A STUDY ON THE THERMAL PERFORMANCE SIMULATION TO EVALUATE THE PREFABRICATED RADIANT FLOOR HEATING PANELS

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ABSTRACT

Computer models used for analyzing heat transfer have been developed and computerized for the precise thermal analysis of two typical prefabricated radiant floor heating panels. The developed computer program was validated by comparing the computer simulation results against the scaled model test results. Computer simulations, for the sensitivity analyses of the various design parameters and control water temperature shows the thermal performance with the variations of these parameters and the relationship among these parameters.

INTRODUCTION

Radiant floor heating(Ondol in Korean) is a domestic heating method which has been used in Korea for more than 1,500 years. In ancient years, the traditional radiant floor heating system(Ondol) was developed to form a furnace, heated air passages, and a chimney(from A.D. 100 to 1950's). Recently, the fuel was changed from fire wood to anthracite coal. Currently the traditional system has been modernizd to use hot water running embedded tubes to heat floors.

With the recent changing trends in housing production methods, from wet construction to prefabricated or dry construction, various types of prefabricated radiant floor heating panels have been developed and used. In order to develop a thermally effective alternative, a thermal evaluation using an expensive and time consuming test would be needed.

In this study, for the precise thermal analysis of the various prefabricated radiant floor heating panels, a heat transfer analysis program has been developed and computerized for two selected typical prefabricated panel types.

In an effort to assist the designer in determining the optimum values for certain design variables in the early design stages, sensitivity analyses using the computer simulation are conducted to see how the panel will perform in different ranges of design and control parameters.

SELECTION OF THE TYPICAL PREFABRICATED PANEL TYPES

The prefabricated radiant floor heating panels can be classified as panels with thermal mass and without thermal mass. In this study, two typical models were selected based on this classification criteria, and analysis models were developed for each of them. The details of two typical models are as follows.

A-model

As shown in Fig.1, its structure is similar to that of the traditional radiant floor heating panel. The differences are a thin thermal mass and the use of a mortar as a thermal mass instead of gravel.

B-model

As shown in Fig.2, heat from the pipe is transferred to the heat emitting board(Aluminum plate) through the heat conducting conduit. The lower part is filled with poly-urethane foam insulation.

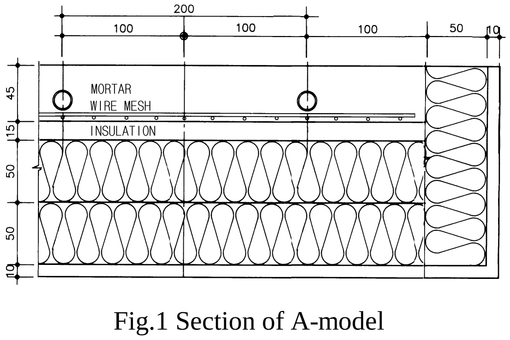


Fig.1 Section of A-model

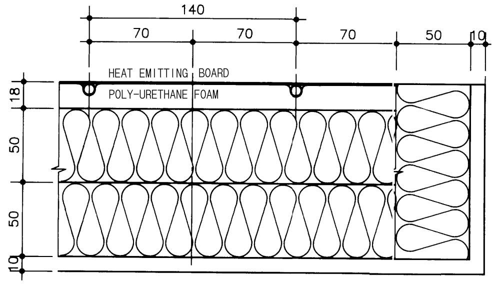


Fig.2 Section of B-model

ALGORITHM

The heat transfer analysis methods, which so far, have been effective for the analysis of the radiant floor heating panel are the steady-state one dimensional analysis by Adlam[10], Kolmar Liese's Method explanied by Kim[3] and the unsteady-state analysis by M. Udagawa[8] and by R.W. Shoemaker[9]. Recently, Zhang et al.(1989)[11] use steady state semi-analytical equations to relate the supply water temperature, panel heat output, and physical characteristics of the panel. But the design method focuses on the panel heat output as a function of supply water temperature rather than panel surface temperature and model may not be applicable to those structures that deviated from the assumptions made during the model development. Another study on the comparison between the thermal performance of a radiant heating system and a warm air system was presented by Zmeureanu et al.(1988)[12], using detailed computer program to simulate the transient heat transfer processes occurring in a room. The simulation mainly focuses on the room air temperature and other thermal comfort index, assuming that the radiant panel surface temperature is kept at 31.5℃.

