Influence of Furniture on the Peak Load of an Intermittent Heating System

-Analysis of Indoor Radiation Heat Transfer-

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ABSTRACT

Room thermal analysis with furniture will generally result in differing heat load values and varying indoor temperatures compared to the same calculation without furniture. This is due to several factors concerning room configuration and its effect on the modes of heat exchange occurring within the indoor environment. The influence of furniture on indoor radiation heat transfer, specifically when computing the peak load of an intermittent heating system, was investigated in this study. The peak load value generally determines the required maximum output capacity of a heating system. Therefore, for reasons of economy and space, it is necessary to calculate an approximate figure.

Computational analysis was done while explicitly computing the radiation heat exchange between all wall and furniture surfaces. A typical office space in Japan was simulated. Several cases with varying furniture configurations were investigated, and peak load values were compared to the case without furniture. During the heating start-up time, with the initial condition of almost uniform temperature everywhere, net radiation heat transfer becomes quite small. The computational results show that if walls and furniture surfaces have similar heat capacity and thermal conductivity, the increase in peak load in the case with furniture will be proportional to the increase in the total surface area on which convective heat transfer with room air occurs. Steady state heat load values in the cases with furniture were also compared and the influence of furniture position and configuration quantified.

Computing the amount of net radiation heat transfer between wall and furniture surfaces requires complicated calculations for the geometrical configuration (shape) factor for each surface-to-surface relation. Another computational analysis was done wherein a partition wall between the interior and perimeter spaces was used to represent furniture in the room. Results from this extreme case show that it can approximate, to a good degree, the influence of furniture on the heat load analysis for an intermittent heating system.

1. INTRODUCTION

1.1 Background

One of the main concerns of the subject of heat transfer in architecture is determining the required space heating and cooling load in order to maintain a comfortable indoor thermal environment for human occupation. In most steady-state heat transfer problems, the conventional method of calculating the amount of heat lost to or gained from the external environment makes use of an overall heat transfer coefficient (U-value). This value includes the interior film coefficient which estimates how easily heat flows through a film of air adjacent to the inner surface of any room enclosure. Usually, the combined surface coefficient of heat transfer, consisting of convective and radiant components, is used as the interior film coefficient.

To obtain reliable results, however, convective and radiative heat transfer must be dealt with separately (ASHRAE 1993). This involves calculating surface-by-surface conductive, convective and radiative heat balance for each room surface and convective heat balance for the room air. In particular, a complicated equation of inter-reflection must be solved for the radiative heat transfer. The computational algorithms and simplified handling have been proposed and investigated from a practical design point of view.

Hutchinson (1964) reexamined the "standard" value of the radiant fraction of the heat transfer through the air film adjacent to the inside surfaces of walls, floor and ceiling given in the 1963 ASHRAE Guide and Data Book. Through the analysis of a cubic room enclosed by a uniform construction and an external air with a uniform temperature, he showed that a significant reduction in the equivalent film coefficient for radiant transfer is possible.

Walton (1980) showed that the approximations of the ASHRAE procedure (1957) in the radiant heat transfer within the room could produce significant errors in calculated room loads. Furthermore, he proposed a different approximation which leads to a more accurate radiant heat balance and whose solution time is in the first order. This method is based on the concept of mean radiant temperature, while the net imbalance of the total radiant energy is handled by its equal redistribution on all surfaces, in order to conserve energy. Although the use of mean radiant temperature is also proposed by other researchers (Davies 1978; Nakamura 1975; Matsuo 1984), the correction of the MRT radiation imbalance introduced first by Walton constituted remarkable progress in the analysis of radiation heat transfer.

Davies (1988) investigated a "binary star" model formed from the radiant and convective star patterns and discussed the possibility of replacing it with a single star system centered on an index temperature like the "environmental temperature" introduced in the UK.

Based on the method proposed by Walton, Steinman et al. (1989) proposed the "MRT correction method", the objective of which was to overcome the limitations of the MRT method while preserving BLAST's (Hittle 1976) concern for computational speed of execution.

It was proven by Hokoi et al. (1994) that, for a single room whose enclosures are of equivalent thermal properties, no net heat transfer by radiation occurs. In this case, only the convective factor of the interior film coefficient should be used. Thus, the conventional solution, which makes use of the standard value of the interior film coefficient, greatly overestimates the peak load of both perimeter and interior spaces.

1.2 Research Objectives

In this study, we investigated how much radiation heat transfer affects computations for the heat load in the case of intermittent heating in a multi-room case. Daily and weekly simulations were done which resemble the operating hours of an intermittent heating system. Of particular interest was the start-up time peak load. A typical Japanese office was simulated in the study. Figure 1 shows the perimeter and interior rooms used in the study.

Room thermal analysis with furniture will generally result in differing heat load values and varying indoor temperatures compared to the same calculation without furniture. This is due to several factors concerning room configuration and its effect on the modes of heat exchange occuring within the indoor environment. A preliminary study on the influence of furniture on indoor radiation heat transfer, specifically when computing for the peak load of an intermittent heating system, was also done.

Computing the amount of net radiation heat transfer between wall and furniture surfaces requires complicated calculations for the geometrical configuration (shape) factor for each surface-to-surface relation (Yoshimura 1986). Another computational analysis was done, wherein a partition wall between the interior and perimeter spaces was used to represent furniture in the room .

2. METHOD OF ANALYSIS

2.1 Fundamental Equations

Following are the fundamental equations used in this paper. Without any loss of generality, air infiltration is assumed to occur through only one external wall for simplification purposes.

