EVALUATION OF HEAT TRANSFER COEFFICIENTS IN VARIOUS AIR-CONDITIONING MODES BY USING THERMAL MANIKIN

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ABSTRACT

Today, radiant floor heating systems are generally used for heating indoor environments of residential buildings. Radiant heating is physically comfortable and energy use largely differs compared to convective heating. It is believed that one reason for that is if there is an air flow, there is a difference in surface heat transfer of the human body. The heat transfer coefficient of the human body is an indispensable parameter to evaluate the indoor thermal environment, thermal comfort and heat loads in air-conditioned and ventilated buildings. Measuring the heat transfer coefficient of the human body under various air-conditioning conditions is rather difficult due to the multitude of factors, such as temporal changes in air temperature, airflow, cold draft, ventilation condition, storage effect, heat due to radiation, etc. Considering the above, in order to measure surface heat transfer precisely, an expensive thermal manikin or highly controlled environment laboratory is necessary. For this, recent developed computational fluid dynamics (CFD) are suitable. With the objectivity of prediction results and the degree of freedom of condition settings, they are able to become a beneficial investigative method.

The purpose of this paper is to verify the heat transfer coefficient of the human body and operative temperature in various air-conditioning environments, such as radiant floor heating and convective air heating using numerical simulation for evaluation of energy saving possibility in residential buildings. First, we measure the heat transfer coefficient of the human body using a thermal manikin in an experimental chamber under natural convection conditions. Then, to confirm CFD calculation accuracy, heat transfer coefficient of the thermal manikin with a temperature difference is verified using an experimental chamber model. Second, we measure the heat transfer coefficient of the thermal manikin surface with CFD calculation to verify its characteristics under forced convection conditions. Finally, we evaluate the operative temperature in an air-conditioned room using computer fluid dynamics to propose the correlation relation in various air-conditioning modes for residential buildings.

The measured results showed the heat transfer coefficient of the thermal manikin increased from temperature difference increase and air velocity. Furthermore, the results showed the radiant floor heating environment is more energy consumption efficient than a convective air heating environment.

INTRODUCTION

The prevalent types of heating systems for residential heating are divided into floor heating and air conditioning heating. Floor heating is radiant heating and air conditioning heating is convective flow heating, also, there is a large difference in comfort. It is believed that one reason for this is that the total thermal transfer at the body surface differs dependent on the presence of an air flow. Additionally, the difference in energy consumption from experimental analysis for each heating system is noted, but comfort is a challenging task. Although human sensible heat loss and the heat sensation are assumed to be of equal importance, expensive thermal manikins or a highly controlled laboratory environment is necessary. For this, recent developed CFD are suitable. With the objectivity of prediction results and the degree of freedom of condition settings, they are able to become a beneficial investigative method.

OBJECTIVE

In this study, as an experimental examination, we utilize a thermal manikin that allows the skin surface temperature and heat to be set arbitrarily, offering new approximations regarding body surface total thermal transfer under a natural convection environment as well as a controlled convection environment. Additionally, numerical analysis is conducted using a numerical thermal manikin, including consistency verification of the experiment values and numerical analysis. The goal of this research is to clarify the heat transfer characteristics from the body as well as secondary input energy when creating a warm environment using different heating systems.

METHOD

1) Total heat transfer coefficient of the thermal manikin under natural convection conditions

• Experimental evaluation

The experiment chamber for measuring the body total heat transfer in a wind free environment is shown in Figure 1(a). A thermal manikin was placed in a ventilated experiment chamber and surrounded by a black-out curtain (Figure 1(b)) to build a windless, stable air condition, as well as an MRT (mean radiant temperature). Under the above conditions, the room temperature was set to 22ºC. We measured the thermal manikin surface total heat transfer by placing it in a windless environment, because the thermal manikin surface temperature can vary from 30.5ºC to 37.5ºC.

