THERMAL STORAGE WITH CONCRETE SLAB OF PRESSURIZED PLENUM IN UNDERFLOOR AIR DISTRIBUTION SYSTEM

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ABSTRACT

In calculating the heat flows around the floor plenum of underfloor air distribution system, the convective heat transfer coefficient is an influential factor, but it is not clearly known which value should be taken. The convective heat transfer coefficient was measured with the airflow velocity, and the relationship between them was clarified, involving the airflow and surface temperatures. Another set of measurements was made with a full-scale experimental rig of floor plenum, and the measurement result was analyzed with newly proposed "1-dimensional model" and "Developed 1-dimensional model", both of which were shown to predict well the heat flows and temperatures, using the above-mentioned convective heat transfer coefficient relationship.

INTRODUCTION

In the underfloor air distribution system of pressurized floor plenum type, conditioned air is first blown into the floor plenum and flows over the concrete slab, exchanging heat with the slab, before coming into the room through floor-mounted air outlets. This heat exchange makes the outlet air temperatures different from one another, and means that the slab stores cooling and heating energy. The outlet air temperatures are preferably to be equalized and the slab may be used effectively as a heat-storing medium. The objective of this study is to make a calculation method which can easily quantify the thermal effect of the slab.

EXPERIMENTAL RIG

Figure 1 shows the whole system of a full-scale experimental rig of floor plenum. A 150 mm thick concrete slab and 600 mm x 600 mm raised floor panels form a floor plenum with a void 150 mm high. Air at a controlled temperature is introduced into the floor plenum, blown up through 9 floor-mounted air outlets of swirl type, and drawn back to the air handling unit. The temperature of the space under the slab is also controlled with another air-conditioning system. Figure 2 shows a 6.8 m x 5.8 m floor plan with measurement locations of temperature and heat flux called "Aa" to "Ie". A 1000 mm wide supply air duct connection is also seen at a corner of the plenum.

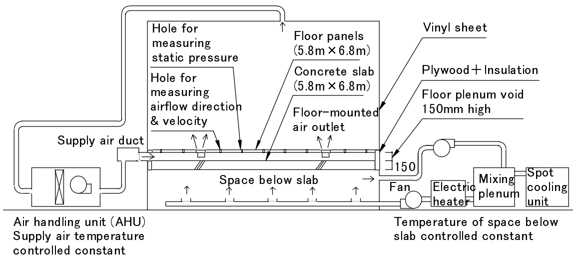

Figure 1 Experimental system of floor plenum

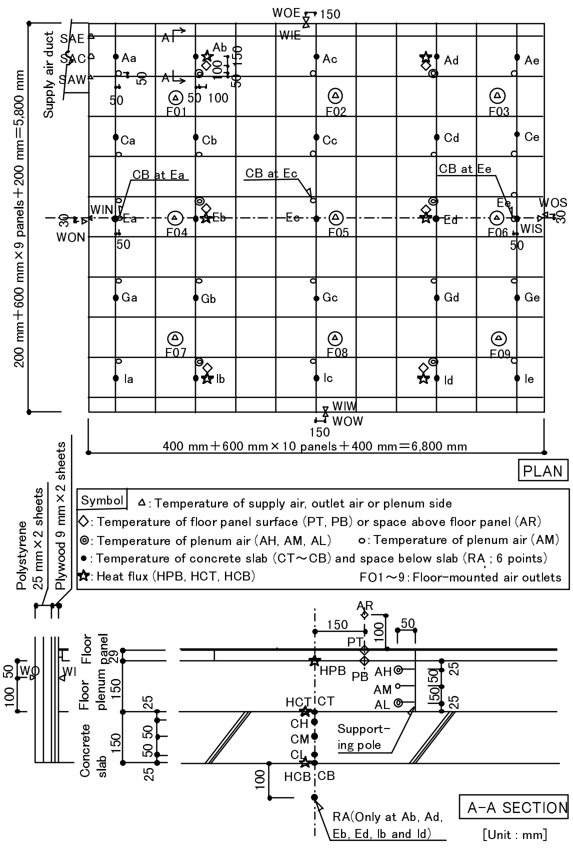

Figure 2 Plan & section of floor plenum

FORCED CONVECTION HEAT TRANSFER COEFFICIENT

Measurement Methods

Temperature, heat flux and airflow velocity were measured at measurement locations Ab and Ad, after the floor plenum was brought in a thermally steady state under several temperature and velocity conditions. As shown in Figure 3 for measurement location Ab, air velocity was measured at 15 mm vertical intervals, giving a velocity profile.