(1) Heat balance of room air enclosed by surface S

$$c\gamma V \frac{dT_r}{dt} = c\gamma V n\left[T_{OV}(t) - T_r(t)\right] + \int_S \alpha_c(s)\left[T_W(s,t) - T_r(t)\right]dS + Q(t) \tag{1}$$

Where;

T_r = room air temperature []
= air density [kg/m^3]
c = air specific heat [J/kgK]
V = room volume [m^3]
n = air exchange rate [1/s]
T_{ov} = outdoor temperature []
$_c$ = convective heat transfer coefficient on interior surface [W/m^{2}K]
Q = heat input by air-conditioning system and internal heat source [W/m^2]
S = inner surface area of walls [m^2]
s = position on inside surface of wall
T_w = (inner surface) temperature of wall []
t = time [s]

The first term on the right-hand side of Equation (1) expresses the heat flux caused by the infiltration of air while the second term expresses the convective heat exchange with the inner surfaces of the enclosing walls.

(2) Heat conduction through the wall

$$\frac{\partial T_w}{\partial t} = \frac{\lambda_w}{c_w \gamma_w}\left(\frac{\partial^2 T_w}{\partial x^2}\right) \tag{2}$$

$_w$thermal conductivity of wall [W/mK]
c_w = specific heat of wall [J/kgK]
$_w$ = density of wall [kg/m^3]

(3) Heat balance on the inner surface on the wall

$$\alpha_C(s)[T_r(t) - T_w(s,t)] + q(s,t) = -\lambda_w \frac{\partial T_w(s,t)}{\partial x} \tag{3}$$

q = net radiative heat flux absorbed by wall [W/m^2]

(4) Longwave radiative heat transfer between surfaces

If every surface is assumed to be completely diffusive, the equation expressing the radiation inter-reflection is given as follows.

$$G_{(s)} = \int_S \varepsilon_{(s')} E_{b(s')} F_{(ss')} dS' + \int_S (1-\varepsilon_{(s')}) G_{(s')} F_{(ss')} dS' \tag{4}$$

$G_{(s)}$ = irradiation [W/m^2]
$_{(s')}$= emissivity of inner surface of wall
$F_{(ss')}$ds' = configuration factor (shape factor)
$E_{b(s)}$= emissive power of a black body [W/m^2]

(5) Net radiative heat flux absorbed by the wall

$$q(s) = \varepsilon_{(s)}\left[G_{(s)} - E_{b(s)}\right] \tag{5}$$

(6) Sum of the net radiative heat flux

By making use of Equations (4) and (5), the following equation can be easily derived.

$$\int_S q(s)dS = 0 \qquad (6)$$

2.2 Calculation Procedure

Here, by comparing the heating load of a simple room obtained by the following three methods of thermal calculation, the importance of the radiative heat transfer is investigated.

(1) Radiation heat transfer is calculated exactly by taking into account inter-reflection.

(2) Heating load is calculated by making use of the conventional method of thermal calculation where the combined surface coefficient is used.

(3) The same method is used as in (2), except that only the convective surface coefficient is used.

The result was used as a base reference for studying the effect of net radiation heat transfer on design heat load calculations.

For the numerical computation, each wall and glazing is divided into rectangular cells. The shape factors are calculated by the well-known formula for a rectangular cell (Wiebelt 1966). Radiative heat transfer is calculated by making use of Hottel's method (Wiebelt 1966). Furthermore, wall depth is divided into several layers for finite difference calculation. The explicit finite difference method is used to solve the one-dimensional heat conduction equation.

The room shown in Figure 1 was analyzed here. It was W5D10H4 [m^3] in size, and the walls and glazing face the outside (or the next room). The wall is 12 cm thick RC (thermal conductivity 1.6 W/mK) and the glazing is a 3 mm thick glass (thermal conductivity 0.79 [W/mK]). The internal room enclosures, except the partition wall, absorb heat, but the center layer of the wall, floor and ceiling cross-sections are adiabatic (Figure 2). This assumes that the thermal conditions of all the surrounding rooms are identical to that of the interior and perimeter rooms being studied. The emissivities of glass and walls are set at 0.9 and 0.8, respectively.

Outdoor air and set point room air temperatures are 0 and 20, respectively. The convective, radiant and the combined surface coefficients of heat transfer were set at 3.5, 5.8 and 9.3 [W/m^2K], respectively. Here, a constant value of 5.8 is merely given for the radiant coefficient$_r$. A value close to this is often used as a standard value for thermal design in Japan.

3. ANALYSIS OF THERMAL CONDITIONS OF SPACE

3. 1 Computational Conditions

The thermal properties of the multi-room space shown in Figure 2 were analyzed. The pattern of operation of the intermittent heat supply is as follows.

The first hour of each day is the heating start-up time, during which room air temperatures are raised to 20. These temperatures are then maintained for the next 9 hours by the heating system, after which the temperatures continuously drop until the onset of the heating period of the following day. This pattern of operation is made to resemble the operating hours of an intermittent heating system for an office space where daily working hours are from 8 AM to 5 PM.

After 14 days of intermittent heating, the heating load and temperature values reach a 24 hours-a-day cyclic steady-state. Another computation of 8 days was done wherein during the first 5 days heat was supplied, then cut off for 2 days and then again resumed on the 8th day, to resemble a 5-day work week. The results of these different heating patterns are detailed here following.

3. 2 Results and Discussion

(1) 1-Day Cyclic Steady-State

Perimeter Room

On the first day, all indoor temperatures and room surface temperatures are 0. When heat is first supplied and the desired room air temperature of 20 has been reached, the conventional solution overestimates the peak load by 130% (Figure 3), although at the end of the 10th hour, right before heat supply is cut off, the deviation from the exact solution decreases. The convective solution, on the other hand, computes heat load accurately during the first hours of operation (error of -0.04%), but at the end of the 10th hour, a slight deviation can be seen. The value of the peak load usually determines the maximum capacity of the HVAC system, and from these results, it can be seen that using the peak load results, calculated from the conventional solution, would yield an over-designed heating system.

When the 1-day cyclic steady-state is reached, the convective solution underestimates the load consistently throughout the heating period by -31%. The conventional solution overestimates the daily peak load by 95%, and by 17% at the end of the heating period.