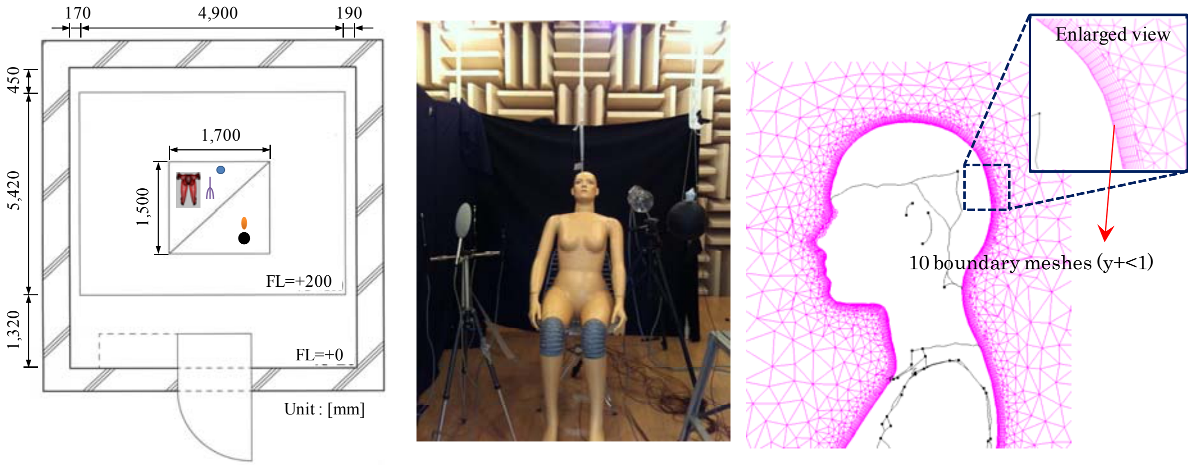
• Reliability evaluation of CFD analysis

To asserting the reliability of experiment results compared to CFD simulation, we placed a numerical thermal manikin under the same experiment conditions, obtaining a heat loss in comparison with the test results. The turbulence model is according to Abe-Kondoh-Nagano low Reynolds number k-ε turbulence model (AKN k-ε turbulence model) [K. Abe, T. Kondoh, and Y. Nagano (1994, 1995)], and a convective and radiative couple analysis (Figure 1(c)) was conducted. A numerical manikin is partitioned into 10 layers and the smallest mesh that contacts the surface has a $Y^{+}<1$ dimension, and the applicability of a low Reynolds number turbulence model was considered.

2) Total heat transfer coefficient of the thermal manikin under forced convection conditions

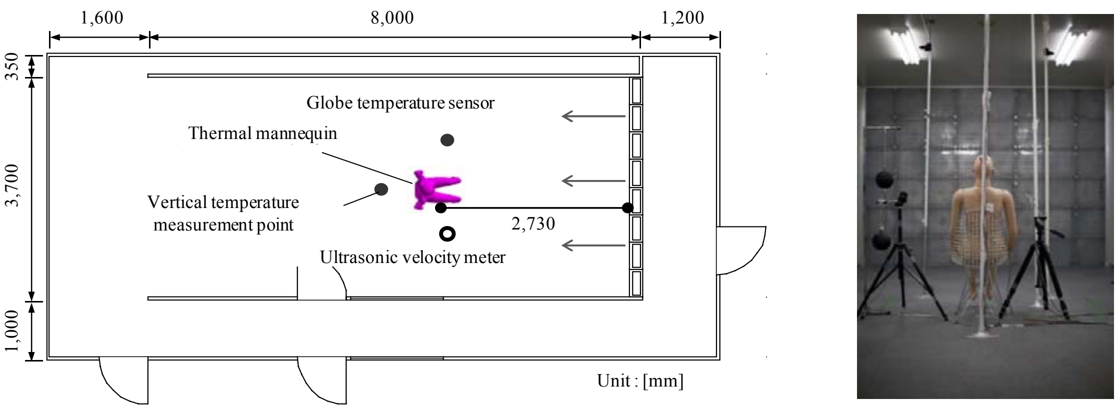
• Experimental evaluation

Figure 2(a) shows a test chamber for measuring the total heat transfer from the body in a wind environment, and Figure 2(b) shows the test situation. The test chamber is such that the convection flows to the front of the thermal manikin. The measurement conditions are shown in Table 1. The manikin is in a seated position and we measured the differences in heat loss transfer from changes in the thermal manikin surface temperature as well as changes in convection flow within the test chamber. With a wall surface air temperature of 28ºC and the thermal manikin surface temperature set to 35ºC, we calculate the total heat



(a) Experimental chamber *(b) Thermal manikin* *(c) CFD model*

Figure 1 Experimental chamber and CFD model on natural convection condition



(a) Experimental chamber and CFD model *(b) Thermal manikin*

Figure 2 Experimental chamber and CFD model on forced convection condition

Table 1 Boundary condition on forced convection condition

Item	Contents
Posture of thermal manikin	Seated posture
Surface temperature of thermal manikin	35.0 ºC
Wall temperature	28.0 ºC
Inlet	0.25, 0.50, 0.75, 1.00 m/s
Outlet	0.0 Pa

transfer from the internal convection changes in the test room. Additionally, we compare an equivalent temperature (see Equations 1 and 2) that provides identical sensible heat loss under a windless environment and an operative temperature under actual conditions. The concept of the equivalent temperature was introduced by Dufton in 1932 [Dufton (1932)] when he studied the heating of buildings. The equivalent temperature is defined as the temperature of an imaginary enclosure with the mean radiant temperature equal to air temperature and still air in which a person has the same heat exchange by convection and radiation as in the actual conditions [Madsen (1984), Nilsson (1999, 2004)].

