International Gas Union Research Conference 2011

INFLUENCE OF BUILDING INSULATION PERFORMANCE AND HEATING SYSTEMS ON THERMAL COMFORT AND ENERGY

Main author
T. Omori
Tokyo Gas Co., Ltd.
Tokyo
JAPAN

Co-author
S. Tanabe
Waseda University
Tokyo
JAPAN

ABSTRACT

Thermal comfort in a heated room is affected by the heating system and characteristics of the building envelope, especially the thermal insulation properties. Coupled simulation of convection and radiation is conducted to estimate the thermal energy required for a floor heating system and a wall-mounted air-conditioning system. Thermal energy inputs required to maintain the same equivalent temperature for a simulated human body under five different building insulation levels, including three levels of the Japanese energy conservation standards for residential buildings (i.e., 1980, 1992, and 1999 Standards), are calculated. It can be concluded that the floor heating system is more comfortable and energy efficient than the air-conditioning system.

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1. INTRODUCTION

Energy consumption by the residential sector accounts for 13.3% of domestic energy consumption in Japan (fiscal year of 2007) and has been steadily increasing. Energy use for room heating accounts for 22% of the residential use.

The energy conservation law was initially enacted in 1980 and created the Japanese energy conservation standard for residential buildings. The standard was later modified by the amendments of 1992 and 1999, resulting in three levels of standards: 1980, 1992, and 1999. The energy conservation law also includes top runner programs, aiming to raise efficiency levels of designated appliances and equipment. The wall-mounted air-conditioning unit efficiency has been improved considerably and has become very popular for room heating. The hot water floor heating system is also popular in Japan for its ability to create a thermally comfortable space. The thermal environments and thermal comforts created by different heating systems are known to differ significantly. A method of evaluating the energy consumption for different heating systems is necessary.

Thermal comfort of a person can be correlated with indoor thermal environments in terms of air temperature, radiant temperature, air velocity, and humidity. Considering only one or two of these properties is insufficient for evaluating heating systems in terms of comfort and energy consumption. A person exchanges heat with their surrounding environment by convection and radiation. The whole-body thermal comfort is analogous to the amount of dry heat loss from a person under different heating conditions at the same humidity. A thermal manikin is used for measuring the dry heat loss of a human body, where the thermal manikin heat loss is normalized to define the equivalent temperature [1].

In this study, coupled simulations of convection and radiation have been conducted for a living room with a seated human body. The human body model, a computer-simulated person, is controlled to maintain its inner temperature at 36.4 °C. The insulation of the room is varied with five levels: three levels of the Japanese energy conservation standard for residential buildings mentioned previously, super insulation, and perfect insulation. The room is heated to maintain the equivalent temperature by either a floor heating system or an air-conditioning system.

2. ANALYSIS MODEL

Room with floor heating system and air-conditioning system

A schematic diagram of the living room is shown in Figure 1. Its inner room dimensions are as follows: 5.33 m wide, 3.77 m deep, and 2.4 m high; these dimensions correspond to a 13.5 Japanese tatami mat. A pair of windows, 1.64 m wide and 2.0 m high each, is mounted on the south wall. The windows are covered with curtains, 1.74 m wide and 2.05 m high for the east window and 1.69 m wide and 2.05 m high for the west window. Each curtain hangs 50 mm away from the window and sits 15 mm above the floor. Air flows into the room from an air inlet on the west wall positioned 1.6 m above the floor, and from 24 air supply cylinders, eight of which are on the south wall and 16 on the west wall; each cylinder is 18 mm × 18 mm. Air is equally supplied from the inlet and the set of cylinders. The entire amount of supplied air is vented through a 90-mm square opening located on the north wall positioned 0.15 m above the floor. The ambient air temperature remains constant at 5.5 °C. Overall heat transfer coefficients of the walls and the ventilation rate for each case are listed in Table 1. Insulation performance of the building model is set to five levels: 1980, 1992, and 1999 Japanese energy conservation standards, super insulation, and perfect insulation. The heat loss of the perfect insulation case occurs only through ventilation loss in order to model the most uniform thermal environment with the least energy consumption. The north and east walls as well as the ceiling are assumed to be thermally insulated by adjacent rooms. The thermal resistance of the curtains is assumed to be 0.01 m^2K/W.