As for the room air temperature during non-heating periods, the convective solution is initially closer to the exact solution, although both the convective and the conventional solutions overestimate the room temperature at all times (Figure 3).

Wall Temperature Profile

The conventional solution gives higher wall temperatures than the exact solution. As for the partition wall, the interior room side surface temperatures are always higher than the perimeter room side except initially during the first day. In contrast, the conventional coefficient solution shows no temperature difference between these two sides during heating system operating hours.

Window surface temperatures were solved using steady-state calculation methods. The conventional solution gives higher surface temperatures at all times, so there is a risk of underestimating the occurrence of condensation when using the traditional method of thermal analysis (Matsumoto et al. 1997).

(2) 1-Week Cyclic Steady-State

A 1-week cyclic steady-state is normally reached after more than two-week calculations. But, our interest is to see how much the peak load changes for the first day of the next week after two days of no heating (Saturday and Sunday). For this purpose, only an 8-day calculation was done.

Perimeter Room

The conventional solution overestimates the heat load by 126% during the 8th day. The difference between the 1st day and the 8th day is the convective solution's consistent deviation from the exact solution (Figure 4). In the first day, the convective solution heat load initially follows the exact solution closely. On the 8th day, it almost consistently underestimates the heat load by -20%. From these results it is evident that the influence of net radiative heat flux increases as the walls and enclosures start to absorb a substantial amount of heat. Room air temperatures are given in Figure 4.

3.3 Several Remarks

(1) Relative importance of radiative heat transfer

Although the situation where a convective film coefficient should be used instead of the combined surface coefficient of heat transfer, has been investigated so far, it must be noted that this is not always the case and there are several situations where the usual combined surface coefficient must be used. The following comments seem appropriate with regard to this point.

Consider a room whose internal walls are all adiabatic. The external temperature is assumed lower than the room air temperature. By exchanging radiation with exterior walls, the adiabatic walls have a net radiation loss so that they will have a temperature below the room air temperature, thus supporting convection from the room air to other than the exterior walls. In this case, radiation is important; the interior walls are increasing the thermal coupling between the room air and the outside air. Thus, the ”film” coefficient must be increased by some amount to account for this radiative effect, depending on the relative size of the transmission walls and adiabatic walls.

In the case of a very small transmitting wall area, the temperature decrease of the internal walls is also very small thus leading to the result that the internal wall temperature almost equals the room air temperature. The appropriate radiant coefficient in this case is the standard one given by ASHRAE Guide and Data Book (1963). As the size of the transmitting wall increases, the weight of the radiant coefficient decreases. With regard to the relative importance of the radiant component, a detailed and thorough examination has been given by Hutchinson (1964) in the case of a cubic room.

(2) Multi-room space

In the case of a multi-room space, it is not possible to obtain an exact heating load by using a convective film coefficient because it is equivalent to regarding the heat flux through a partition wall as heat flux from the outside air (external heat source), that is, neglecting the heating load of the adjacent room.

Consider a partition wall that separates two symmetrical rooms. Here, the boundary condition at the center of the partition wall becomes adiabatic (no heat flow) if the conditions of the adjacent room are the same as the room of interest, which makes the equivalent thermal resistance of the partition wall infinite. Since the thermal resistance of the other walls is usually finite, the condition of uniform resistance will not be satisfied and thus the analysis with a convective film coefficient can not be allowed. This must be dealt with as a multi-room problem.

(3) Relative importance of heat transmission

Since no internal heat generation and small heat loss through infiltration are assumed in the present example, a large amount of heat is lost by heat transmission through the window and the external wall without insulation. Thus, the value of the surface film coefficient significantly affects the heating load and room temperature. This extreme model was adopted in order to emphasize the influence that radiative heat transfer has on the heating load and to examine the applicability of the overall heat transfer coefficient. The relative importance of heat transmission, thus the influence of the surface film coefficient, is considered to be much less in usual cases.

Furthermore, although intermittent heating was investigated in this example because the examination of the influence that the radiative heat transfer has on the sizing of HVAC systems was mainly emphasized, the difference between the exact solution based on the radiation inter-reflection calculation and that based on the overall coefficient is not so large in continuous air-conditioning.

On the other hand, no net heat transfer by radiation occurs in the case of a single room whose enclosures are of equivalent thermal properties as described in section 1-2. Since the present model room is surrounded by relatively non-uniform enclosures with a large window and (well-insulated) partition walls, the radiation heat transfer works to some extent. Thus, the adopted room can not be said an extreme case with respect to the relative importance of the radiative heat transfer.

4. INFLUENCE OF FURNITURE

Room configuration affects radiation heat exchange and consequently influences heat load analysis. Therefore, room furniture will have varying effects depending on factors related to the room configuration. Including heat capacity and radiation heat transfer, furniture affects space heat load. Here, studies were focused on the effects that furniture has on radiation heat transfer. Furthermore, studies were made into a method of approximate evaluation.

4.1 Analytical Subject

Here, analysis was made of the room without partition walls shown in Figure 1. The room was adjoined by another room of the same conditions except on sides which were in contact with outside air.

Furniture was arranged in the three configurations shown in Figure 5. Types B and C presumed a 3 [m] tall locker or book shelf arranged parallel and vertical to the window, respectively. The furniture was made of wood 1 [cm] thick on the top and lateral sides (heat capacity: 194 [J/kgK], thermal conductivity: 0.13 [W/mK]). The furniture was directly coupled to the floor on the bottom side.

4.2 Method of Numerical Analysis

(1) Computing Coefficient of Configuration for a Room With Furniture

To calculate the coefficient of configuration, walls and furniture were divided into a 11 [m^2] grid. Furniture was located so that it would overlap the wall or floor grid. Radiation emission was calculated for the entire cell, while the center of the cell was taken as the radiation receiving point.