$$h_0 \cdot (T_s - T_{eq}) = h_t \cdot (T_s - OT) \quad (1)$$

$$T_{eq} = T_s - \frac{h_t}{h_0} \cdot (T_s - OT) \quad (2)$$

where, T_s is the surface temperature of thermal manikin [ºC], T_{eq} is the equivalent temperature [ºC], OT is the operative temperature [ºC], h_0 is the total heat transfer coefficient on natural convection condition [W/(m^2·K)], h_t is the total heat transfer coefficient on actual condition [W/(m^2·K)].

• Examination from CFD analysis

We elicit heat loss from a thermal manikin under conditions similar to the test, conducting a comparison with the experiment values. In a high speed convection environment, the weaknesses of turbulence models become easily appearant. We conducted a comparison of the AKN k-ε turbulence model and the shear-stress transport k-ω turbulence model (SST k-ω turbulence model). The SST k-ω turbulence model was developed by Menter [Menter (1993, 1994)] to effectively blend the robust and accurate formulation of the k-ω model in the near-wall region with the free-stream independence of the k-ε model in the far field. This SST k-ω turbulence model is generally recommended for high accuracy boundary layer simulations.

3) Examination of the differences in heating systems through CFD analysis

It is clear that thermal manikin heat loss can be accurately predicted using CFD analysis under windless and wind environments. We discuss the heat transfer characteristics and secondary input energy for homes with different heating systems. The subject models we undertake are homes with installed hot water floor systems and air conditioning heaters, the ventilation frequency is 0.5 times/h, outside air temperature is 5ºC, floor surface area is 20.8 m^2, and the air volume is 50.0 m^3. Figure 3 shows the analysis model and Table 2 shows the CFD analysis conditions. The SST k-ω turbulence model is used as the turbulence model of CFD analysis. And, we examine heat transfer characteristics from the presumed air condition and floor heating differences for input energy comparison and a sweat covered body (0.75 clo is presumed, as well as an estimate clothed average surface temperature of 28.6ºC). Additionally, we calculate the thermal manikin site specific total heat transfer by assuming the globe temperature room to be the action temperature using Equation (3).

$$h_t = h_r + h_c = q / (T_s - T_g) \quad (3)$$

where, q is the sensible heat loss from thermal manikin [W/m^2], T_g is the globe temperature [ºC], h_r is the radiant heat transfer coefficient [W/(m^2·K)], h_c is the convective heat transfer coefficient [W/(m^2·K)].

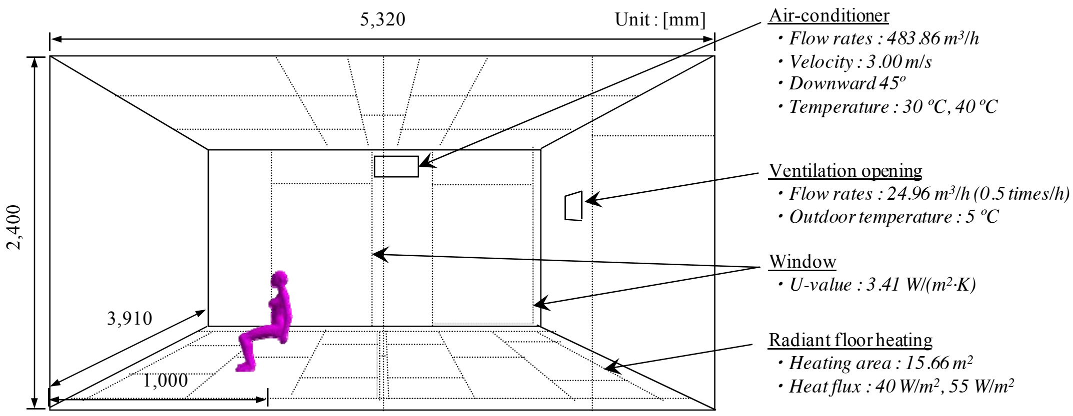


Figure 3 CFD model to compare with air-conditioning heating and radiant floor heating system