The effect of floor temperature on comfort is very significant in domestic housing because the Koreans usually sit or sleep on the warm floor which supplies heat to the feet as well as the body by conduction. So the contact thermal sensation is unique due to the increased contact between body and floor surface from this kind of life style. It was indicated that comfort can be achieved by maintaining the floor temperature 25～38.8℃, average 31℃ under the room air temperature condition 17.5～24.5℃. Furthermore the uniformity of surface temperature is also major factor on contact thermal comfort. So in this study, heat transfer analysis computer models, which can calculate the temperatures of all the nodes in a panel, have been developed with the unsteady-state two dimensional analysis method using the finite difference method(FDM) for the accurate analysis of the time varying thermal characteristics of the panel. As the heat flow in the panel is a three dimensional unsteady state heat transfer, convection heat transfer of hot water in the pipe and conduction heat transfer in the panel should be calculated simultaneously in three dimensional coordinates. This will require tremendous amounts of calculations. The section of panel has equally spaced pipe layout and the specific heat of the water(the heating medium in the pipe) is higher than that of the thermal mass. So, it can be assumed that the temperature gradient in the direction of the hot water pipes is negligible compared to that in the direction perpendicular to the pipes. Thus the heat transfer in the panel was simplified by analyzing in section which is perpendicular to the pipes. This will make it possible to get very close results with the minimum amount of calculations. As the hot water pipes are equally spaced in this section, it can be assumed that the same unit section is symmetrically repeating.

In this study, the heat transfer in the panel is analyzed as a two dimensional unsteady-state heat transfer in the unit section perpendicular to the pipe.

Conduction in panel

The two dimensional unsteady-state conduction equation becomes,

$$k\left(\frac{\partial^2 T}{\partial X^2}+\frac{\partial^2 T}{\partial Y^2}\right) = \rho\cdot Cp\cdot\frac{\partial T}{\partial t}\quad . \qquad (1)$$

Convection and radiation on panel

The heat flow on the panel surface can be found by the following equation according to Newton's cooling law.

$$q = E\cdot A\,(T_{air} - T_{surface}) \qquad (2)$$

Where, $E = h_c + h_r$

(A) Convective heat transfer coefficient

The convective heat transfer coefficient(h_c) becomes,

$$h_c = 2.26\times\sqrt[4]{\Delta T}\ . \qquad (3)$$

(B) Radiative heat transfer coefficient

The radiative heat transfer coefficient(h_r) between the panel surface and the wall surface is calculated as follows.

$$h_r = 4\times\varepsilon\times\sigma\times T_m^{\ 3} \qquad (4)$$

Convection in the pipe

(A) Convective heat transfer coefficient in water flow

The convective heat transfer coefficient(h_w) between the water and the smooth inside surface of the circular pipe is as follows.

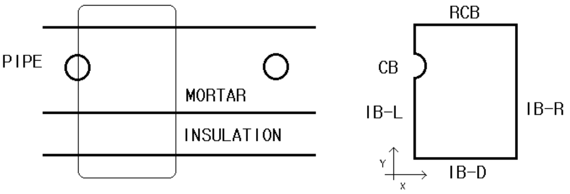
$$h_w = Nu_D\times\frac{k}{D} \qquad (5)$$

For the fully developed turbulent flow(Re_D>2,300) in the circular pipe, values from Dittus-Boelter's equation[2] is adapted.

$$Nu_D = 0.023\times Re_D^{\ 0.8}\times Pr^{\ 0.3} \qquad (6)$$

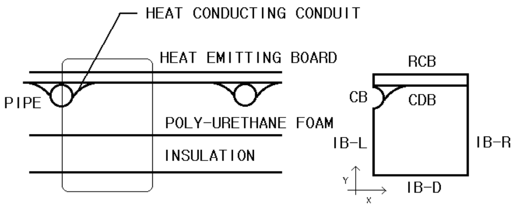
(B) Heat transfer when the flow is stopped

After the flow is stopped, the lumped capacitance method[2] is applied assuming the stopped fluid as a uniform solid body.