In the case shown in Figure 6, wherein computing the coefficient of the configuration viewed from a radiation receiving point, parts in the shadow of the furniture present a problem. In such case, decisions were based on whether the center point of the cell was in the shadow or not. In short, center points 1 and 2, for example, are not in the shadow, therefore it was assumed that the entire cell was not in the shadow. On the other hand, center point 3 and 4 are in the shadow, therefore the entire cell is so considered. Judging from the computational examples shown later, the amount of error that accompanies this approximation is not considered to be all that large.

(2) Heat balance in furniture

Furniture is thermally conductive just as walls are. Therefore, heat conduction was calculated using a finite difference method. The heat balance in air temperature inside the furniture was dealt with as room heat balance. However, for the sake of simplicity, combined coefficient of heat transfer (9.3 [W/m^2K]) was used without considering radiation heat transfer.

4.3 Result and discussion

Figure 7 shows the supplied heat load over time, when initial temperature in all walls, furniture and window glass was 0 []; room temperature was 20 [] for t0 ; and outdoor air temperature was 0 [].

During the initial heating phase, assuming an initial condition of uniform temperature everywhere, the influence of net radiation heat transfer on the heating load can be neglected. If walls and furniture surfaces have similar heat capacity and thermal conductivity, the increase in peak load will be proportional to the increase in total surface area on which convective heat transfer by room air occurs.

In the initial heating phase, heat load increased approximately 6 for Type A, while approximately 11 for Types B and C, in comparison to the room without furniture. This is attributed to the difference in surface area related to heat transfer. Reason being that surface area increased approximately by 6 in Type A, and by approximately 11 in Types B and C.

In the steady-state, supplied heat load was only slightly higher than that in the room without furniture. In the room with furniture, the amount of heat radiated to windows should be minimal, thus the opposite effect is produced. This is considered as an error in computing the coefficient of configuration. A grid with smaller cells is needed, but the difference is not that large.

4.4 Approximated Evaluation of the Influence of Furniture

(1) In this section, the results from calculations which consider an extreme case for furniture will be discussed. The same perimeter-interior room as Figure 1 was investigated while varying the position of the partition wall in the room, in order to simulate the effect of furniture. Nine cases of differing room configurations, that is varying furniture (partition wall) positions, were made and the results compared to the case when there was no furniture in the perimeter-interior room.

(2) Heat-up Phase

The initial total peak loads of all the nine cases with furniture, which is the sum of the perimeter and interior room values, show little difference. But, if these total peak load values are compared to the peak load of the case without furniture, the cases with furniture are 18.4 - 18.6% more (Figure 8). This may be due to the fact that the total surface area of the furniture increases the total room surface area by 18.2%.

(3) Steady State

Adding furniture to the room will effectively block a fraction of this radiation heat transfer to the window. In other words, the total amount of heat radiated towards the window depends on the temperature of the surfaces facing this window.

If a piece of furniture stands close to the window, its surface temperature will be less than the surface temperature of the more distant walls. Therefore, due to the lower temperature of the furniture surface facing the window, less radiation will reach the window from where the furniture now stands.

Referring to Figure 9, we see how the value changes as the furniture is placed further away from the window. There is a difference of 9.8% between the two extreme left and right values.

(4) Comparison to Room with Furniture

Because the amount of surface area differs between the two setups, comparison is difficult. But, at the start of heating, the increase of heat load is 1.67 [kW] in the room with furniture (surface area increased to 24 [m^2] in Types B and C), while 2.86 [kW] in the room with partition wall (without furniture) (surface area increased to 40 [m^2]). When examined from the perspective of surface area, both setups can be considered the same. This is also the case in the steady state. Taking into account the programming troubles for the room with furniture, the approximation calculation proposed in this paper can be used effectively towards simplifying calculations.

5. CONCLUSION

During 1-day and 1-week cyclic steady-states, the conventional solution, which makes use of the standard interior film coefficient, does not yield reliable results when used to calculate the peak load of a room, and in fact, drastically overestimates it. On the other hand, the convective solution underestimates the heat load. The influence of indoor radiation heat transfer increases substantially, therefore the amount of radiation inter-change between surfaces should be accurately calculated. Since the deviation of the convective solution from the exact solution is moderate, it can be useful in determining the appropriate capacity of the air-conditioning system.

The influence of furniture on the heat load of single room was investigated. The results show that higher initial peak loads in cases with furniture are due to the additional surface area on which convective heat transfer occurs. In the steady state, furniture position determines how much radiation heat transfer can be effectively prevented from escaping to the external wall, thus reducing required heat loads.

A methodology for simplifying the use of radiation heat transfer was studied, wherein a partition wall between the interior and perimeter spaces was used to represent furniture in the room. Results from this extreme case show that it can approximate, to a good degree, the influence of furniture on heat load analysis for an intermittent heating system.

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NOMENCLATURE

c = air specific heat [J/kgK]
c_w = specific heat of wall [J/kgK]
$E_{b(s)}$ = emissive power of a black body [W/m^2]
$F_{(ss')}$ = configuration factor (shape factor)
$G_{(s)}$ = irradiation [W/m^2]
n = air exchange rate [1/s]
Q = heat input by air-conditioning system and internal heat source [W]
q = net radiative heat flux absorbed by wall [W/m^2]

r = thermal resistance of medium from wall’s inner surface to outside air [m^2 K/W]
S = inner surface area of walls [m^2]
s = position on inside surface of wall
T_w = inner surface temperature of wall []
T_r = room air temperature []
T_{ov} = outdoor temperature []
t = time [s]
V = room volume [m^3]

Greek Letters

$_c$ = convective heat transfer coefficient on interior surface [W/m^{2}K]
= air density [kg/m^3]
$_w$ = density of wall [kg/m^3]
$_{(s')}$ = emissivity of inner surface of wall
$_w$thermal conductivity of wall [W/mK]

