o'clock) in the case of weight one, is reduced to 3,500[kcal/h] on the first day. Since the cooling load during night increases to about 5,000[kcal/h], however, the larger air-conditioning capacity is required. Also, the room air temperature drops to about 23℃ at 8:00 o'clock, when the cooling storage ends and the air-conditioning starts. The result is to supply heat for warming for a few minutes in order to control the room air temperature at 26℃.

In this paper, the capacity of the air conditioning system with floor thermal storage is not compared with that without floor thermal storage. But, another paper (reference 10) shows that the capacity of the system with floor thermal storage can be reduced by about 30% from 7,000[kcal/h] to 5,000[kcal/h].

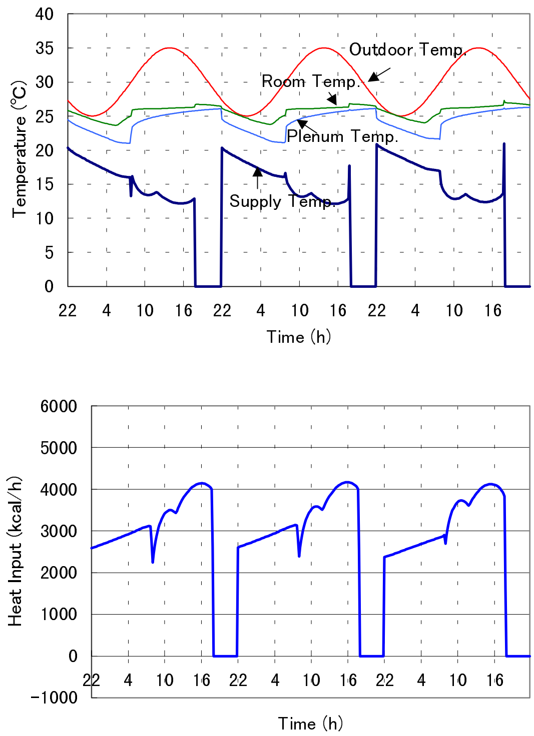

Fig. 5. Room air temperature and heat input

($g_2 = g_2'$)

2) In case of considering radiation (operative temperature) into criterion function (J_2)

A three-day simulation taking operative temperature into consideration is carried out, where the same weight of the heat input is assumed during day and night, that is, g_1=10.0, g_1'= 0.0, g_2 =1.0×10^{-6}. The results are compared with those in the case of using the room air temperature in the criterion function (J_1). The calculated room, plenum chamber, supply air temperatures and heat input are shown in Fig. 7. As a whole, the room and plenum air temperatures considering radiation into the criterion function is higher than those when only the room air temperature is taken into account. The room air temperature in the former case (criterion function J_1) is close to the set point temperature, 26℃, while that in the case of J_2 is a little higher.

The heat input during thermal storage in the case of J_1 is larger than that in the case of J_2. The heat inputs during the air-conditioning time are almost the same in both cases.

A simulation is carried out, where weight of the heat input during night is one-third. The calculated room, plenum chamber, supply air temperatures and heat input are shown in Fig. 8. The heat input in the case of criterion function J_1 is compared with that in the case of criterion function J_2 in Table 1. In the case of

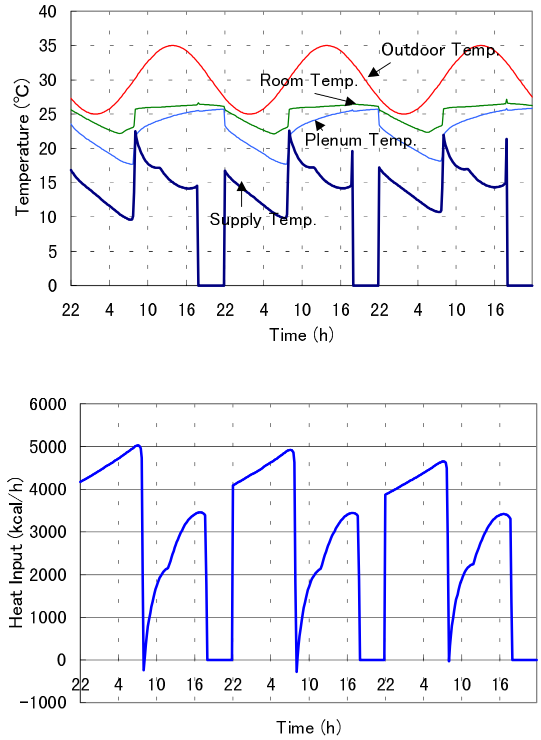

Fig. 6 Room air temperature and heat input (heat input term in the criterion function J_1 is weighed by one-third)

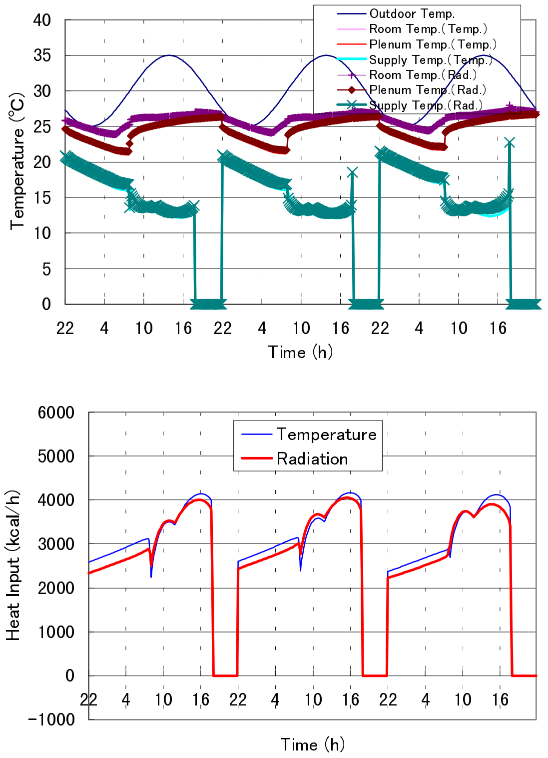

Fig. 7. Room air temperature and heat input (In case of considering radiation into criterion function)