The underside of the floor panel was covered with galvanized iron sheet, and considered to have an emmissivity much less than 1. Therefore the thermal radiant energy reflected on its surface was measured with a thermal camera, resulting in 0.55 for the emmissivity on the floor panel underside. Because the floor panel underside and the slab upper surface form parallel flat surfaces, the radiative heat flux between them is given by

$$q = \sigma \ (T_p^4 - T_c^4) / ((1/\varepsilon_p) + (1/\varepsilon_c) - 1) \qquad (1)$$

where the emmissivity of the slab upper surface (ε_c) is assumed to be 0.88.

The measured conductive heat flux plus the above-mentioned radiative heat flux makes the convective heat flux from the slab upper surface to the air. Division of it by their temperature differential gives the conductive heat transfer coefficient on the slab upper surface. The airflow temperature was measured at the middle of the plenum void height, i.e. 75 mm above the slab upper surface.

Results

Figure 4 shows the measured conductive heat transfer coefficient on the slab upper surface in relation to the airflow velocity at the middle of the plenum void height, and includes Jurges's equation[3] given by

$$h_c = 3.95\ v + 5.8 \quad (\text{for } v<5\text{m/s}) \qquad (2)$$

For velocities above 2 m/s, the measured convective heat transfer coefficient follows Jurges's equation, but comes under it for velocities below 2 m/s, and becomes the lowest in the case of cool supply air, i.e. the airflow temperature lower than the slab surface temperature. This temperature-depending difference in convective heat transfer coefficient is presumably accounted for by a difference in velocity of airflow adjacent to the slab surface.

The airflow velocity profile is assumed to be expressed as

$$v = C\ z^{1/n} \qquad (3)$$

Substituting the measured velocities at 15 mm and 30 mm above the slab surface into Equation (3) gives values to C and n, and consequently enables to calculate an airflow velocity at any location close to the surface.

Figure 5 shows the measured convective heat transfer coefficient in relation to the calculated airflow velocity at 1 mm above the surface. The differences due to temperatures become much less than seen in Figure 4. The ratio of the airflow velocity at 1 mm above the surface to that at the middle of the plenum void is expected to relate to the velocity itself and the temperature differential between the airflow and the surface, both of which can be combined into Archimedes number (Ar). Note that Archimedes number is so defined that it is minus when the temperature differential is minus. Figure 6 implies that the velocity ratio can be approximated by a linear expression of Archimedes number. Combining Figures 5 and 6 gives

$$h_c = 10.3\ (1.45\ Ar + 0.52)\ v_m + 1.54 \qquad (4)$$

for the airflow velocity at the middle of the plenum void height below 3 m/s, above which Jurges's equation is applicable.

Discussion

Several equations have been proposed for the convective heat transfer coefficient on a flat plate exchanging heat with air in contact, as listed in Table 1 on the last page. According to the equations for forced convection, the convective heat transfer coefficients were calculated and found different from one another, as shown in Figure 7, which includes Equation (4) with Archimedes number of 0 as well as CIBSE Guide's equations with several distances from the leading edge. ASHRAE Handbook's equation is very close to Jurges's equation.

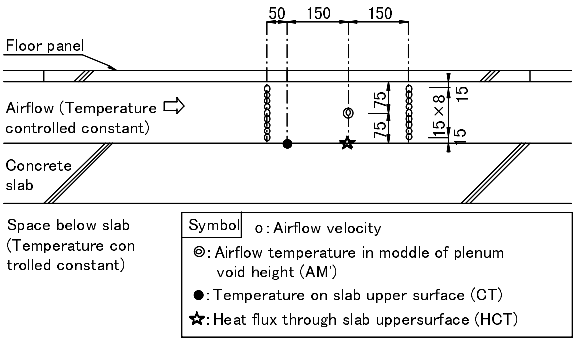

Figure 3 Section of convective heat transfer coefficient measurement at Ab

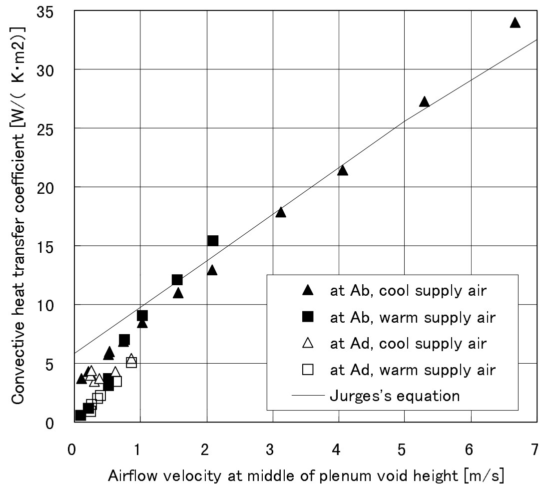

Figure 4 Measured convective heat transfer coefficient in relation to airflow velocity at middle of plenum void height