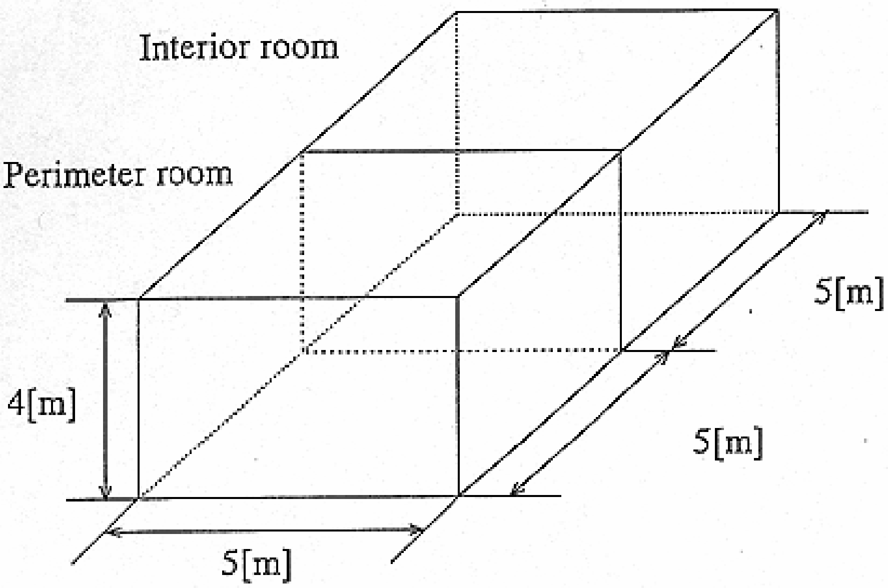

Figure 1 Schematic diagram of multi-room space

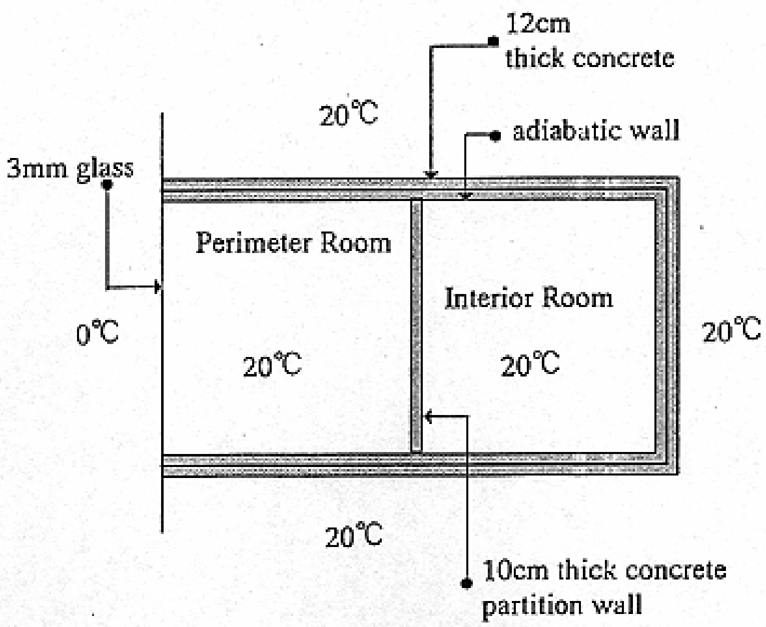

Figure 2 Thermal condition of investigated space

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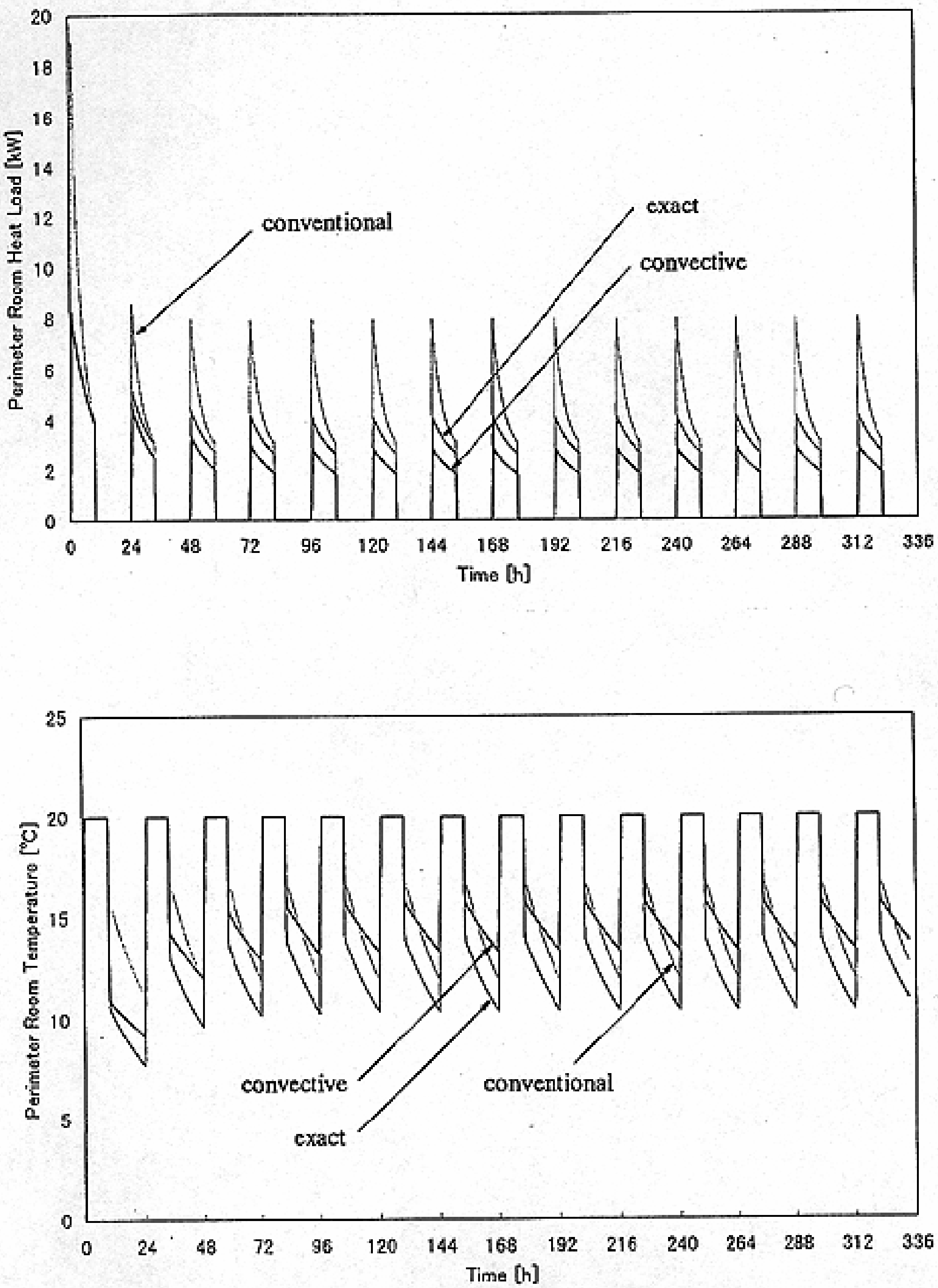