The structure of the floor heating system is shown in Figure 1. A pair of heating panels, 2.38 m wide and 3.29 m deep each, covers 78% of the floor area. Thermal resistance from the upper heater

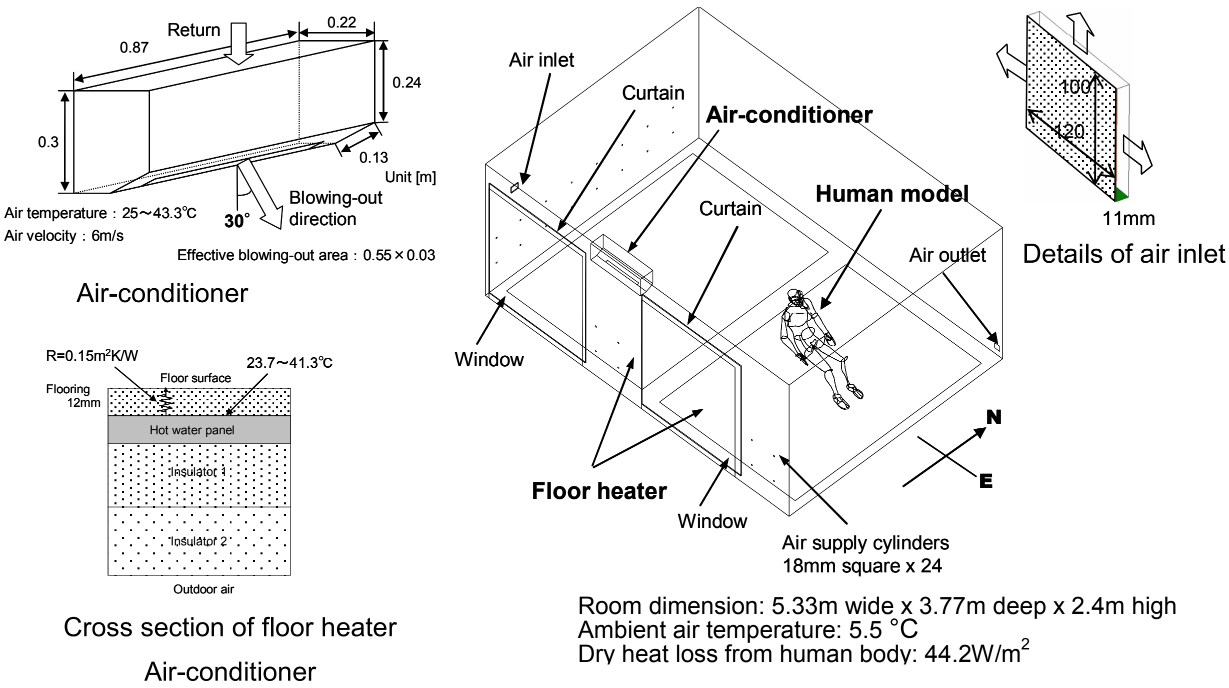


Figure 1 Analysis model

panel to the floor surface is 0.15 m^2K/W. The air-conditioning unit is installed on the south wall 2.1 m above the floor. Hot air is blown downward at a 30° angle from the vertical axis and at an air speed of 6.0 m/s. The effective dimensions of the outlet are taken from data of an actual air-conditioner and are set at 0.55 m × 0.03 m. Approximately 22,000 solid surface meshes and 260,000 volume cells are used for the computations.

Table 1 Energy conservation standards and insulation performance

Insulation level		1980 Standard	1992 Standard	1999 Standard	Super Insulation	Perfect Insulation
Heat loss coefficient [W/m^2K]		5.58 (5.58)	3.95 (3.95)	2.7 (2.7)	1.7 (1.9)	0.4
Overall heat transfer coefficient [W/m^2K]	North wall	—	—	—	—	—
	South wall	1.32 (1.05)	0.78 (0.78)	0.53 (0.53)	0.53 (0.5)	—
	West wall	1.32 (1.05)	0.78 (0.78)	0.53 (0.53)	0.53 (0.5)	—
	East wall	—	—	—	—	—
	Floor	1.32 (1.05)	0.51 (0.62)	0.26 (0.34)	0.18 (0.2)	—
	Ceiling	—	—	—	—	—
	Window	6.98 (6.98)	6.5 (6.5)	4.65 (4.65)	1.9 (1.9)	—
Ventilation rate [1/h]		1.0 (1.0)	0.7 (0.7)	0.5 (0.5)	0.5 (0.5)	0.5

Figures in parentheses are the standard data in an energy conservation handbook.

