RESEARCH ON CALCULATION METHOD OF THERMAL DESIGN LOAD IN RADIANT HEATING AND COOLING SYSTEMS

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ABSTRACT

A new calculation method is proposed for designing space radiant heating/cooling systems by improving conventional methods of steady and unsteady heat transfer calculation theory which has usually been used for designing of buried pipe floor heating systems. On the new method to calculate room thermal load in radiant heating/cooling system, an operative temperature surrounding occupants are used as system control parameter instead of a space air temperature. Theoretical algorithms and calculation examples are described.

INTRODUCTION

In recent years, the significance of radiant heating and cooling system has been recognized again, and it is now widely adopted in buildings for various purposes. In particular, it is considered as an indispensable system for atrium or stairwell. Although the radiant heating

Table 1 Conventional basic formulae to calculate heat transfer rate (An example of intermediate stage)

A. Basic formula of fin heat transfer rate of flat finned coil

Heat transfer rate Q_F from flat finned coil is given as:

$$Q_F = \alpha_F \bullet \eta_F(t_P - t_A) \quad ....(1)$$

where η_F represents fin efficiency, and it is determined by the ratio of $2\alpha_F$ (expressing heat transfer property from fin to the air) to $\lambda_F D_F$ (expressing thermal conductivity of fin) and by fin length 0.5 (W - Z).

$$\eta_F = \tan h(Z)/Z \quad ....(2)$$

$$Z = 0.5\,(W - D)\{2\alpha_F/(\eta_F \bullet D_F)\}^{1/2} \quad ....(3)$$

B. Conventional method to calculate heat transfer rate in floor heating

(1) Kollmar-Liese method

In the floor, a virtual fin is assumed, which has thickness equal to pipe diameter D. As heat transfer coefficient of the virtual fin, a sum of upper side and lower side thermal conductance values C_{FU} and C_{FD} from upper or lower end of the pipe to the air is used. That is, using the value of Z obtained from the following equation, fin efficiency η_F is obtained from the equation (2).

$$Z = 0.5\,(W - D)\{(C_{FU} + C_{FD})/(\lambda_{FF} \bullet D)\}^{1/2} \quad .....(4)$$

The upward heat transfer rate Q_U is expressed as:

$$Q_U = \{C_{FU} \bullet D(t_P - t_A) + C_{FU}(W - D)\eta_F(t_P - t_A)\}/W$$
$$= C_{FU} \bullet \eta(t_P - t_A) \quad .....(5)$$

where η is overall efficiency of fin - pipe, and it is given as:

$$\eta = D/W + \eta_F(W - D)/W \quad .....(6)$$

The downward heat transfer rate Q_D is also given by the following
equation:

$$Q_D = C_{FD} \bullet \eta(t_P - t_A) \quad .....(7)$$

(2) Method by Kilkis et al.

Fin efficiency is applied only on the upward heat transfer rate. Floor material from piping center depth to floor surface is considered as a virtual fin. To match this, the upper floor surface is regarded as a heat transfer surface. Floor surface temperature immediately above the pipe is given by $t_{F,MAX}$ and the upward heat transfer rate Q_U is expressed by the following equation:

$$Q_U = \eta \bullet \alpha_U(t_{F,MAX} - t_A) \quad .....(8)$$

η is derived from the equation (6). Using the value of Z obtained from the following equation, fin efficiency η_F is obtained from the equation (2).

$$Z = 0.5(W - D)\{(\alpha_U/\Sigma\,\lambda_i \bullet D_i)\}^{1/2} \quad .....(9)$$

(i: Floor material above the piping center depth)

Using thermal conductance C_U from the upper end of pipe to floor surface, it is assumed that the following equation is established:

$$Q_U = C_U(t_P - t_{F,MAX}) \quad .....(10)$$

From the equations (8) and (10),

$$Q_U = C_{FU}'(t_P - t_A) \quad .....(11)$$

$$C_{FU}' = \{1/(\eta \bullet \alpha_U) + 1/C_U\}^{-1} \quad .....(12)$$

It is assumed that intra-floor horizontal temperature at the lower end of the pipe is equal to pipe surface temperature, and the downward heat transfer rate Q_D is obtained by the following equation:

$$Q_D = C_{FD}(t_P - t_A) \quad .....(13)$$

[Symbols]
W and D: Pitch and diameter of pipe
t_P and t_A: Pipe surface temperature and ambient air temperature
α_F: Heat transfer coefficient of fin surface
λ_F: Thermal conductivity of fin
D_F: Thickness of fin
α_U: Heat transfer coefficient of floor upper surface
λ_{FF}: Thermal conductivity of virtual fin
λ_i: Thermal conductivity of floor material i
D_i: Thickness of floor material i
[Units]
Length: m
Temperature: ^oC
Heat rate: W/m^2
Heat transfer coefficient and thermal conductance: W/m^2K
Thermal conductivity: W/mK

and cooling system has been known and used for long years, there are a number of controversial problems in its design theory. This paper can be roughly divided into the following three parts:

1) As a simple method to calculate steady heat transfer rate of hot-water floor heating, there is a method to substitute the floor with flat finned coil. In this paper, the features and accuracy of Kollmar-Liese method and Kilkis method are reviewed, and a modified method with higher calculation accuracy is proposed.