Table 2 Boundary condition on CFD calculation by comparison of heating system

Item	Contents
Surface temperature of thermal manikin	28.6 ºC
Outdoor temperature	5.0 ºC
Inlet temperature of air-conditioner heating	30, 40 ºC
Heat flux of radiant floor heating	40, 55 W/m^2
Ventilation	0.5 times/h

RESULTS

1) Total heat transfer coefficient of the thermal manikin under natural convection conditions

• Examination results from actual measurments

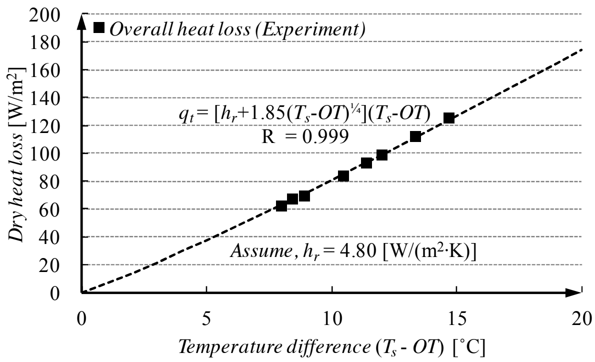
Under a windless environment, sensible heat loss from a thermal manikin with differing temperatures and total heat transfer changes are shown in Figure 4. Within the scope of the experiment, we discern an approximation formula (Equation 4) that is proportional to sensible heat loss with a temperature difference. Here, we assume the radiant heat transfer coefficient h_r is 4.8 W/(m^2·K).

• Reliability evaluation of CFD analysis

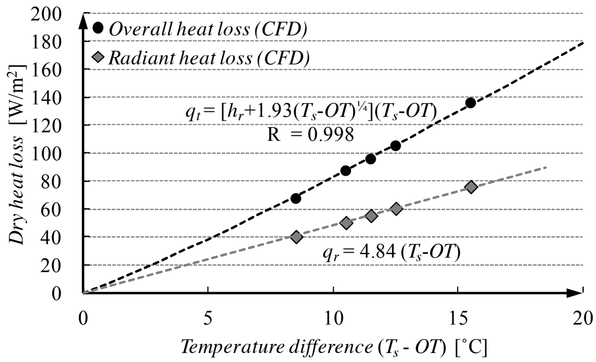
Figure 4(b) shows the heat loss results from the thermal manikin in a windless environment using CFD analysis. We show the heat transfer coefficient using an approximation formula (Equation 5) for the

$$q = (h_r + h_c)\cdot(T_s - OT) \approx [h_r + 1.85(T_s - OT)^{1/4}]\cdot(T_s - OT) \tag{4}$$

$$q = (h_r + h_c)\cdot(T_s - OT) \approx [h_r + 1.93(T_s - OT)^{1/4}]\cdot(T_s - OT) \tag{5}$$



(a) Experimental results



(b) CFD results

Figure 4 Results of sensible heat loss from thermal manikin on natural convection condition

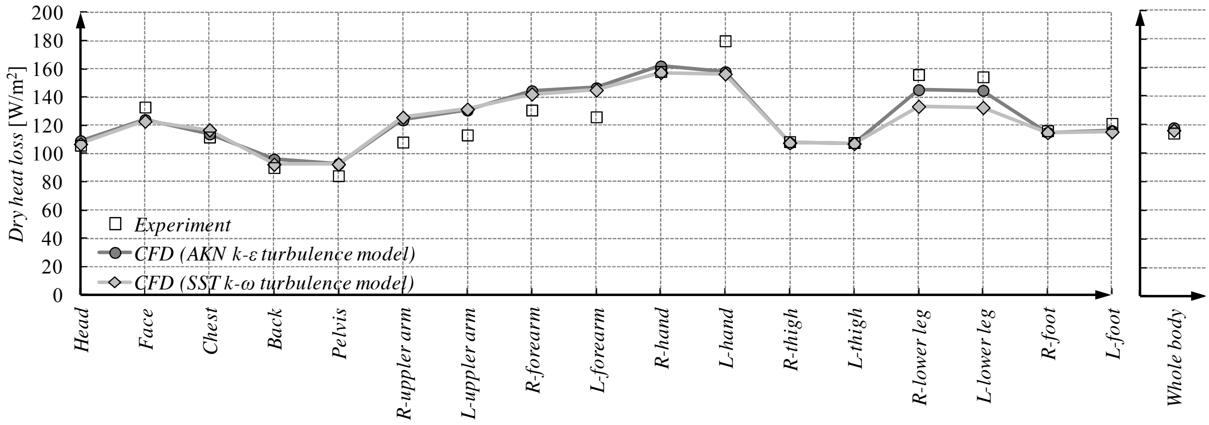
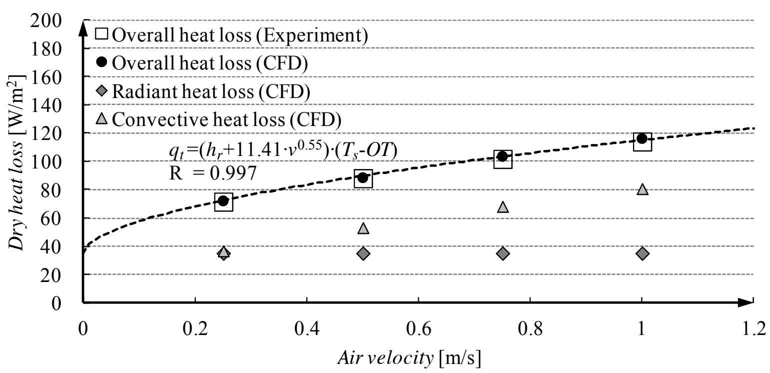
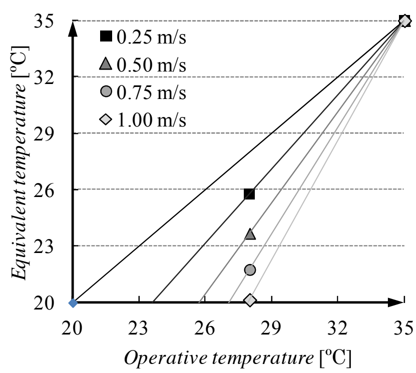