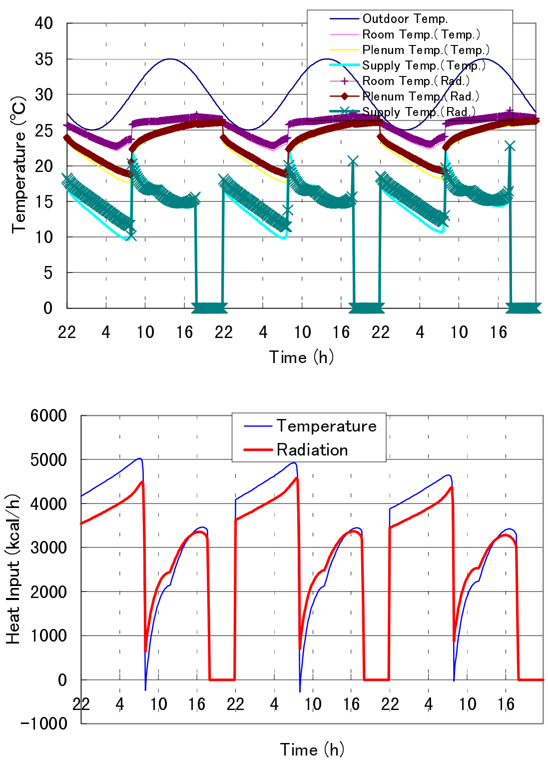

Fig. 8 Room air temperature and heat input (heat input term in the criterion function is weighed by one-third)

J_2, the heat input during the thermal storage is lower than that in the case of J_1, while the heat input during the air-conditioning is higher. Furthermore, the heating (warming) is required at the changing time from thermal storage operation to air-conditioning in the case of J_1, it is not necessary in case of J_2. However, the heat input increases rapidly just before the start of the air-conditioning in the case of J_2.

Table 1. Comparison of heat inputs between J_1 and J_2 (a unit: [kcal/h])

Heat Input	J_1	J_2
Thermal Storage	132,600	117,300
Air-Conditioning	73,300	79,900
Total	205,900	197,200

CONCLUSIONS

In this paper, optimal control strategy of the air-conditioning system with floor thermal storage was investigated based on the optimal control theory under prescribed external climatic conditions. An optimal control of heat input to the plenum chamber and the air-conditioned room was determined on a criterion.

Two kinds of criterion functions are studied. The first one requires small deviation in room temperature from a set-point value and low energy consumption. The optimal heat input to the plenum chamber and the air-conditioned room is determined by minimizing this function. It is shown that the optimized control is to store heat through the whole storage time and to increase storage rate gradually with time. Furthermore, the case is investigated where the heat input term in the criterion function is weighed by one-third to the case above mentioned by taking into account a discount power rate during night. The temperature of the plenum chamber becomes lower by about 3 degrees than the case, where weight to the heat input term in the criterion function is set at one.

As the second case, a criterion that both a deviation of operative temperature from a set-point temperature and an energy consumption should be minimized is adopted. An optimal heat input to the plenum chamber and the air-conditioned room is determined. The cooling load during night storage time is reduced and the fluctuation of the room temperature is smoothed out, compared with the results when a criterion function considering only the room temperature is used. And, in the case of weighing by one-third during night, the heating is not seen which is required in the case of using temperature in criterion.

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NOMENCLATURE

α_{ip} : inside heat transfer coefficient in plenum [kcal/m^{2}h℃]
α_{ir} : inside heat transfer coefficient in room [kcal/m^{2}h℃]
α_o : outside heat transfer coefficient [kcal/m^{2}h℃]
Δx : mesh increment [m]
θ_{fs} : floor surface temperature on room side [℃]
θ_o : outdoor temperature [℃]
θ_p : air temperature in plenum chamber [℃]
θ_{ps} : ceiling surface temperature on plenum side [℃]
θ_r : room air temperature [℃]
θ_{sat} : sol-air temperature [℃]
θ_{ws} : wall surface temperature on room side [℃]
λ_f , λ_w : thermal conductivities of floor and wall, respectively [kcal/mh℃]
a_f , a_w : thermal diffusivities of floor and wall, respectively [m^2/h]
$c\gamma$: volumetric heat capacity of air [kcal/m^3℃]
$c_f \gamma_f$, $c_w \gamma_w$: volumetric heat capacity of floor and wall, respectively [kcal/m^3℃]
g_p : unit function related to heat supply to plenum
g_r : unit function related to heat supply to room
k_c : overall heat transfer coefficient of the ceiling [kcal/m^{2}h℃]
k_g : overall heat transfer coefficient window [kcal/m^{2}h℃]
n : air exchange rate [1/h]
Q_r : volume rate of supply air to room [m^3/h]
Q_{r0} : prescribed volume rate of supply air to room during air-conditioning time [m^3/h]
q_{in} : inside heat source generated by human body, illumination, etc. [kcal/h]
q_{sol} : solar radiation [kcal/m^{2}h]
S_c : ceiling area [m^2]
S_f : floor area [m^2]
S_g : window area [m^2]
S_{rw} : wall area enclosing room [m^2]
u : heat input [kcal/h]
V_p : volume of plenum chamber [m^3]
V_r : room volume [m^3]