Figure 3 Perimeter room heat load and room air temperature for 14 days
(1-day cycle steady state)

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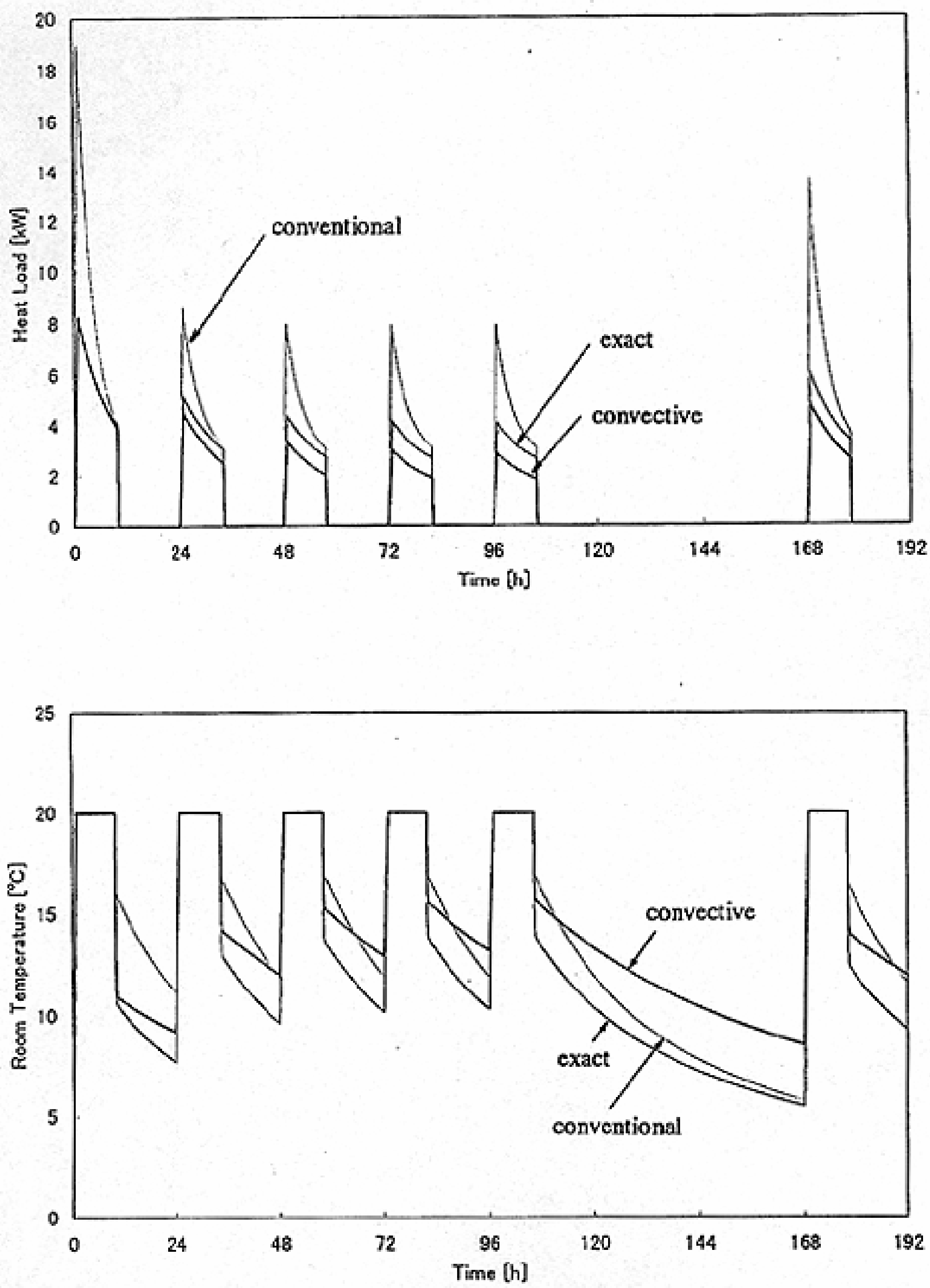

Figure 4 Perimeter room heat load and room air temperature for 8 days
(1-week cycle steady state)