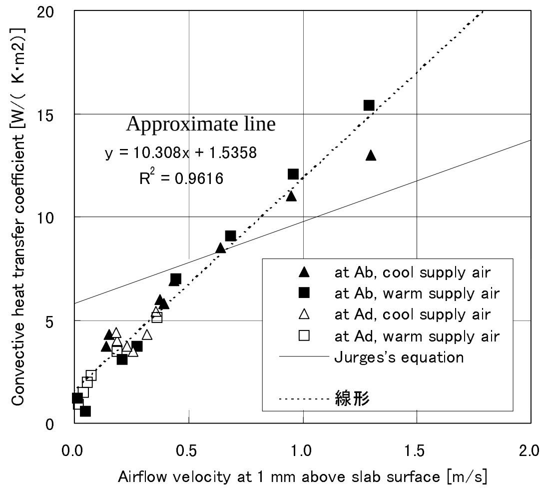

Figure 5 Measured convective heat transfer coefficient in relation to airflow velocity at 1 mm above slab surface

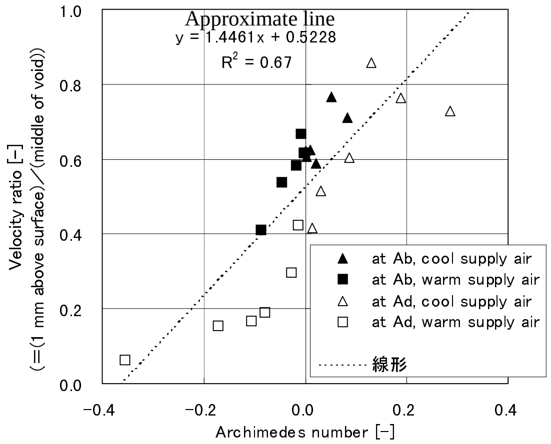

Figure 6 Velocity ratio in relation to Archimedes number

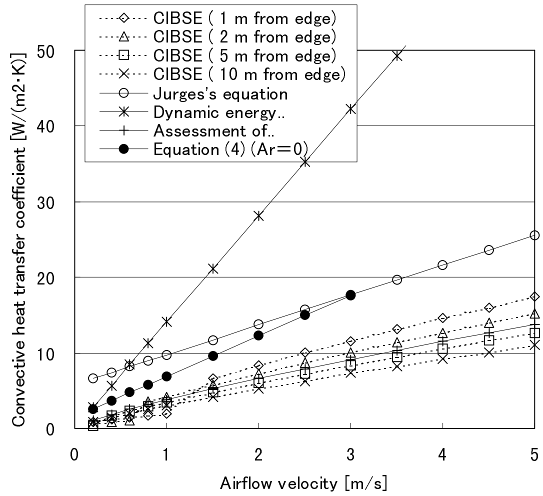

Figure 7 Convective heat transfer coefficients from references

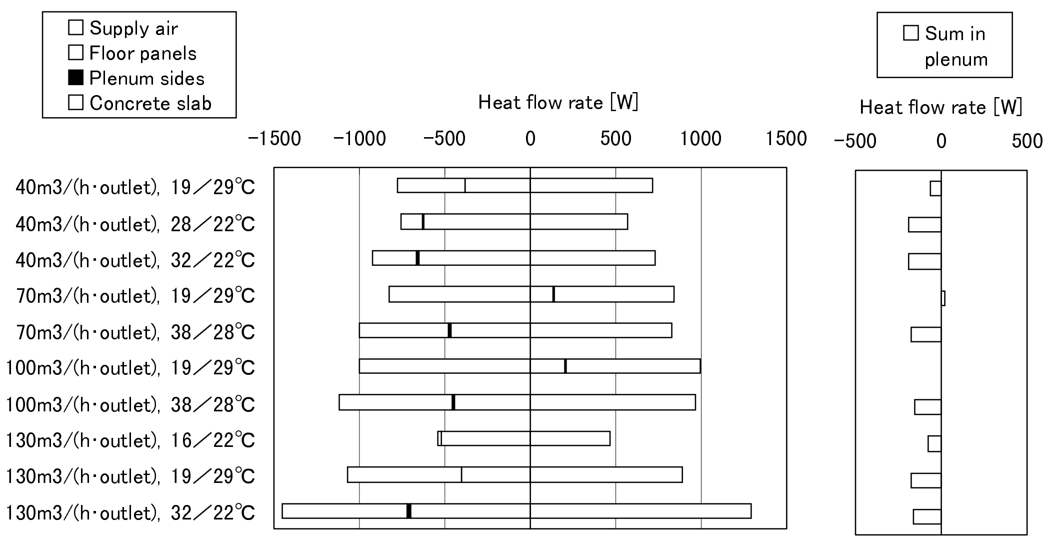

Figure 8 Measured heat flow balance in floor

MEASUREMENT IN WHOLE PLENUM

Experimental Conditions

The measurements were made in the whole floor plenum, under various conditions of supply air volume flow rate and temperature, as listed in Table 2. All these measurements can be grouped into steady state measurements and transition ones.

Airflow

By comparing the measured airflows in a non-isothermal steady state with that in an isothermal state, it is found that the corresponding airflow directions are almost the same, and that the velocities vary in proportion to the supply air volume flow rate.

Conductivity

Plotting the measured heat fluxes with regard to the temperature differentials across the floor panel or slab in the steady state measurements shows a linear relationship between them. Its slope indicates the conductivity of the floor panel or slab at 0.6W/(K m) or 1.9W/(K m) respectively.

Heat Flow Balance

In a steady state measurement, a sum of the heat flow rates through all the inner surfaces of the plenum should be 0, and this heat flow balance was checked in Figure 8, where the heat flow rates through the floor panel and slab were calculated by multiplying the measured heat fluxes by the corresponding areas. The heat flow sums are not necessarily close to 0, and these errors occur probably because some measured heat fluxes are far from the averages for the corresponding areas.

ANALYSIS WITH MODELS

1-Dimensional Model