Human Body Model

A digital human body model simulating a 1.72 m tall adult male with a surface area of 1.86 m^2 is seated as illustrated in Figure 2. It is controlled by Fanger's thermally neutral equation defined in Equation 1, which is also used for the physical thermal manikin as comfort control [2].

$$q = \frac{36.4 - t_{cl}}{0.054 + 0.155 I_{cl}} \qquad (1)$$

where q is the dry heat loss [W/m^2], t_{cl} is the surface temperature of the clothing [□], and I_{cl} is the resistance of clothing [clo]. The dry heat loss is 44.2 W/m^2 under the thermally neutral condition with a metabolic rate of 1.1 Met and standard winter clothing. The dry heat loss from the human body can be correlated to the whole-body thermal comfort. Therefore, the amount of dry heat loss can be considered equivalent to the thermal environment as a whole. The dry heat loss corresponds to the equivalent temperature [1].

The center of the model's buttocks is positioned 1.9 m and 1.6 m away from the south and east walls, respectively, and 20 mm above the floor. The amount of clothing, 1 clo, corresponds to standard clothing for winter time, except for 0.3 clo for the feet, 2 clo for the head, and 0 clo for the face, neck, and hands. The number of human surface meshes is approximately 9,000. Three layers of prism-shaped boundary layer meshes extend up to 1 mm from the surface of the human model with an expansion ratio of 1.1. Values of y^+ are less than unity in most locations. Furthermore,

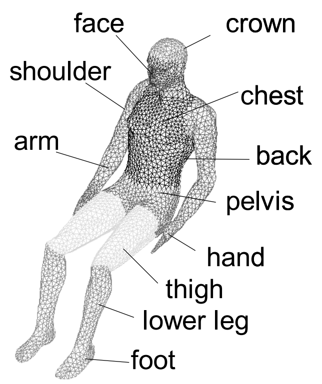


Figure 2 Human body model

three layers of 3-mm-thick boundary layer meshes with an expansion ratio of 1.1 are distributed near the walls and the curtains.

3. ANALYSIS METHOD AND BOUNDARY CONDITIONS

Coupled simulations of convection and radiation were conducted for the room with a human body under the thermally neutral condition. Fluid flow computations are based on the SIMPLE algorithm and a second-order finite difference scheme MARS (Monotone Advection and Reconstruction Scheme). A low-Reynolds number turbulence model from Lien et al. [3] is applied. Radiation is treated using a Monte Carlo method [4] with improved accuracy through symmetrization.

Boundary conditions are summarized in Table 2. For each insulation level, the heat release from the heating system to the room is determined such that the average dry heat loss of the seated human body model corresponds to the thermally neutral condition (i.e., 44.2 W/m^2). This is achieved by controlling the upper panel temperature within a range of 23.7–41.3 °C for the floor heating, or by controlling the expelled air temperature between 25.0–43.3 °C for the air-conditioner. Thermal resistance across the interior wall surface to ambient air is calculated by subtracting the heat transfer resistance in the room, 0.13 m^{2}K/W, from the inverse of the overall heat transfer coefficient provided in Table 1.