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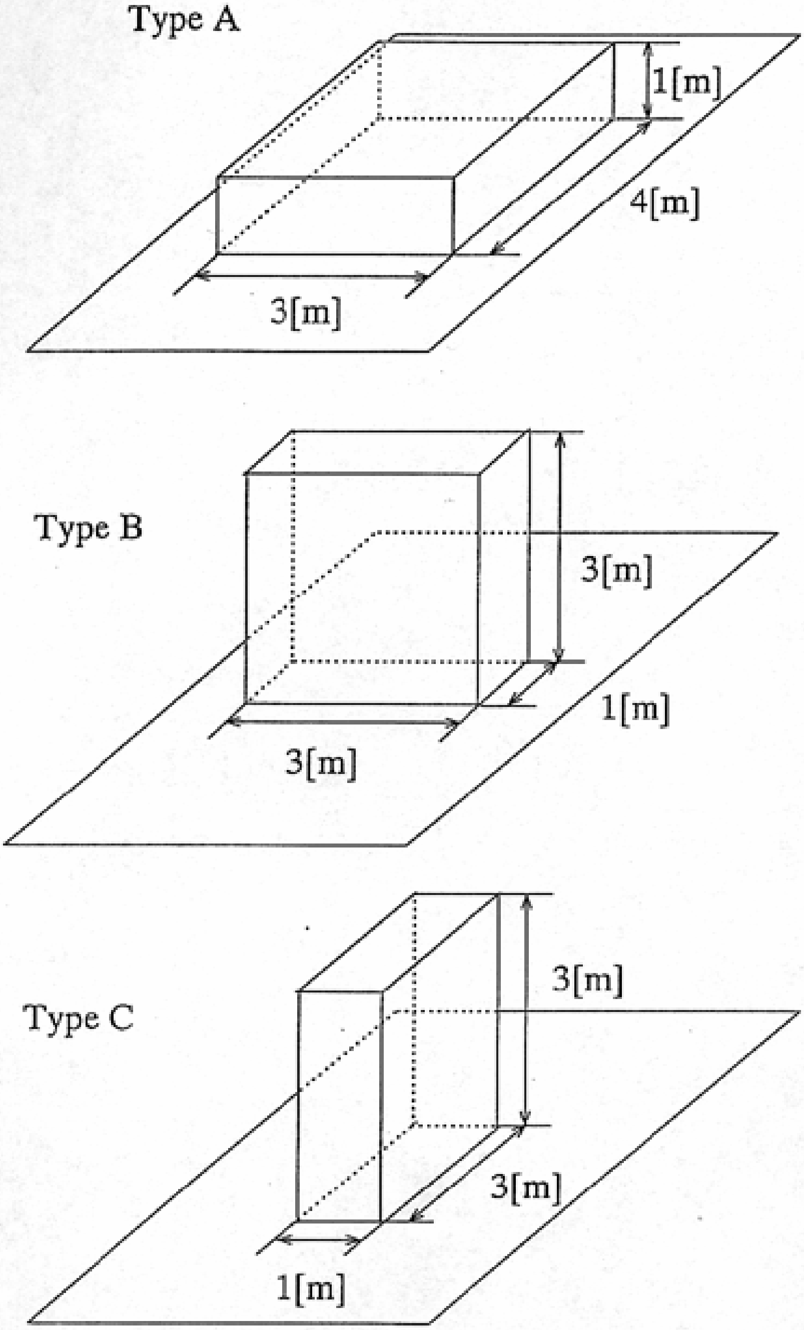

Figure 5 Furniture configuration

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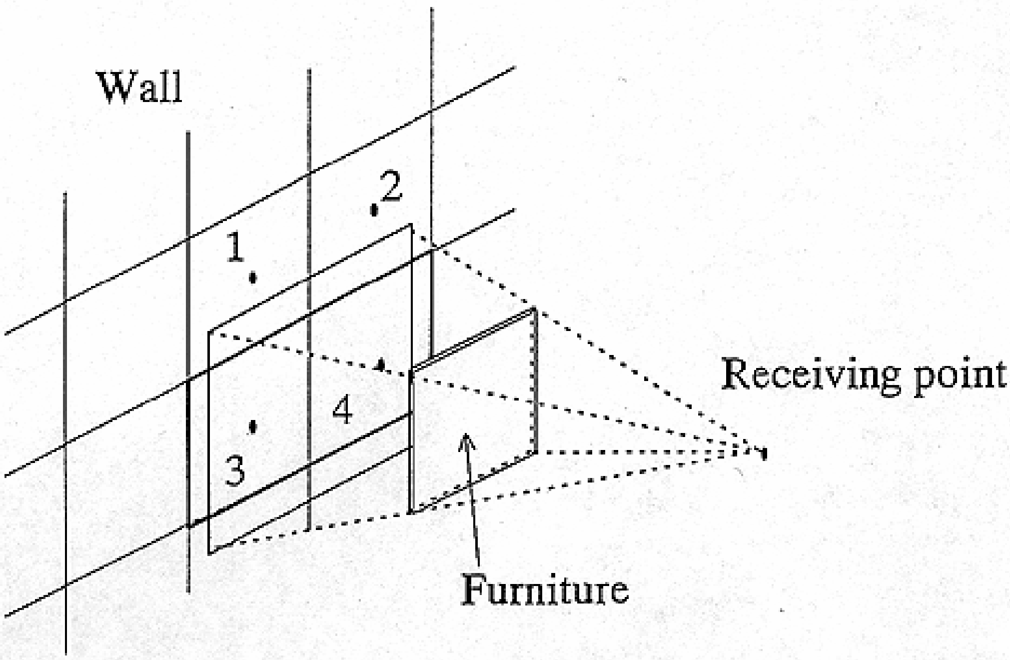

Figure 6 Concept for computing coefficient of configuration for room with furniture