(a) Section of A-model (b) Control volume

Fig.3 Boundary conditions of A-model



(a) Section of B-model (b) Control volume

Fig.4 Boundary conditions of B-model

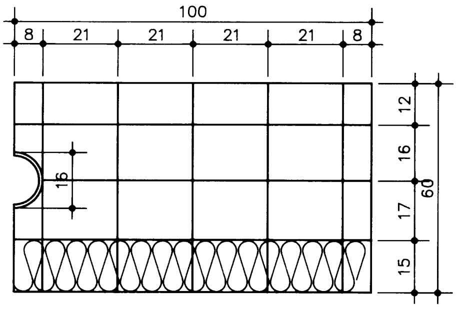


Fig.5 Grid system of A-model

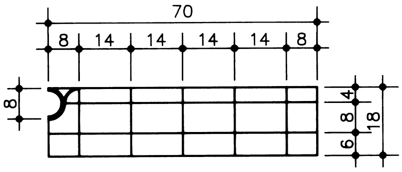


Fig.6 Grid system of B-model

Boundary Conditions

The following boundary conditions as in Fig.3 and Fig.4, are established for the analysis models.

(A) Convection boundary (CB)

During the hot water supply, the convective heat transfer occurs between the water and the inside surface of the pipe.

(B) Isotropic boundary(IB-L, IB-R, IB-D)

The vertical center line of the pipe and the vertical line at half of the pitch are the isotropic boundary, and an symmetrical temperature distribution is attained. Below the panel is perfectly insulated, and it is assumed that no heat loss occurs.

(C) Radiation and convection boundary(RCB)

The surface of the panel is exposed to the radiative and the convective heat transfer.

(D) conduit boundary(CDB)

For the analysis of the complex heat transfer along the heat conducting conduit of B-model, a simplified conduction model is developed.

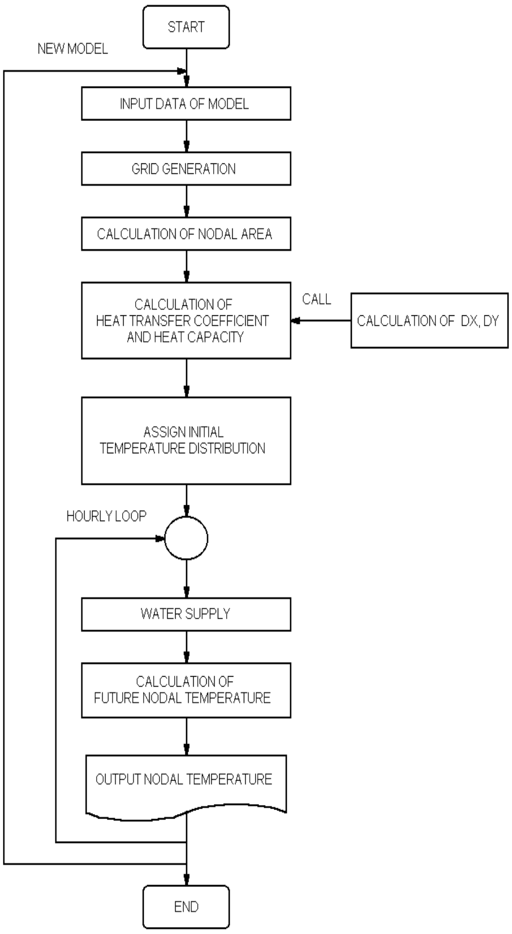


Fig.7 The flow chart of the program

DEVELOPMENT OF A COMPUTER MODEL

In this study, the numerical solution of the finite difference method is attained by the implicit method which can get the stable result regardless of the time increment. For the solution of the implicit method, the Gauss-Seidel iteration method is utilized. In that iteration, the calculations, to get the temperatures of nodes for the next iteration step, are repeated until the solution converges. The convergence criterion adapted for this study was,

$$\frac{\sum \left| T_{i,j}^{k-1} - T_{i,j}^{k} \right|}{\sum \left| T_{i,j}^{k} \right|} \le 1 \times 10^{-3} . \qquad (7)$$

Grid system

The grid systems established in dividing the inside of the panel into finite nodes are illustrated in Fig. 5 and Fig. 6.