2) To deal with unsteady characteristics of floor heating, a method to use uniform heat generating surface on the intra-floor buried pipe depth is widely used, and this has made it possible to treat thermal equilibrium of the entire room including floor in one-dimensional manner. However, in this paper, it is positively pointed out that the substitution with uniform heat generating surface leads to underestimation of actual delay, and a method to approximate using polynomial with fixed roots is proposed.

3) The design load calculation of radiant heating and cooling system should be thermal load calculation with operative temperature to human body as the set point, while calculation is made mostly by substituting it with room air temperature. In this paper, operative temperature for calculating wall conduction heat load and operative temperature for panel heat transfer calculation are defined, and a thermal load calculation method is proposed, by which it is possible to design a panel surface satisfying the design operative temperature for thermal comfort.

1. APPROXIMATION OF STEADY HEAT TRANSFER RATE OF CONCRETE SLAB

Table 1 summarizes conventional methods to substitute floor with flat finned coil, and Fig. 1 shows fin substitution method in each of the calculation methods. Fin efficiency of the flat finned coil can be expressed by the equations (2) and (3) shown in Table 1. The value of fin efficiency is determined by the ratio of $\lambda_F \cdot D_F$ indicating thermal conductivity of fin to $2\alpha_F$ indicating heat transfer property to the air as well as by fin length $0.5\,(W - D)$. In Kollmar-Liese method, a fin having thickness equal to pipe diameter is assumed in the floor in order to apply fin efficiency to floor heating, and thermal conductance from upper and lower surfaces of this virtual fin to the air is regarded heat transfer coef-

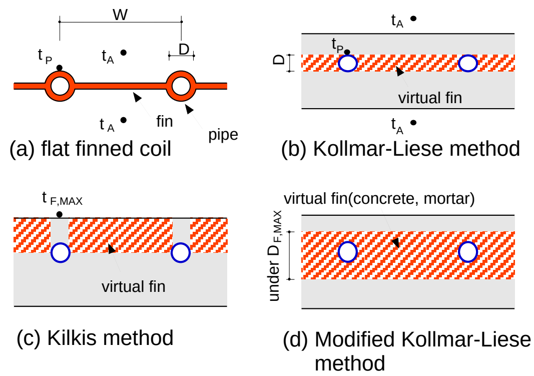


Fig.1 Fin substitution method in each calculation method

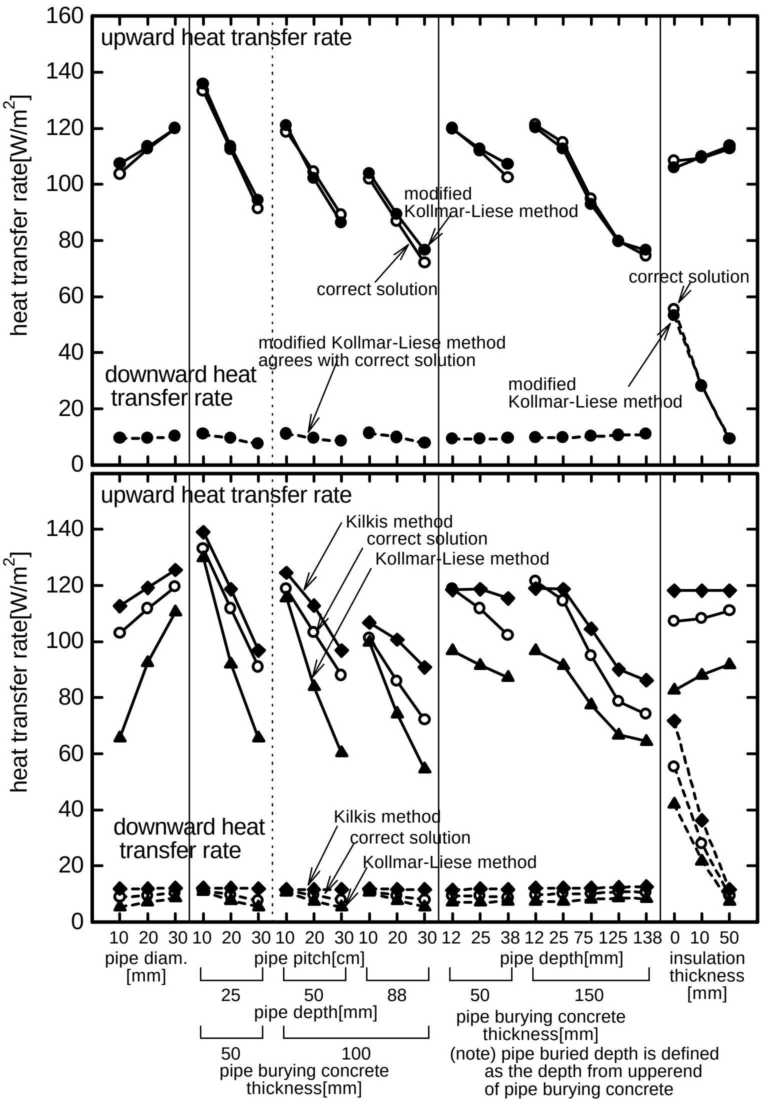


Fig.2 Evaluation of accuracy of modified Kollmar-Liese method and conventional calculation method

Table 2 Newly proposed heat transfer calculation method of floor heating (Modified Kollmar-Liese method)

Concrete and mortar layers around the piping are regarded as a virtual fin. However, upper limit of virtual fin thickness is defined as $D_{F,MAX}$. $D_{F,MAX}$ is obtained by the following equation from the pipe pitch W:

$$D_{F,MAX} = 0.045 + 0.1(W - 0.2) \quad .....(1)$$

For heat transfer coefficient of the virtual fin, the value of (C_{FU} + C_{FD}) is used as in Kollmar-Liese method.