Figure 5 Results of sensible heat loss from thermal manikin on forced convection condition (T_s=35 ºC, v=1.0 m/s)



(a) Variation of sensible heat loss from thermal manikin



(b) Variation of equivalent temperature

Figure 6 Results of sensible heat loss and equivalent temperature on forced convection condition (T_s=35 ºC) relationship between heat loss and temperature difference. When looking at the convection heat transfer coefficient, for the experiment it is h_c=1.85(T_s-OT)$^{0.25}$ and for analysis it is h_c=1.93(T_s-OT)$^{0.25}$. From these results, when the temperature difference is 10°C, experiment values and CFD analysis discrepancy is less than 2.3% (it is less than 0.8% for radiation heat transfer coefficient). We verified the analysis reliability.

2) Total heat transfer coefficient of the thermal manikin under forced convection conditions

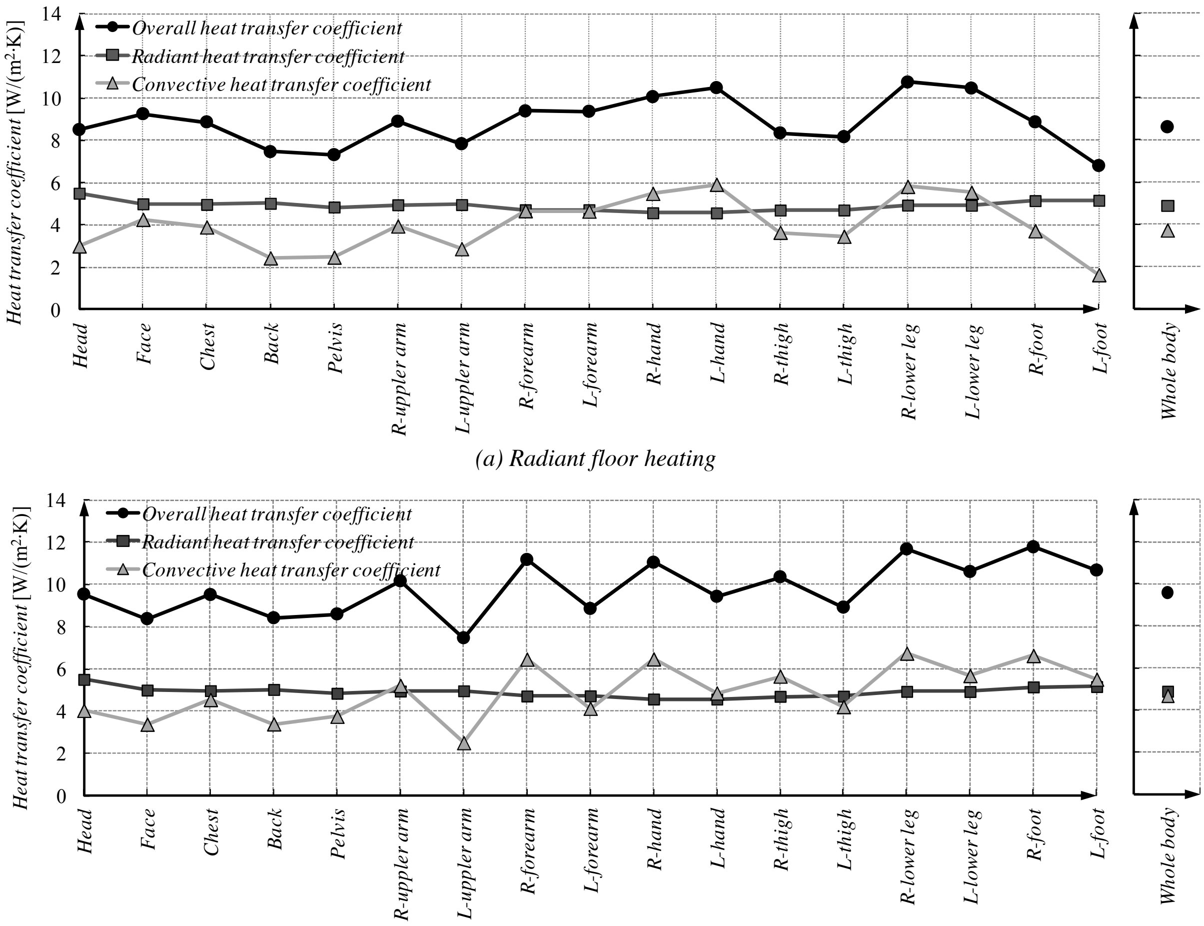
Figure 5 shows the site specific total heat transfer for a 35°C thermal manikin with a convection flow speed of 1.0 m/s in a wind environment. From these results, we compare both the SST k-ω turbulence model and the AKN k-ε turbulence model and find that both models generally match experiment values. Figure 6(a) shows the total heat transfer coefficient change in air current speed around a thermal manikin. We can see that convection heat loss is facilitated when the surrounding air current becomes faster. We see that this relationship is proportional as seen in the approximation formula (Equation 6). Additionally, CFD analysis results are proportional as in the approximation formula (Equation 7) and agree well with the experiment values. Operative temperature relationships are shown in Figure 6(b) for values obtained under a windless environment when converted into equivalent temperature using the equivalent temperature calculation. We can quantitatively assess the sensory temperature (= windless equivalent temperature) changes in response to air flow around the body.

3) Examination of heating system differences from CFD analysis