Table 2 Boundary conditions

<table>
<tr><td rowspan="10">CFD routine</td><td colspan="2" rowspan="2">Inlet/Outlet</td><td colspan="2">Air-conditioner</td><td rowspan="2">Air supply</td><td rowspan="2">Cylinders</td><td rowspan="2">Exhaust</td></tr>
<tr><td>Outlet</td><td>Intake</td></tr>
<tr><td rowspan="4">Inflow BC</td><td>Velocity [m/s]</td><td>6.0
(30°downward from the vertical axis)</td><td>- 0.49 ※※</td><td>Upward: 0.84
Horizontal: 1.01</td><td>0.429※※※</td><td>- 0.865※※※</td></tr>
<tr><td>Temperature [℃]</td><td>25.0～43.3※</td><td>—</td><td>5.5</td><td>5.5</td><td>—</td></tr>
<tr><td>Turbulence intensity [−]</td><td>0.1</td><td>—</td><td>0.1</td><td>0.01</td><td>—</td></tr>
<tr><td>Turbulence length scale [m]</td><td>0.055</td><td>—</td><td>0.01</td><td>0.0018</td><td>—</td></tr>
<tr><td colspan="7"></td></tr>
<tr><td colspan="2">Solid surface</td><td colspan="2">Human model</td><td colspan="3">Walls</td></tr>
<tr><td rowspan="2">Wall BC</td><td>Velocity</td><td colspan="5">no-slip</td></tr>
<tr><td>Temperature</td><td colspan="5">Temperature set through the radiation routine</td></tr>
<tr><td colspan="2" rowspan="4">Radiation routine</td><td>Solid surface</td><td colspan="2">Human model</td><td colspan="2">Floor heating</td><td>Wall</td></tr>
<tr><td>Temperature</td><td colspan="2">$q = \frac{36.4 - t_{cl}}{0.054 + 0.155 I_{cl}}$
q : heat flux [W/m^2]
t_{cl} : surface temperature of clothing [℃]
I_{cl} : resistance of clothing [clo]</td><td colspan="2">Upper surface of heating panel: 23.7～41.3 ℃※
Thermal resistance up to the floor surface:0.15 m^{2}K/W</td><td>Thermal resistance between room-side surface and outdoor air is set,
Outside air temp: 5.5 ℃</td></tr>
<tr><td>Emissivity [−]</td><td colspan="2">0.9</td><td colspan="3">0.9</td></tr>
<tr><td>Convective heat flux</td><td colspan="5">Convective heat flux set through the CFD routine</td></tr>
</table>

※ For each energy conservation standard, dry heat loss from the human body is controlled to maintain 44.2 W/m^2

※※ To match the mass flow rate of hot air to that of air intake, air speed is adjusted for every hot air temperature

※※※ Indicates ventilation rate of 0.5 h^{-1}

4. RESULTS

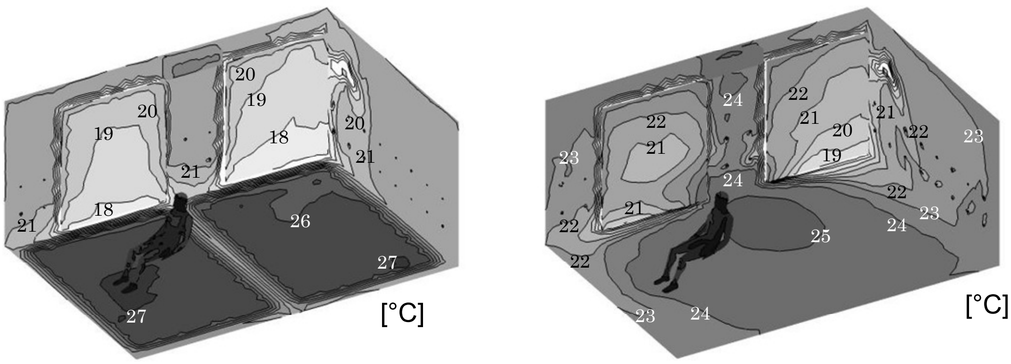
Results are compared in terms of the insulation levels.

Surface temperature distribution

Surface temperature distributions are shown for the 1999 Standard of energy conservation in Figure 3. Under the condition of 44.2 W/m^2 of dry heat loss of the human body, floor surface temperatures are 2–3 °C higher and wall temperatures are 1–3 °C lower for the floor heating compared to the air-conditioning.