The value of Z to obtain fin efficiency η_F is given as:

$$Z = 0.5(W - D)\{(C_{FU} + C_{FD})/ (\Sigma \lambda_i \bullet D_i)\}^{1/2} \quad .....(2)$$

(i: Floor material in the virtual fin)

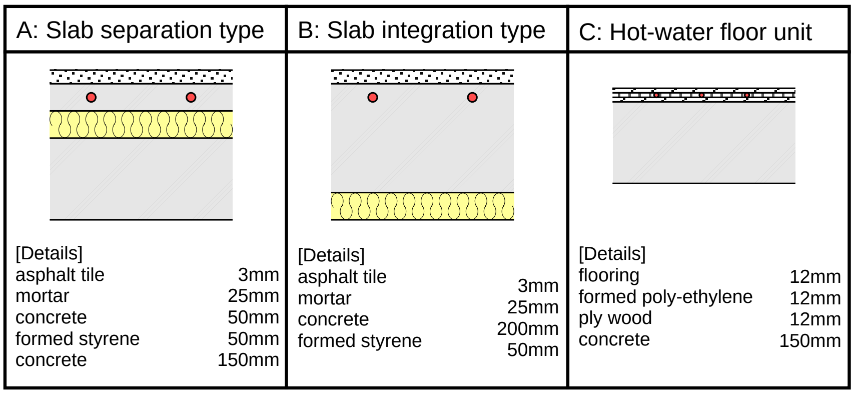
Upward and downward heat transfer rate are obtained from (2) in Table 1, and the equations (5) - (7).

ficient of upper and lower surfaces of the fin. In Kilkis method, it is assumed that floor material from center depth of the pipe to floor surface has fin effect with respect to upward heat transfer, and fin efficiency is obtained from floor specification of this portion. The values of heat transfer rate according to Kollmar-Liese method and Kilkis method were compared with 2-dimensional finite element method as an accurate calculation method. The results of this comparison is shown in the lower half of Fig. 2. In Kilkis method, in case the pipe is at shallow position in concrete layer, error is not very big. But the deeper the depth is, the more the error is increased. In Kollmar-Liese method, the virtual fin is thin and the fin effect of concrete is not sufficiently evaluated. Thus, the more wide the pipe pitch is to piping system, the more the error is increased. In Kilkis method, it appears that thickness of virtual fin is overestimated.

When substituting with flat finned coil, in addition to the problem of virtual fin thickness to induce thermal conduction in horizontal direction, there are the following problems: In case of flat finned coil, vertical temperature distribution within fin can be regarded as even distribution, while, in case of virtual fin in concrete, vertical temperature distribution within fin is not negligible when it is thick, and it is necessary to consider the problems, e.g. how far the influence of heat resistance in vertical direction of virtual fin can be estimated, at which position the average horizontal temperature in floor should be set to estimate fin efficiency, and whether it is necessary or not to substitute the pipe in addition to the fin. Calculation method was designed according to several types of fin substitution methods, and a modified method derived from Kollmar-Liese method is proposed as shown in Table 2 as a recommendable approach. This method has been selected not only because it provides high accuracy but also because this has less changes compared with the conventional methods and the equations are less complicated. In this method, the thickness less than $D_{F,MAX}$ of a portion of concrete where the pipe is buried is treated as a virtual fin, and the value of $D_{F,MAX}$ conductivity of virtual fin, this method is the same as Kollmar-Liese method. The evaluation results of the accuracy of the modified Kollmar-Liese method are shown in the upper half of Fig. 2. heat transfer rate can be estimated with error of less than 5%.

2. APPROXIMATION OF UNSTEADY HEAT TRANSFER RATE OF CONCRETE SLAB

For unsteady heat transfer of radiant panel surface, heat transfer response based on 2-dimensional unsteady heat conduction calculation within floor was analyzed, and it was attempted to present an approximate expression by taking more influential factors into consideration. Step heat transfer response from upper and lower surfaces of floor to room for excitation of surface temperature of the pipe was obtained from 2-dimensional calculation, and this was turned to dimensionless by dividing with steady heat transfer rate on upper side or lower side, and this was used as an index of unsteady characteristics. Here, slab separation type, slab integration type and hot-water floor unit in Fig. 3 are called Types A, B and C respectively. In Types A and B where pipe is buried in concrete, the influence of major fac-



[Note] In case of slab separation type and slab integration type: pipe diameter 25mm, pipe pitch 200mm, burying depth 25mm from concrete upper end, supply water temperature 40°C.
In case of hot-water floor unit: pipe diameter 5mm, pipe pitch 80mm, burying depth 2.5 mm from upper end of foamed polyethylene, supply water temperature 65°C.