"1-dimensional (1D) model" is for quantifying the heat flow balance in the plenum, and consists of horizontal layers of floor panel, plenum and slab, as presented in Figure 9. Each layer has one representative temperature, except the plenum where air is assumed to flow unidirectionally, as shown in Figure 10, changing its temperature. This assumption gives

$$z_v\, v_v / x_v = V_s / A \qquad (5)$$

The heat flow balance in each layer gives an equation in Table 3.

In 1D model, the convective heat transfer coefficients on the floor panel and slab surfaces are so adjusted that the heat flow rates through all the plenum inner surfaces are balanced. This adjustment is necessary to offset the difference in convective heat transfer coefficient between the assumed unidirectional airflow and the real one. The adjusted coefficient is called "1D convective heat transfer coefficient" .

Table 2 Experimental conditions

	Average supply air volume flow rate per air outlet [m^3/(h outlet)]	Temperature (Supply air / Space below slab) [C]
Steady state measurement	40	19 / 29
	40	28 / 22
	40	32 / 22
	70	19 / 29
	70	38 / 28
	100	19 / 29
	100	38 / 28
	130	16 / 22
	130	19 / 29
	130	32 / 22
Transition measurement	130-->Stop	16 / 22
	130-->Stop	32 / 22
	130	(22-->16) / 22
	130	(22-->32) / 22
	40	(22-->28) / 22

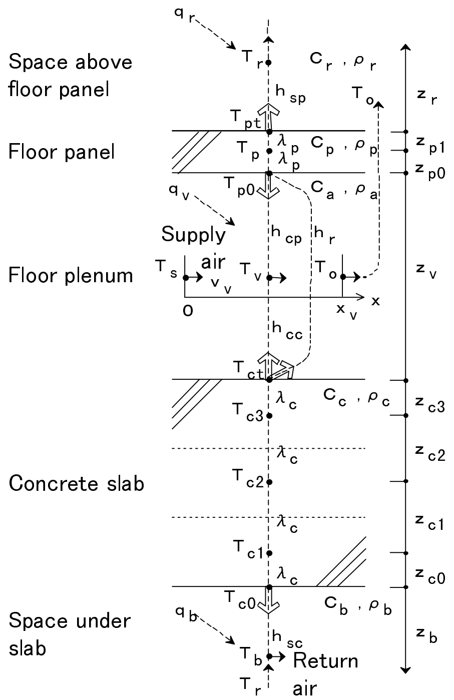

Figure 9 1-dimensional model configuration

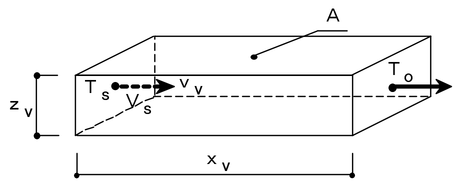

Figure 10 Unidirectional airflow assumed in 1-dimensional model

Developed 1-Dimensional Model

1D model deals with space-average temperatures and cannot express horizontal temperature variations, which can be handled with "Developed 1-dimensional (D1D) model". The plan of floor plenum is divided into small areas such as floor panels. 1D model is applied to each small area, which is connected to the adjacent small areas with the airflow temperatures, as seen in Figure 11, making up D1D model as a whole.