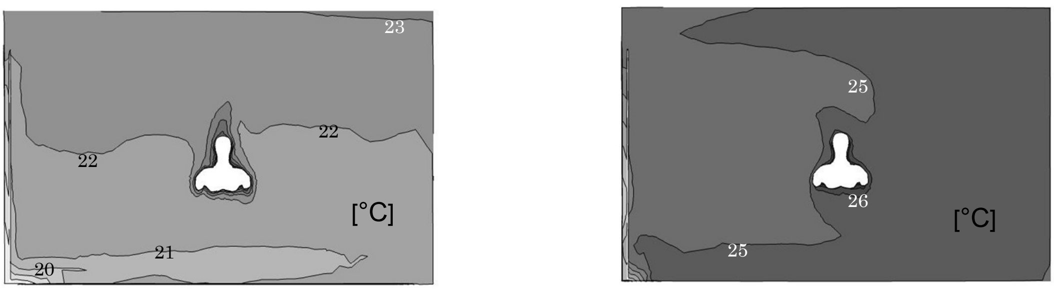
Air temperature and velocity distributions

Air temperature and scalar velocity distributions are shown for the 1999 Standard in Figure 4 and 5. Air temperatures are almost uniformly distributed for both heating systems. For the floor



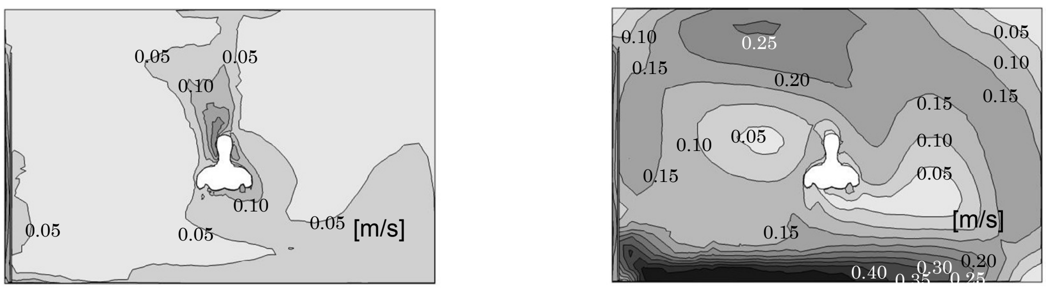
(a) Floor heating (b) Air-conditioning

Figure 3 Surface temperature distribution (1999 Standard)



(a) Floor Heating (b) Air-Conditioning

Figure 4 Air temperature distribution（1999 Standard）



(a) Floor Heating (b) Air-Conditioning

Figure 5 Scalar velocity distribution（1999 Standard） heating, air movement is calm and the vertical temperature difference is about 2 °C. For the air-conditioning, forced convection with large expelled air velocity makes the room air velocity higher (see Figure 5(b)), and the room air temperature uniform.

Skin temperature distribution

Figure 6 depicts the skin temperature distribution for the 1999 Standard. The results show that the temperatures of the hands and feet located near the floor are approximately 0.5 °C lower and the temperatures of the upper body are approximately 0.1 °C higher in the case of air-conditioning compared to the floor heating. Similar patterns are also found in the remaining insulation levels.

Vertical temperature distribution

Figure 7 represents air temperature distributions in the vertical direction at a location 0.6 m away from the east wall and 0.4 m away from the north wall. Poorer thermal insulation of the building leads to a lower floor temperature in the case of air-conditioning. The trend is reversed for the floor heating due to the increased heat output. As the building insulation performance increases, the temperature profiles of both systems approach one another.

Contrary to the idea that an air-conditioner creates larger temperature differences between cooler near-floor regions and hotter near-ceiling regions as the thermal insulation performance of a building degrades, an almost uniform distribution pattern results. This distribution is due to the high air velocity of the air-conditioning unit (6 m/s), which causes hot air to circulate throughout most of the room. For the perfect insulation case, the vertical air temperature profile is uniform at 24.6 °C. It is rational that the expelled air velocity decreases as the building insulation performance increases. An additional calculation with a velocity of 1 m/s was conducted and the resulting vertical air temperature profile is similar to that of the floor heating as shown in Figure 7.

In the case of floor heating, a nearly uniform temperature distribution is present in the vertical direction, except near the floor.