Fig.3 Standard conditions of major floor types tors on 2-dimensional heat transfer response was analyzed, and comparison was made at the same time with one-dimensional heat transfer response where it is assumed that there is a uniform heat generating surface at intra-floor pipe burying level. In Type C of hot-water floor unit, when heat transfer response of the standard model was evaluated, upward heat transfer reached steady level within one hour in both one-dimensional and 2-dimensional calculations, while there was no substantial difference between one-dimensional and 2-dimensional calculation values in the downward heat transfer, and it was judged that the substitution with conventional one-dimensional calculation would be effective. Fig. 4 shows upward and downward heat transfer response values in the standard models of Types A and B. In Types A and B, floor specification above the piping is the same. Thus, the values of one-dimensional upward heat transfer responses are the same. However, the upward heat transfer response in 2-dimensional calculation is delayed in Type B (thickness of concrete with buried pipe: 150 mm) compared with Type A (thickness of concrete with buried pipe: 50 mm). These two cases are delayed considerably more compared with one-dimensional heat transfer response. In the downward heat transfer response, the difference between one-dimensional calculation value and 2-dimensional calculation value is lower when compared with the upward heat transfer response. In particular, in Type A where pipe burying concrete layer is installed separately from slab, the difference is very small. It is assumed that, if floor is insulated well in order to suppress the downward heat radiation, there would not be much influence even when one-dimensional heat transfer response is substituted with the downward heat transfer response. In this respect, the influence of various factors will be analyzed below on the upward heat transfer response. Fig. 5 represents an evaluation of the pipe buried depth and the delay of heat transfer response per thickness of pipe burying concrete layer. The difference of the initial heat transfer response due to the pipe buried depth in case of 2-dimensional calculation tends to be smaller than in one-dimensional calculation. In case the concrete thickness is 50 mm, there is no substantial change in the heat transfer response due to the pipe buried depth.

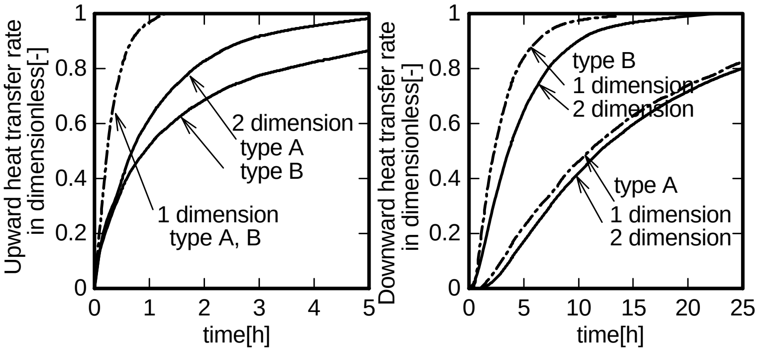
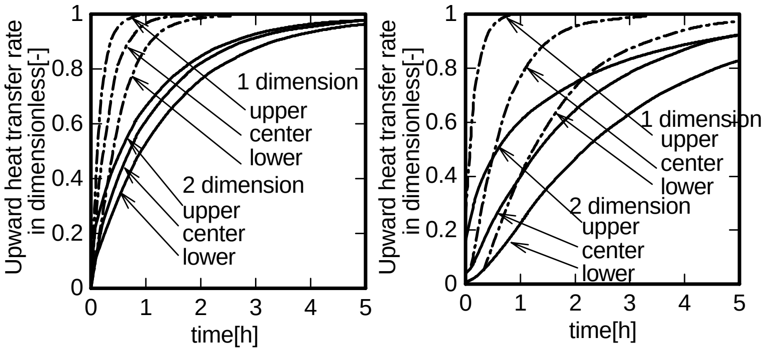


Fig.4 Upward and downward heat transfer rate of standard model

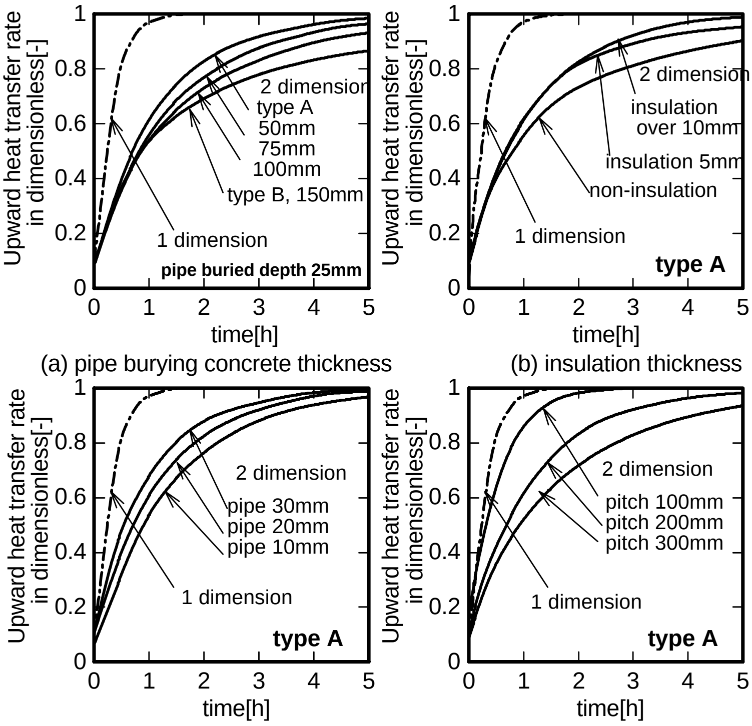


(a) concrete depth 50mm (b) concrete depth 100mm

[note] Pipe depth means position of the pipe buried in concrete layer. "upper" and "lower" mean that the distance from upper end or lower end of burying concrete is 12mm respectively.