In Figure 7, the CFD analysis results from the calculation method in Equation 3 regarding heating systems for the thermal manikin surface site specific total heat transfer coefficient is shown. When viewed from the thermal manikin (whole body) the total heat transfer coefficient results were:

$$q = (h_r + h_c)\cdot(T_s - OT) \approx [h_r + 11.44(v)^{0.55}]\cdot(T_s - OT) \tag{6}$$

$$q = (h_r + h_c)\cdot(T_s - OT) \approx [h_r + 11.41(v)^{0.55}]\cdot(T_s - OT) \tag{7}$$



(a) Radiant floor heating

(b) Air-conditioning heating

Figure 7 Heat transfer coefficient for each part of the thermal manikin (q=45 W/m^2)

in the case of floor heating, it was 8.62 W/(m^2·K); in the case of air conditioning heating, it was 9.58 W/(m^2·K); in the case of radiant heating, both were an equal 4.89 W/(m^2·K). On the other hand, convective heat transfer coefficient in the case of floor heating was 3.73 W/(m^2·K) and air conditioning heating was 4.69 W/(m^2·K). From this, we understand that air conditioning heating convective heat transfer coefficient becomes greater than that of floor heating. We assume from these results that radiant heating inhibits thermal manikin heat loss effectively and under an air conditioning heating environment, the increased heat transfer coefficient around the thermal manikin promotes heat loss.

DISCUSSION

1) Discussion relating to the thermal manikin surface total heat transfer coefficient under a wind free environment

The average Nusselt number over the entire surface can be determined from Equation (8) for an isothermal sphere [Churchill (1983)], such as a globe temperature sensor. Therefore total heat transfer coefficient of the 15cm diameter's globe temperature sensor has the correlation to Equation (10) from Equations (8) and (9).

$$\mathrm{Nu} = 2 + \frac{0.589 \cdot \mathrm{Ra}_\mathrm{L}^{1/4}}{[1 + (0.469/\mathrm{Pr})^{9/16}]^{4/9}} = \frac{h_c \cdot L}{k} \tag{8}$$

$$\mathrm{Ra}_\mathrm{L} = \mathrm{Gr}_\mathrm{L} \cdot \mathrm{Pr} = \frac{g\beta(T_s - OT)L^3}{\nu\alpha} \tag{9}$$