The structure of the program

The flow of the developed computer program is shown in Fig. 7.

Validation

The developed computer simulation program was validated by comparing with the scaled model test results, which were conducted in the previous study[7]. The comparisons of the panel surface temperatures above the pipe and at half of pitch, and panel bottom temperatures below the pipe and at half of pitch are shown for the validation from Fig.8 to Fig.11. The temperature differences between the simulated results and test results were generally less than 0.5℃. But some temperature deviation showed at the panel bottom nodes when hot water is supplying.

SIMULATION

For the analysis of the thermal performance of the panel, the maximum difference of surface temperatures and the average surface temperature of the panel were evaluated while the room temperature is maintained at 21℃.

Evaluation

KIER(Korea Institute of Energy Research)[7] proposed the acceptable ranges of the temperature as follows ; room temperature 21±2℃, panel surface temperature 31±2℃, and maximum surface temperature difference limit 4℃. The Proclamation No.396 (1987.8.19) [14], that restricts the floor panel structure and sizing, by the Ministry of Construction in Korea is as follows ; thermal mass layer 40～70mm, finishing layer 15mm～25mm above the pipe, pipe diameter over 15mm, and pipe spacing 150～400mm.

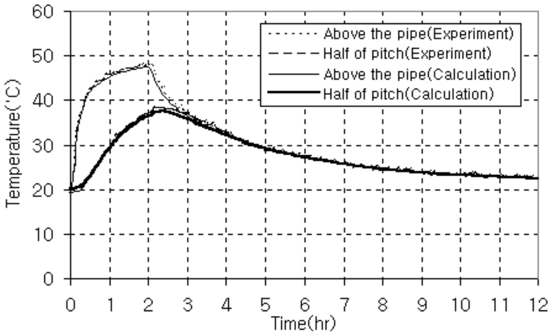


Fig.8 Validation of A-model(surface temperature)

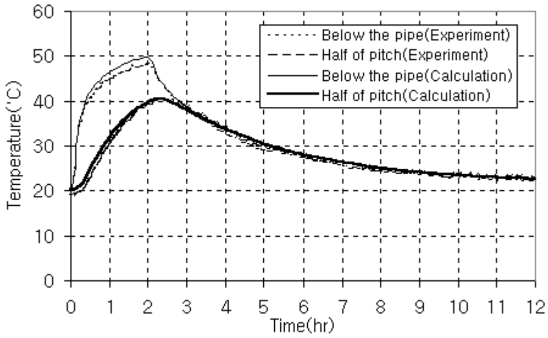


Fig.9 Validation of A-model(bottom temperature)

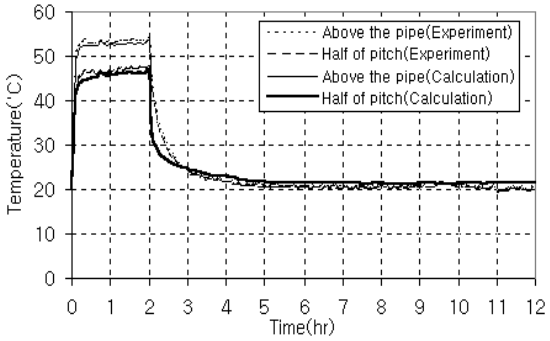


Fig.10 Validation of B-model(surface temperature)