Fig.5 Pipe buried depth and upward heat transfer rate (Type A)

If the specification of the floor above the pipe is the same, there is no change in one-dimensional upward heat transfer response, while, in actual response, there may be influences of lower side floor specification, pipe diameter, pipe pitch, etc. This was evaluated mostly on Type A, and the results are shown in Fig. 6. (a) In the drawing to show comparison of thickness of the pipe burying concrete layer, the cases with concrete thickness of 75 and 100 mm are added to the cases of Types A and B in Fig. 4. (b) From the drawing to show comparison of heat insulation thickness, it is evident that the lower slab can exclude the phenomenon to delay the upward heat transfer rate. There may be some differences in heat transfer response due to pipe diameter, but the difference due to pipe pitch is much higher. In case the pitch is set to 100 mm, the difference from one-dimensional heat transfer response would be considerably reduced. In addition, the difference of the upward heat transfer response due to floor finishing material was assessed, but there was no substantial difference.



(a) pipe burying concrete thickness (b) insulation thickness

(c) pipe diameter (d) pipe pitch

Fig.6 Relationship between major factors and upward heat transfer rate

The influence of major factors on heat transfer response was analyzed, and it was found that the influence was high on the pipe burying concrete thickness, pipe buried depth and pipe pitch. In this respect, to select typical heat transfer response, Categories I, II and II are prepared as shown in Table 3 from the pipe burying concrete thickness, and it is assumed that these Categories represent the cases of 50 mm, 100 mm and 150 mm respectively. It is assumed that Categories I and II correspond to Type A of slab separation type, and Category III corresponds to Type B of slab integration type. Further, it is assumed that, depending on the pipe buried depth, Category I is represented by one of central burying type in the concrete layer, three types of upper, central and lower burying are represented by Category II, and only upper burying type is represented by Category III. For each of these cases, typical three types of 100 mm, 200 mm and 300 mm of pipe pitch are picked up. If an approximate expression is prepared for the typical upward heat radiation response, it is possible to readily incorporate it into one-dimensional calculation of the entire room without performing 2-dimensional calculation on floor sector. An approximate expression r(t) of heat transfer response expressed in dimensionless manner was given as described below using 5 fixed roots (5 roots given in advance) of α_1 to α_5.

$$r(t) = 1 + \sum_{k=1}^{5} A_k e^{-\alpha_k t} \qquad \text{............(1)}$$

where t represents time [h]. As the values of α_1 to α_5, 0.10, 0.27, 0.71, 1.88 and 5.0 were used respectively which are known to suit approximation of heat absorption response in the entire room. These 5 roots can cover from a component with less response delay to a component with more delay and are considered as suitable for approximation of floor heat transfer response. Also, if heat transfer response of floor heating is approximated by these roots, the results can be directly incorporated in indoor cal-

Table 3 Approximation coefficient of dimensionless step heat transfer response r(t) of floor

$$r(t) = 1 + \sum_{k=1}^{5} A_k e^{-\alpha_k t}$$

Pipe burying concrete thickness	Pipe buried position	Pipe pitch	a_1	a_2	a_3	a_4	a_5
			0.10	0.27	0.71	1.88	5.0
			A_1	A_2	A_3	A_4	A_5
I: 50mm	optional	100mm	—	– 0.022	0.058	– 1.186	0.150
		200mm	– 0.027	0.084	– 0.874	– 0.169	– 0.014
		300mm	0.053	– 0.402	– 0.450	– 0.201	—
II: 100mm	upper	100mm	—	– 0.006	– 0.195	– 0.812	0.013
		200mm	0.067	– 0.368	– 0.523	– 0.077	– 0.099
		300mm	0.044	– 0.752	– 0.012	– 0.280	—
	center	100mm	—	0.069	– 0.709	– 0.790	0.430
		200mm	0.087	– 0.410	– 0.873	0.196	—
		300mm	0.051	– 0.786	– 0.333	0.068	—
	lower	100mm	– 0.023	0.030	– 1.607	0.600	—
		200mm	0.102	– 0.839	– 0.649	0.386	—
		300mm	—	– 1.042	– 0.146	0.188	—
III: 150mm	upper	100mm	—	– 0.102	– 0.037	– 0.885	0.024
		200mm	– 0.010	– 0.474	– 0.090	– 0.488	0.062
		300mm	– 0.114	– 0.645	0.104	– 0.345	—

[Note] Approximate expression was prepared in the case with the conditions changed as shown in Table based on slab separation type for I and II, and slab integration type for III. Pipe buried position was set to: Center for I; 25 mm from upper end of the pipe burying concrete and the center for II; 25 mm from upper end of concrete for III.

culation using fixed 5 roots. Coefficients A_1 - A_5 are determined in such manner that correct solution agrees with approximation solution at 5 points on the curve. First, the coefficients were determined, assuming that the condition of compatibility exists when dimensionless heat transfer rate was 0, 0.4, 0.7, 0.9 and 0.95. Point of compatibility was changed when necessary, or the curve was made smooth by reducing the number of roots. Table 3 shows coefficients of approximate expression of typical heat transfer response. Approximation error after 10 minutes of step change is approximately within 0.01 in dimensionless heat transfer rate.