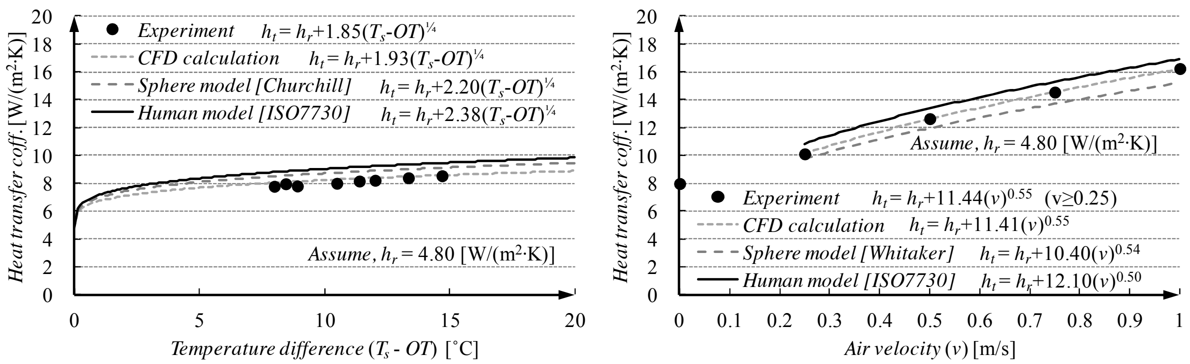
$$h_t = h_r + h_c \approx h_r + 2.20(T_s - OT)^{0.25} \tag{10}$$

where, Nu is the Nusselt number, Ra is the Rayleigh number, Gr is the Grashof number, g is the gravitational acceleration [m/s^2], β is the coefficient of volume expansion [1/K], L is the characteristic length of the geometry [m], ν is kinematic viscosity [m^2/s], α is the thermal diffusivity [m^2/s].

The total heat transfer coefficient over the entire surface can be determined from Equation (11) with temperature difference for a human body [ISO7730 (2005)].

$$h_t = h_r + h_c \approx h_r + 2.38(T_s - OT)^{0.25} \tag{11}$$

In contrast, from this paper as well as CFD analysis, the approximation formulas (Equations 12 and 13) show total heat transfer coefficients under a wind free environment obtained from the examination results. The approximation formulas are as shown in Figure 8(a). Total heat transfer coefficient from the



(a) Relationship between heat transfer coefficient and temperature difference on natural convection condition

(b) Relationship between heat transfer coefficient and air velocity on forced convection condition

Figure 8 Discussion about total heat transfer coefficient of the thermal manikin

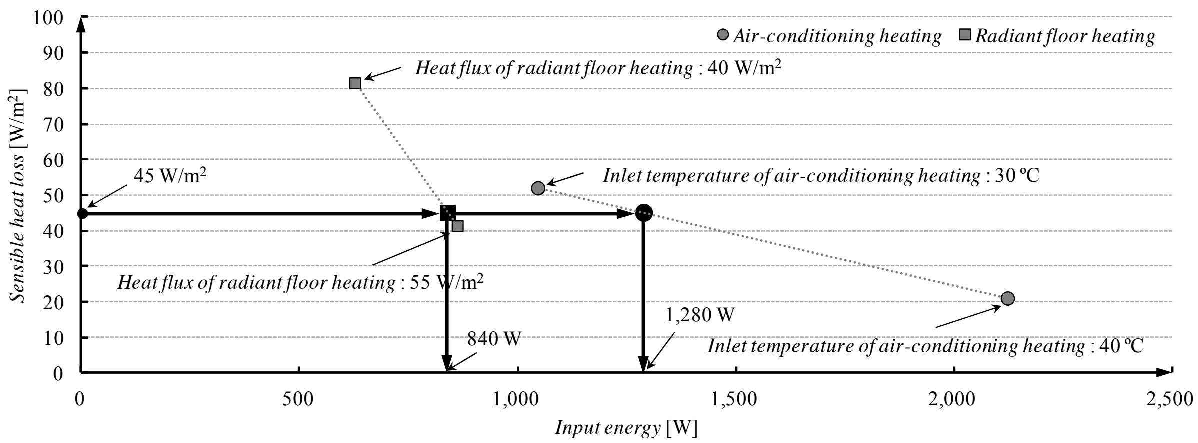


Figure 9 Comparing input energy with heating system

ISO7730 standard body showed even lower values than the values of the spherical total heat transfer coefficient brought forward by Churchill, but we obtained similarly increasing results in accordance with differential heat increase.