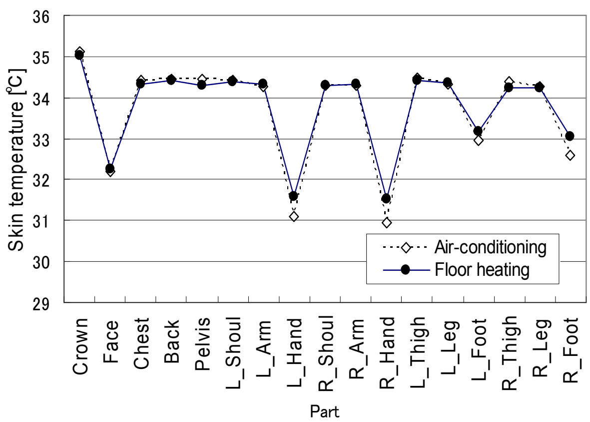


Figure 6 Skin temperature distribution (1999 Standard)

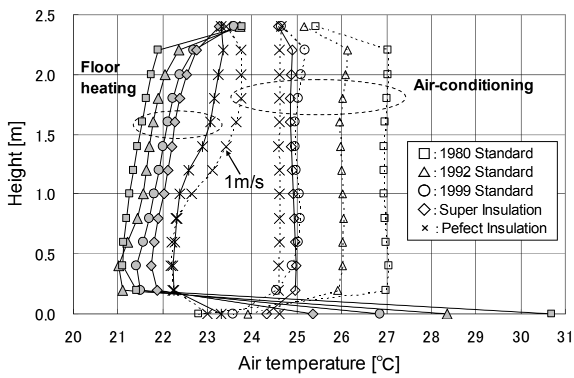


Figure 7 Vertical air temperature profile

Variation of air and wall temperatures with heat loss coefficient

As the heat loss coefficient decreases, air temperature increases and the heated floor surface temperature decreases, but the average temperature of all surfaces is nearly constant in the case of floor heating system as shown in Figure 8. In the case of air-conditioning, air temperature decreases to approach the constant average temperature of all surfaces under the perfect insulation condition. All temperatures of both systems approach 23 °C under the perfect insulation condition. The equivalent temperature of this heated room is regarded as 23 °C, since velocity in the room is nearly zero.

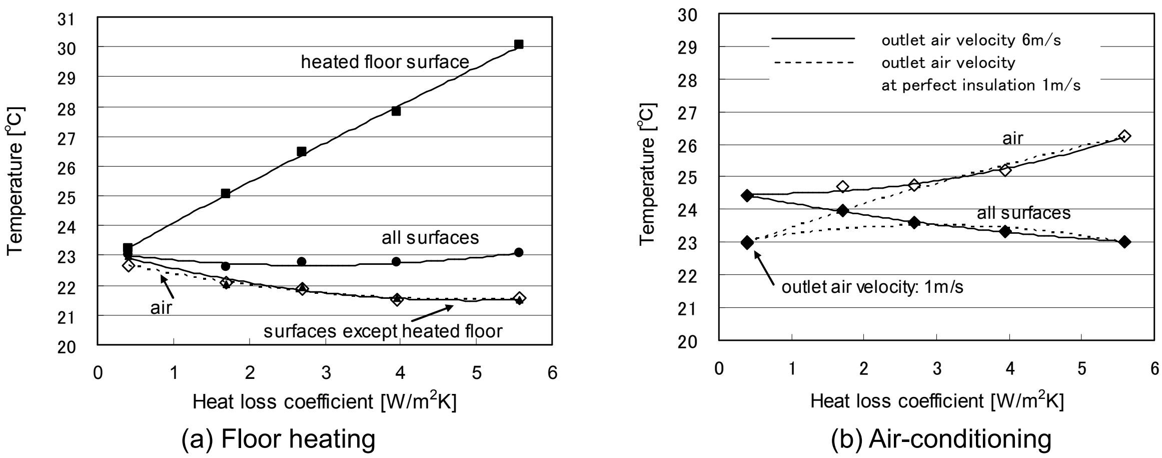


Figure 8 Variation of temperatures with heat loss coefficient

Heat release to the room

Heat inputs required to heat the room for both systems are presented in Figure 9 in terms of the insulation performance of the building model with a 44.2 W/m^2 dry heat loss of the human body. The thermal energy supplied to the heating panel of the floor heating system rises to heat the inside of the room; however, it also travels downward causing a heat loss referred to as the floor heat loss. When the room beneath the floor is also heated, the floor heat loss is zero. Heat release from a person, dry heat loss, is added to the room. The floor heat loss is estimated by equation (2).