3. CALCULATION METHOD FOR THERMAL DESIGN LOAD

When designing radiant heating and cooling system, it is the best approach to use design calculation method, by which the operative temperature is given as the condition for indoor designing and the effect of radiant heat to human body from radiation panel can be evaluated. In the conventional design calculation, it has been difficult to calculate, for example: to which value the panel temperature and indoor air temperature should be set to satisfy the design operative temperature when position and size of the panel are assumed, or which temperature value the peripheral wall could take. In addition to the design of the radiant heating and cooling system, in the conventional air-conditioning design, there was no distinction between indoor air temperature and operative temperature or between indoor air temperature and indoor surface temperature in wall

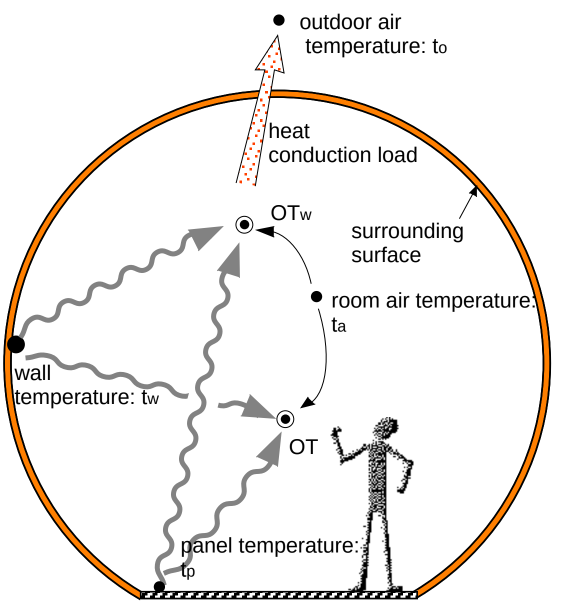


Fig.7 Simplified space comprising radiant panel surface and wall surface

Table 4 Basic formulae for thermal design load calculation method of radiant heating and cooling system

- Basic formula relating to operative temperature OT [°C] and wall conduction thermal load q_W [W] (Simple room comprising outer wall surface and panel surface)

$$OT = x_{rh}\{F_{h,P} \bullet t_P + \{(1-F_{h,P})t_W\} + (1-x_{rh})t_a \quad .....(1)$$

where $x_{rh} = \alpha_{rh}/\alpha_{Th}$ .....(2)

$$q_W = \alpha_{rW} \bullet A_W[\{F_{W,P} \bullet t_P + (1-F_{W,P})t_W\} - t_W] + \alpha_{cW} \bullet A_W(t_a - t_W) \quad ...(3)$$

$$= \alpha_{TW} \bullet A_W(OT_W - t_W) \quad .....(4)$$

where OT_W is operative temperature for calculating wall conduction thermal load, and it is given as:

$$OT_W = x\{F_{w,p} \bullet t_P + (1-F_{W,P})t_W\} + (1-x_{rW})t_a \quad .....(5)$$

where $x_r = \alpha_{rW}/\alpha_{TW}$ .....(6)

On the other hand,

$$q_W = K_W \bullet A_W(OT_W - t_O) \quad .....(7)$$

- Room air temperature to satisfy design operative temperature OT

If it is simply assumed that $x_r = x_{rh}$, the following equation can be obtained from the equations (1) - (7):

$$t_a = OT/(1-x_r) - x_r\{F_{h,P} \bullet t_P + (1-F_{h,P})t_W\}/(1-x_r) \quad .....(8)$$

where

$$t_W = a_W \bullet t_O + (1-a_W)OT_W \quad .....(9)$$

$$OT_W = OT - \Delta OT \quad .....(10)$$

$$\Delta OT = [\, x_r \bullet \Delta F_P[t_P - \{a_W \bullet t_O + (1-a_W)OT\}]\,] / \{1 - x_r \bullet \Delta F_P(1 - a_W)\} \quad .....(11)$$

However, $\Delta F_P = F_{h,P} - F_{W,P}$ .....(12)

$a_W = K_W/\alpha_{TW}$ .....(13)

- Substitution method to one wall surface in case the room comprises a multiple of wall surfaces

1) α_{rW} and α_{cW} have the same value regardless of wall surface.
2) As shape factor between two furfaces, area ratio is used.
3) For inner wall, adjacent room air temperature difference coefficient is given.

By the introduction of hypothesis and technique for simplification as given above, a multiple of wall surfaces can be substituted with one wall surface when each value is obtained by the following equations for outer wall i and inner wall j:

$$A_W = \Sigma_i A_{Wi} + \Sigma_j A_{Wj} \quad .....(14)$$

$$F_{W,P} = A_P/A_W \quad .....(15)$$

$$K_W = (\Sigma_i K_{Wi} \bullet A_{Wi} + \Sigma_j f_j \bullet K_{Wj} \bullet A_{Wj})/A_W \quad .....(16)$$

- Panel heat transfer rate q_P [W] and air- conditioning heat rate q_{AC} [W]

$$q_P = \alpha_{rP} \bullet A_P(t_P - t_W) + \alpha_{CP} \bullet A_P(t_P - t_a) \quad .....(17)$$

$$= \alpha_{TP} \bullet A_P(t_P - OT_P) \quad .....(18)$$

where OT_P is operative temperature for calculating panel thermal load and it is given as:

$$OT_P = x_{rP} \bullet t_W + (1-x_{rP})t_a \quad .....(19)$$

where $x_{rP} = \alpha_{rP}/\alpha_{TP}$ .....(20)

$$q_{AC} = q_W + q_{inf} - q_P \quad .....(21)$$

where $q_{inf} = C_P \bullet \rho \bullet Q_{inf}(t_a - t_O)$ .....(22)

[Symbols]

t_a, t_P, t_W and t_O: Room air temperature, panel surface temperature, wall surface temperature and outdoor air temperature [°C]