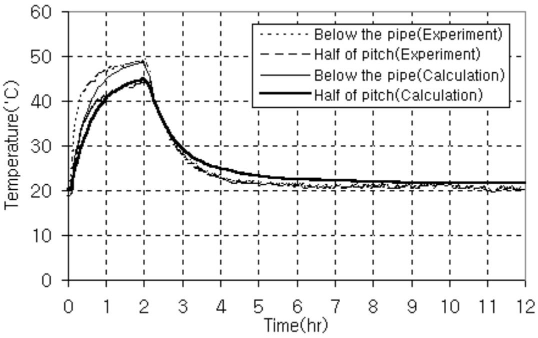


Fig.11 Validation of B-model(bottom temperature)

The apartment building in Korea is operated based on two kinds of heating modes (intermittent heating with time schedule and continuous heating with indoor temperature feedback). The former is non-feedback control which just gives 3 or 4 times of hot water supply from the central plant. Despite of a large room temperature fluctuations, this intermittent heating is more dominant than the continuous heating because of the advantages in the initial cost, simple installation, ease of maintenance and the lower stand-by loss of the system.

The survey of the intermittent heating schedule in existing apartment building[13] shows that the heating cycle rate is 2～4 cycles per day, the period of one cycle is 2 to 3 hours, and the supply water temperature is 50～70℃. The simulation results were evaluated on the basis of the comfort range, operating condition and structural regulation.

Simulation variables

Thermal performance of floor heating panel is affected by two main parameters. Those are the design parameter and the control parameter. So, the simulation was done through the following procedures.

First, both models were simulated with the variations of pipe spacing, pipe diameter and supply water temperature to examine the interrelated effects of them. Second, the application of both models in two heating modes were checked and the desirable improvement was suggested. Both models were simulated with the variations of pipe spacing and supply water temperature, additionally B-model with the variations of thickness and thermal conductivity of heat emitting board, for the continuous heating mode. A-model was intensely simulated with the variations of thermal mass and depth of pipe bury for the intermittent heating mode.

Supply water temperature and pipe diameter

Fig.12, Fig.13, Fig.14 and Fig.15 show the average surface temperature and the temperature difference of the panel with the variations of the supply water temperature and pipe diameter based on two different pipe spacing (A-model : 200mm, 100mm, B-model : 140mm, 70mm). As the pipe diameter increases, the average surface temperature becomes higher and the temperature difference slightly increases. The pipe diameter has more effect on the average surface temperature than the surface temperature difference. In the case of narrow pipe spacing, this effect becomes trivial(Fig.13, Fig.15). And the surface temperature is more affected by the pipe spacing and supply water temperature rather than the pipe diameter.

Supply water temperature and pipe spacing

The simulation with the variations of supply water temperature and pipe spacing was conducted. Fig.16 and Fig.17 show that the average surface temperature is inversely proportional to the pipe spacing. The surface temperature difference between maximum and minimum surface temperatures of the panel is proportional to the pipe spacing. The slope of these two curves becomes smooth as the supply water temperature decreases. The desirable supply water temperature and pitch can be selected by the average surface temperature and the temperature difference limit.

A-model : 35℃ of water temperature and 260mm of pipe spacing are desirable due to the average surface temperature and difference within the comfort range proposed by KIER. With the existing panel design, A-model is not applicable to intermittent heating mode because high water temperature causes discomfort surface temperature. But the increase of material thickness could make it possible to fit for the intermittent heating mode.

B-model : 35℃ of water temperature and 175mm of pipe spacing are desirable due to the average surface temperature and difference within the comfort range proposed by KIER. B-model is not applicable to intermittent heating mode, either. It has no effective alternatives for the application of the intermittent heating mode. So this model must be installed for the continuous heating mode if it were not for the great modification of panel design.

Depth of pipe bury in fixed thermal mass thickness (A-model)

A-model was intensely simulated with the thickness variations of thermal mass and depth of pipe bury for the intermittent heating mode (60℃ of supply water temperature and 1 cycle heat supply for 2-hours). For the intermittent heating mode, the increase of material thickness is needed. So within 95mm of the maximum allowable thickness of thermal mass and finishing layer according to Proclamation No.396 (1987.8.19), the effect of depth of pipe bury on the panel surface temperature is examined. In Fig.18 and Fig.19, when the pipe is embedded in upper part of the panel, the average surface temperature becomes very high and the panel has quick response to heating start and stop. When the pipe is embedded in lower part of panel, the surface temperature difference becomes less and even temperature distribution can be obtained.