$$Q_p = A(t_p - t_{out}) / R_p \quad (2)$$

Where A is the heating floor area [m^2], R_p is the thermal resistance between the heating panel and under floor air [m^{2}K/W], t_p is the panel surface temperature [°C], and t_{out} is the under floor air temperature [°C]. R_{p_1999} is calculated by assuming the fraction of the heat rising, 0.83, for the 1999 Standard [5]. R_p for other insulation levels is modified using equation (3).

$$R_p = \frac{\text{Heat loss coefficient for 1999 Standard}}{\text{Heat loss coefficient for arbitary Standard}} \times R_{p\_1999} \qquad (3)$$

The heat release from the floor heating system to the inside of the room is found to have ratios of 0.75 (1999 Standard), 0.71 (1992), and 0.62 (1980) when the heat release of the air-conditioning for each case is set as unity. When a comparison is made to the total heat release, the ratios are 0.92 (1999 Standard), 0.89 (1992), and 0.81 (1980) as depicted in Figure 9.

The dotted line connects the points of the 1980, 1992, and 1999 Standards for the perfect insulation condition with a 1 m/s outlet air velocity of the air-conditioner. This indicates that the difference of heat release tends to decrease as the building insulation performance increases.

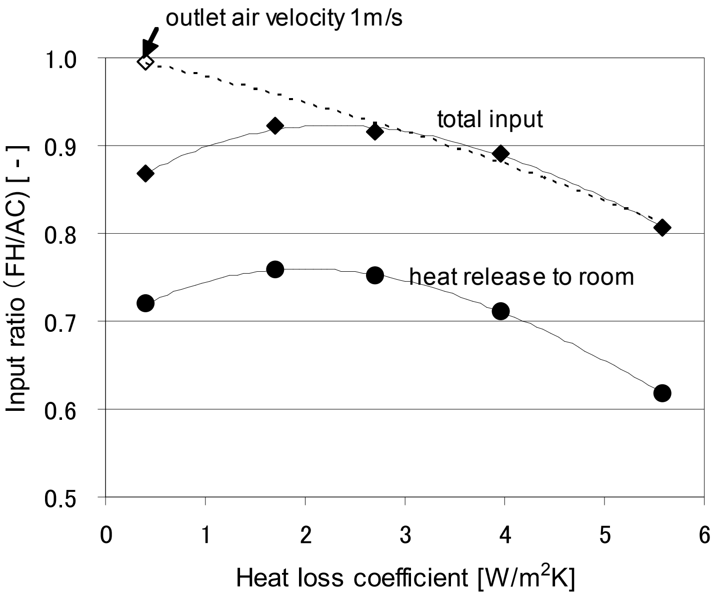


Figure 9 Variation of Input ratio with heat loss coefficient

5. CONCLUSIONS

Coupled simulations of convection and radiation were conducted for a room with a seated human body. The room was heated by a floor heating system or an air-conditioning system to maintain the equivalent temperature. The insulation of the room was varied at five levels: 1980, 1992, and 1999 Japanese energy conservation standards for residential buildings, super insulation, and perfect insulation.

The heat release from the floor heating system to the inside of the room was found to have ratios of 0.75 (1999 Standard), 0.71 (1992), and 0.62 (1980) when the heat release of the air-conditioning for each case was set as unity. When a comparison was made to the total heat release including the floor heat loss, the ratios were 0.92 (1999 Standard), 0.89 (1992), and 0.81 (1980).

The required thermal energy of the floor heating system with proper under-floor insulation was consistently lower than that of the air-conditioning system for each insulation level. The ratio of the thermal energy of the floor heating system to the thermal energy of the air-conditioning system converges as the thermal insulation increases.

The results indicate that the floor heating system is more comfortable and energy efficient than the air-conditioning system.

ACKNOWLEDGEMENT

The present study was conducted as part of activities of a research committee on evaluation of energy consumption for various heating systems fulfilling thermal comfort (Chairman, Professor M. Kamata, School of Engineering, Kanagawa University) organized by Better Living. The authors wish to express their gratitude for the cooperation provided by committee members.

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