$F_{h,P}$ and $F_{w,P}$: Shape factor between human body and panel or between wall and panel [-]

α_{rh} and α_{Th}: Radiative and total heat transfer coefficient of human body surface [W/(m²•K)]

α_{rW}, α_{cW}, and α_{TW}: Radiative, convective and total heat transfer coefficient of wall surface [W/(m²•K)]

K_W, A_W and A_P: Overall heat transfer coefficient of wall [W/(m²•K)], and surface area of wall and panel [m²]

f: Temperature difference coefficient from adjacent room (= air temperature difference from adjacent room/air temperature difference between outdoor and indoor) [-]

Suffix i: Outer wall

Suffix j: Inner wall

α_{rP}, α_{CP}, and α_{TP}: Radiative , convective and total heat transfer coefficient of panel [W/(m²•K)]

q_{inf}: Infiltration load [W]

C_P and ρ Specific heat of the air [J/gK] and density [g/l]

Q_{inf}: Infiltration rate [l/sec]

Table 5 Relational expression of panel surface temperature and each temperature value

- Operative temperature OT_W for calculating wall thermal load

$$OT_W = A_{OTW} \bullet t_P + B_{OTW} \quad .....(1)$$

$$A_{OTW} = -x_r \bullet \Delta F_P / \{1 - x_r \bullet \Delta F_P (1-a_W)\} \quad .....(2)$$

$$B_{OTW} = OT + x_r \bullet \Delta F_P \{a_W \bullet t_O + (1-a_W) OT\} / \{1 - x_r \bullet \Delta F_P (1-a_W)\} \quad .....(3)$$

- Room air temperature t_a

$$t_a = A_{ta} \bullet t_P + B_{ta} \quad .....(4)$$

$$A_{ta} = -x_r \{F_{h,P} + (1-F_{h,P})(1-a_W) A_{OTW}\} / (1-x_r) \quad .....(5)$$

$$B_{ta} = OT/(1-x_r) - x_r (1-F_{h,P}) \{a_W \bullet t_O + (1-a_W) B_{OTW}\} / (1-x_r) \quad .....(6)$$

- Operative temperature OT_P for calculating panel thermal load

$$OT_P = A_{OTP} \bullet t_P + B_{OTP} \quad .....(7)$$

$$A_{OTP} = x_{rP} (1-a_W) A_{OTW} + (1-xr_P) A_{ta} \quad .....(8)$$

$$B_{OTP} = x_{rP} \{a_W \bullet t_O + (1-a_W) B_{OTW}\} + (1-x_{rP}) B_{ta} \quad .....(9)$$

conduction load calculation. In order to design by evaluating radiation effect, which is an advantageous feature of the radiant heating and cooling system, it is necessary to use thermal load calculation method, in which these temperature values are clearly distinguished from each other.

The calculation as described in this paper is characterized in that operative temperature OT_W for wall thermal load calculation and operative temperature OT_P for panel thermal load calculation are introduced as indoor temperature values. Fig. 7 is a schematic drawing of a simple room, which comprises an outer wall surface and a panel surface. In case the air-conditioning system and the radiant heating and cooling system are simultaneously used, Table 4 summarizes equations to obtain shape factor between the panel and a man, room air temperature, OT_W, OT_P and panel heating rate, and air-conditioning rate, when panel area and temperature are assumed. The influences of radiation heat such as sunlight, artificial lighting and influences of furniture are not included. The relationship of each of room air temperature, OT_W and OT_P to panel temperature is expressed by linear equation, and these are obtained from the equations of Table 4. The results are summarized in Table 5.

Table 6 Conditions for calculation

(Model 1: Living room; total floor heating)
- Floor panel area: 20 m^2
- Infiltration: 0.5 c/h
- Shape factor between man and panel or wall :
  - $F_{h,P}$ = 0.40 (sitting position)
  - $F_{w,P}$ = 0.28

-A: Area of each surface [m^2],
U: Overall heat transfer coefficient of wall[W/m^{2}K],
f: Adjacent room temperature difference coefficient:
  - Window: (A=7.4, U=6.5, -)
  - Outer wall: (A=17.5, U=0.9, -)
  - Inner wall 2: (A=6.4, U=2.7, f=0)
  - Ceiling: (A=20, U=1.7, f=0.7)

(Model 2: Office room; total floor heating)
- Outer wall is sufficiently long in longitudinal direction, and calculation is made for a space of 10 m in depth and 1 m in outer wall length.
- Floor panel area: 10 m^2
- Infiltration: 0.2 c/h
- Shape factor between man and panel or wall :
  - $F_{h,P}$ = 0.46 (sitting position)
  - $F_{w,P}$ = 0.66
- A, U, f
  - Window: (A=1.8, U=6.5, -)
  - Outer wall: (A=0.8, U=1.1, -)
  - Inner wall: (A=2.6, U=2.4, f=0.5)
  - Ceiling: (A=10, U=2.1, f=0.15)

(Model 3: Entrance lobby of atrium; total floor heating)
- Glass outer wall is sufficiently long in longitudinal direction, and calculation is made for a space of 10 m in depth and 1 m in outer wall length.
- Floor panel area: 10 m^2
- Infiltration: 1.0 c/h
- Shape factor to see panel:
  - $F_{h,P}$ = 0.44 (standing position)
  - $F_{w,P}$ = 0.33

-A, U, f
  - Glass outer wall and ceiling: (A=20, U=6.5, -)
  - Inner wall: (A=10, U=4, f=0)