The material thickness above the pipe has more effect on the uniform surface temperature and the thermal storage effect than the thickness below the pipe. Through the additional simulations, it can be noticed that the increase of material thickness below the pipe has no great effect on the thermal storage and surface temperature differences of the panel. So the increase of the thickness below the pipe is not desirable for the intermittent heating application.

Material thickness above the pipe (A-model)

The step increment of material thickness above the pipe from existing thickness was simulated. As shown in Fig.20, it fails to maintain the comfortable surface temperature ranges (31±2℃) proposed by KIER (They didn’t specify the heating mode to maintain that comfort range. It seems that ranges are not appropriate for the intermittent heating mode). But the thickness from 42.5 to 65mm keeps surface temperature within the comfortable ranges (25～38.8℃) for longer periods compared to other thickness. And it shows a little slow response to heating start and stop. In Fig.21, 65mm of thickness above the pipe can satisfy the temperature difference limit (4℃). So it is desirable to have less than 65mm above the pipe for the intermittent heating mode for A-model.

Thickness and thermal conductivity of heat emitting board(B-model)

With the increased thickness and thermal conductivity of the heat emitting board, the average surface temperature becomes higher and the temperature difference lessens. But the change was so small, and has no great effect on panel surface temperature.

CONCLUSIONS

For the precise thermal analysis of the various prefabricated radiant floor heating panels, a heat transfer analysis model was developed and computerized for the selected two typical panel types, and verified by comparing with the results of model tests. The developed computer program simulation results can be summarized as follows.

1. The average surface temperature is more affected by the hot water pipe spacing and supply water temperature rather than by the pipe diameter. The size of pipe diameter has more effect on the average surface temperature than on the surface temperature difference.
2. The pipe spacing and the water temperature are the important variables. The desirable supply water temperature and pipe spacing for the continuous heating mode can be selected as follows.

a) A-model : 35℃ of water temperature and 260mm of pipe spacing are desirable. With the existing panel design, A-model is not appropriate for intermittent heating mode.

b) B-model : 35℃ of water temperature and 175mm of pipe spacing are desirable. B-model is not appropriate for intermittent heating mode, either.

3. The increase of thickness above the pipe is effective in A-model for the intermittent heating mode. Less than 65mm above the pipe is desirable for the comfortable panel surface temperature.
4. The variations of thickness and thermal conductivity of heat emitting board show no great effect for B-model. So this model must be installed for the continuous heating mode if it were not for the great modification of panel design.

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NOMENCLATURE

T	Temperature in node
k	Thermal conductivity
ρ	Density
C_p	Specific heat
h_c	Convective heat transfer coefficient
h_r	Radiative heat transfer coefficient
ε	Effective emittance
σ	Stefan-Boltzmann constant
ΔT	The temperature difference between the room air and the panel surface
T_m	The average surface temperature of the walls and the panel
h_w	Convective heat transfer coefficient in pipe
Nu_D	Nusselt number
k	Thermal Conductivity of water
D	Diameter of the pipe
Re_D	Reynols number
Pr	Prantle number

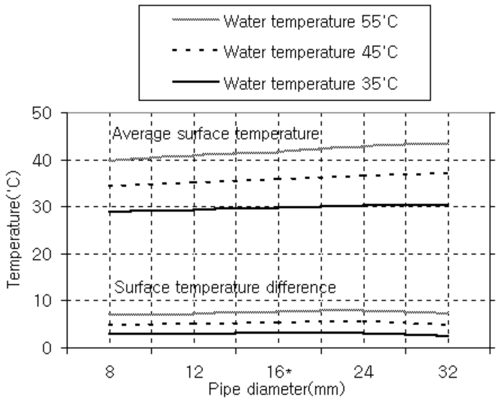

Fig.12 Variations of supply water temperature and pipe diameter(A-model, pipe spacing 200mm)