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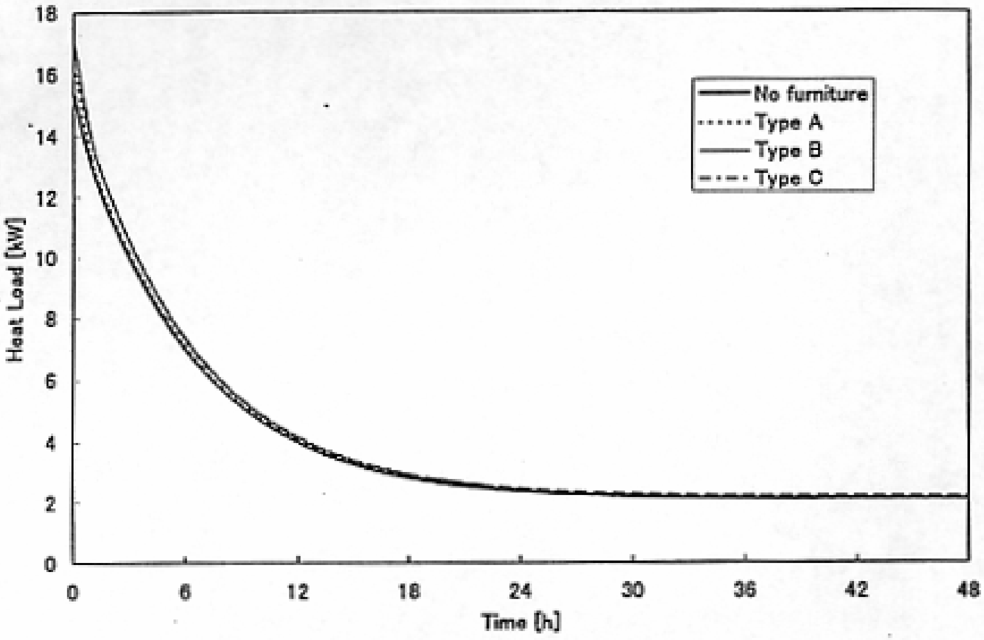

Figure 7 Comparative graph of peak load between case with and without furniture

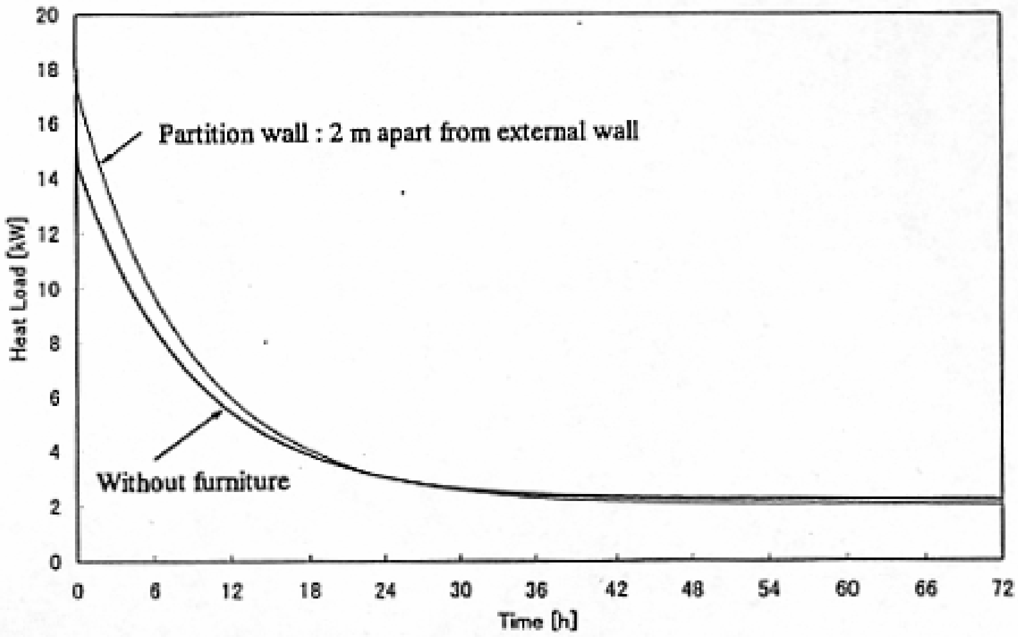

Figure 8 Comparative graph of peak load between case with and without furniture by approximate method

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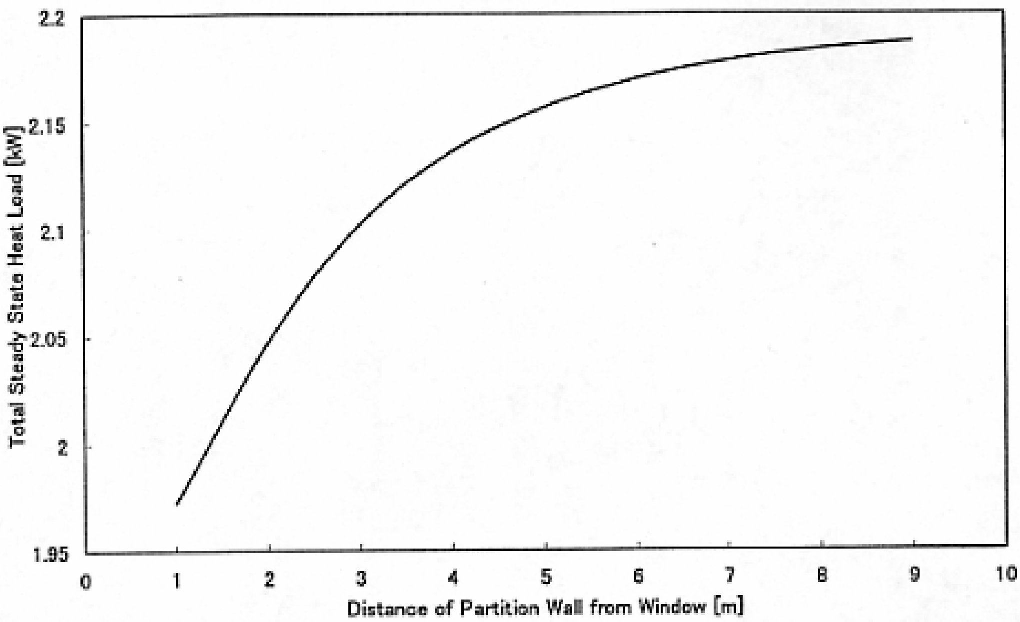

Figure 9 Steady state heat loads of cases with furniture

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