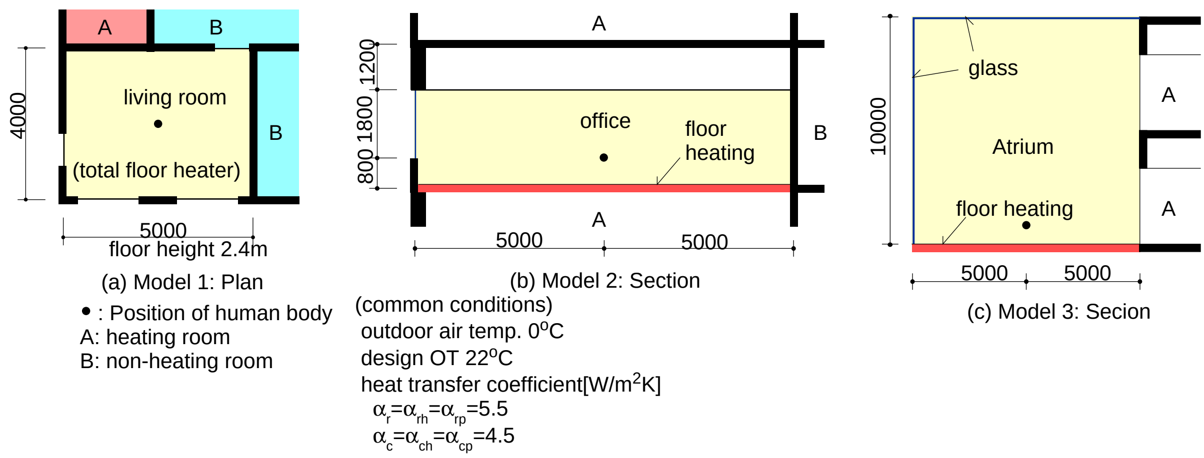
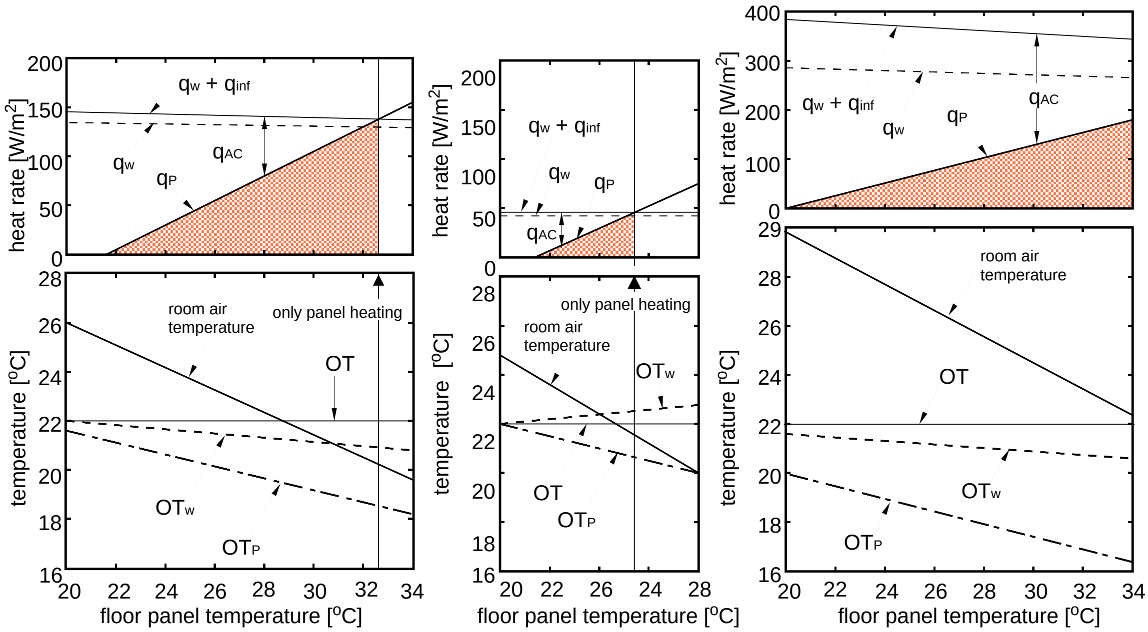


Fig.8 Calculation model



(a) Model 1 (living room) (b) Model 2 (office room) (c) Model 3 (entrance lobby in Atrium)

Fig.9 Relationship between floor panel temperature and each of room air temperature, various operative temperatures and heat rate

Based on the relationship as described above, characteristics of various types of radiant heating and cooling system can be understood, and case study can be readily performed. Here, a case study of the latter case is given. Three building models were considered as shown in Fig. 8 and Table 6, and the relationship of each of the temperature values to floor panel temperature and the relationship of load sharing in air-conditioning and panel heating were calculated on trial basis. The results are as shown in Fig. 9. The higher the dependency on radiant heating is, the more OT_p is decreased to lower than OT, and the calculation of panel heating rate distinguishing OT_p from OT becomes important. As it is remarkable in case of Model 3, the higher the dependency on radiant heating is, the more the room thermal load is decreased.

CONCLUSIONS

In this paper, it is proposed as follows: to simplify the calculation of heat transfer characteristics around the buried pipe in a radiant heating and cooling system, to give approximate expression of unsteady heat transfer using fixed 5 roots, and to provide a design thermal load calculation method using operative temperature as objective function.

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