$$h_t = h_r + h_c \approx h_r + 1.85(T_s - OT)^{0.25} \quad (12)$$

$$h_t = h_r + h_c \approx h_r + 1.93(T_s - OT)^{0.25} \quad (13)$$

2) Discussion of the total heat transfer coefficient of the thermal manikin under forced convection conditions

Heat transfer around a sphere can be well modeled by the approximation formula for $3.5 < Re_D < 7.6 \times 10^4$, $0.7 < Pr < 380$, and $1 < (\mu_\infty/\mu_s) < 3.2$ given by Equations (14) and (15) [Whitaker (1972)].

$$\mathrm{Nu} = 2 + [0.4\,\mathrm{Re}^{1/2} + 0.06\,\mathrm{Re}^{2/3}]\,\mathrm{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4} \quad (14)$$

$$h_t = h_r + h_c \approx h_r + 10.40(v)^{0.54} \quad (15)$$

The total heat transfer coefficient over the entire surface on the forced convection condition can be determined from Equation (16) with air velocity for a human body [ISO7730 (2005)].

$$h_t = h_r + h_c \approx h_r + 12.10(v)^{0.50} \quad (16)$$

And, the approximation formulas obtained from air current speed rates from the experiment in this paper as well as CFD analysis is shown in Equations (17) and (18). They are as given in Figure 8 (b), but we see it is a smaller value than the total heat transfer coefficient from the ISO7730 standard body, and a value larger than the spherical heat transfer coefficient brought forward by Whitaker.

$$h_t = h_r + h_c \approx h_r + 11.44(v)^{0.55} \quad (17)$$

$$h_t = h_r + h_c \approx h_r + 11.41(v)^{0.55} \quad (18)$$

3) Discussion of input energy by heating system

Figure 9 shows the input energy and thermal manikin heat loss comparison results from heating systems. With this, to conduct a suitable comparison, when the underfloor heat loss is not included, the predicted mean vote (PMV) shows a heat loss result of 45 W/m from a 0 corresponding body. When comparing the input energy calculated using CFD analysis, air conditioning heating is approximately 1,280 W and floor heating is 840 W, showing floor heating input energy is smaller. From these results, to create a similarly warmer environment, it is believed that the floor heating will have an energy saving of approximately 34.4%.

CONCLUSION

In this study, the airflow, temperature and heat transfer coefficients were examined in a room model that included a sitting thermal manikin. A summary of the general findings from this study is as follows.

- From the experiment, we see that body surface total heat transfer coefficient under a windless environment corresponds to Equation 12 and that under a wind environment corresponds to Equation 17.
- CFD analysis consistency was confirmed through comparison with the experiment under a windless as well as a wind environment.
- From comparison of equivalent temperature and operative temperature, we understood that when air speed currents around a body increase, operative temperature decreases.
- When creating a similarly warm environment, secondary input energy is lower for floor heating than that of air conditioning heating, an energy saving effect of approximately 34.4%.

FUTURE PERSPECTIVE

Further study is required to evaluate various thermal manikin postures such as standing, sitting, lying down, and the effect of body movement by transient calculation. Moreover, thermal comfort should be measured for an appropriate room by considering the effects of any cold drafts and the energy-saving effects of various heating and cooling systems in residential buildings.

NOMENCLATURE

T_s = *Surface temperature of thermal manikin* [ºC]
T_{eq} = *Equivalent temperature* [ºC]
T_g = *Globe temperature* [ºC]
OT = *Operative temperature* [ºC]
h_c = *Convective heat transfer coefficient* [W/(m^2·K)]
h_r = *Radiant heat transfer coefficient* [W/(m^2·K)]
h_0 = *Total heat transfer coefficient on natural convection condition* [W/(m^2·K)]
h_t = *Total heat transfer coefficient on actual condition* [W/(m^2·K)]
q = *Sensible heat loss from thermal manikin* [W/m^2]
Nu = *Nusselt number* [-]
Ra = *Rayleigh number* [-]
Gr = *Grashof number* [-]
Pr = *Prandtl number* [-]
Re = *Reynolds number* [-]
g = *Gravitational acceleration* [m/s^2]
β = *Coefficient of volume expansion* [1/K]
L = *Characteristic length of the geometry* [m]
ν = *Kinematic viscosity* [m^2/s]
α = *Thermal diffusivity* [m^2/s]
μ = *Dynamic viscosity* [m^2/s]
v = *Air velocity* [m/s]

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