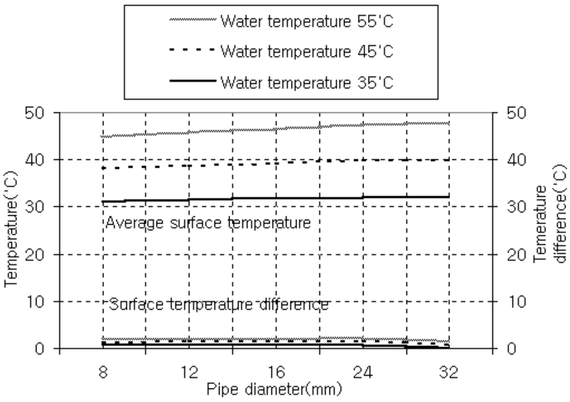

Fig.13 Variations of supply water temperature and pipe diameter(A-model, pipe spacing 100mm)

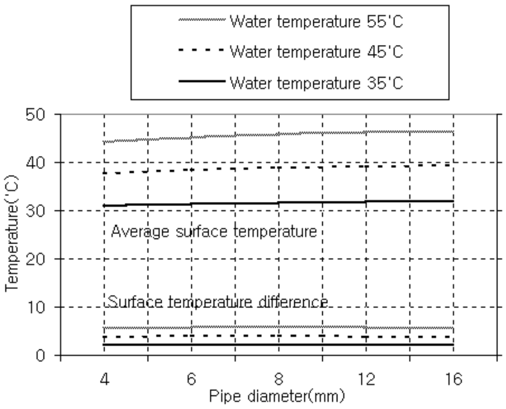

Fig.14 Variations of supply water temperature and pipe diameter(B-model, pipe spacing 140mm)

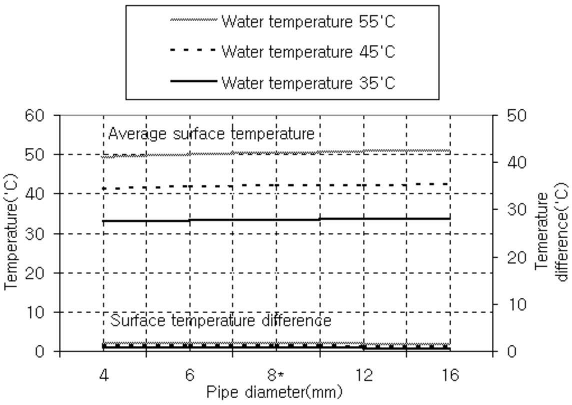

Fig.15 Variations of supply water temperature and pipe diameter(B-model, pipe spacing 70mm)

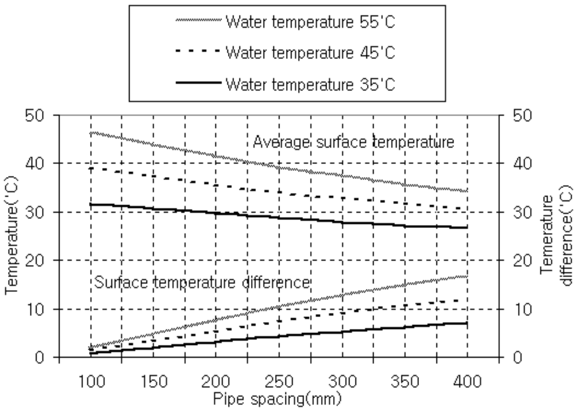

Fig.16 Variations of supply water temperature and pipe spacing (A-model)

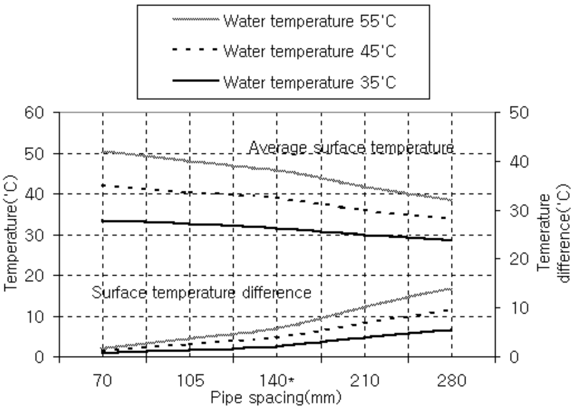

Fig.17 Variations of supply water temperature and pipe spacing (B-model)

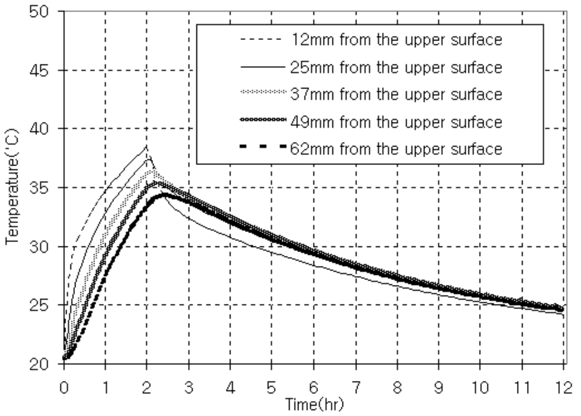

Fig.18 Variations of depth of pipe bury in fixed thermal mass thickness (A-model, average surface temperature)

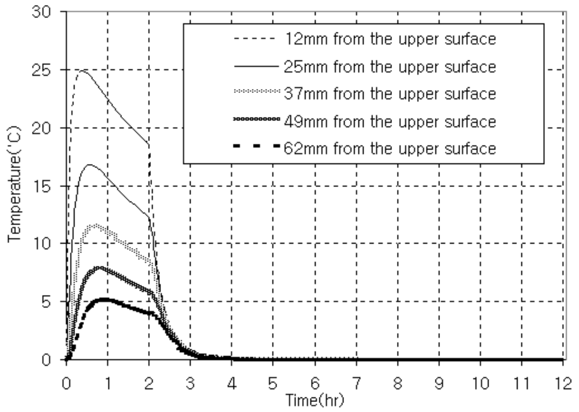

Fig.19 Variations of depth of pipe bury in fixed thermal mass thickness (A-model, surface temperature difference)

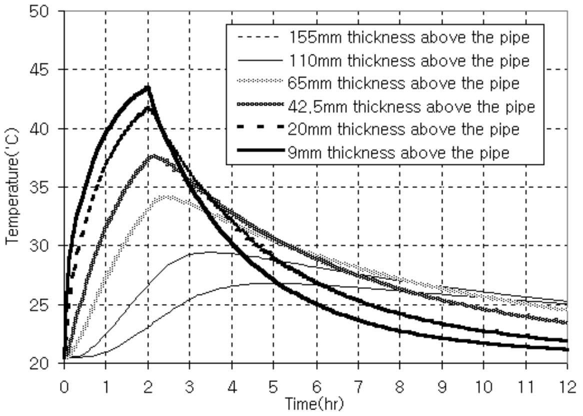

Fig.20 Variations of material thickness above the pipe(A-model, average surface temperature)

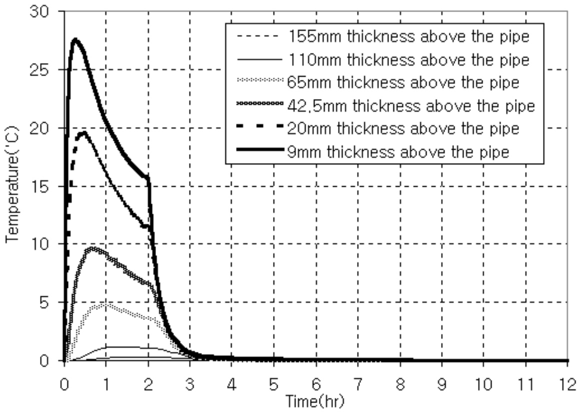

Fig.21 Variations of material thickness above the pipe(A-model, surface temperature difference)