D1D model requires all the volume flow rates through the boundary surfaces of small areas, which can be calculated by computational fluid dynamics (CFD). Note that, for a good calculation result, all the poles supporting the floor panels should be incorporated into CFD mesh.

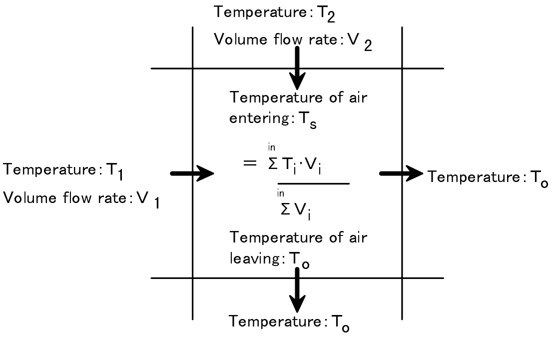

Figure 11 Airflow temperature connection in Developed 1-dimensional model

Natural Convection Heat Transfer Coefficient

Table 1 also presents several equations for the natural convection heat transfer coefficient. For heat flow downwards, the convective heat transfer coefficient seems 0, as indicated in Figure 4 and proposed by Reference [6].

For heat flow upwards, each equation in Table 1 was applied to 1D model, which calculated the temperature of the plenum air with the measured temperatures of the floor panel and slab surfaces in the case of transition measurements with natural convection. As a result, M. ten Bosch's equation showed the best agreement between the calculated and measured plenum air temperatures, and was adopted for the natural convection heat transfer coefficient.

The natural and forced convection heat transfer coefficients should be continuously connected so that the heat flow calculation converges. It is proposed, as shown in Figure 12, that a straight line connects the natural convection heat transfer coefficients at an airflow velocity of 0 and the forced ones at the following velocity:

Table 3 Heat balance equations in 1-dimensional model

Location (Air supply to plenum)	Heat balance equation
Space above floor panel (Operating)	$c_r \rho_r z_r \frac{\partial T_r}{\partial t} = c_a \rho_a \frac{V_s}{A}(T_a - T_r) + h_{sp}(T_{pt} - T_r) + q_r$
Ditto (Not operating)	$c_r \rho_r z_r \frac{\partial T_r}{\partial t} = h_{sp}(T_{pt} - T_r) + q_r$
Upper surface of floor panel	$h_{sp}(T_{pt} - T_r) = \frac{\lambda_p}{z_{p1}}(T_p - T_{pt})$
In floor panel	$\frac{\partial T_p}{\partial t} = \frac{\lambda_p}{c_p \rho_p} \cdot \frac{\partial^2 T_p}{\partial z_p^2}$
Underside of floor panel	$h_{cp}'(T_{p0} - T_v) = \frac{\lambda_p}{z_{p0}}(T_p - T_{p0}) + h_r(T_{ct} - T_{p0})$
In floor plenum (Operating)	$c_a \rho_a z_v v_v \frac{\partial T_v}{\partial x} = h_{cp}'(T_{p0} - T_v) + h_{cc}'(T_{ct} - T_v) + q_v$
Ditto (Not operating)	$c_v \rho_v z_v \frac{\partial T_v}{\partial t} = h_{cp}'(T_{p0} - T_v) + h_{cc}'(T_{ct} - T_v) + q_v$
Upper surface of concrete slab	$h_{cc}'(T_{ct} - T_v) = \frac{\lambda_c}{z_{c3}}(T_{c3} - T_{ct}) - h_r(T_{ct} - T_{p0})$
In concrete slab	$\frac{\partial T_c}{\partial t} = \frac{\lambda_c}{c_c \rho_c} \cdot \frac{\partial^2 T_c}{\partial z_c^2}$
Underside of concrete slab	$h_{sc}(T_{c0} - T_b) = \frac{\lambda_c}{z_{c0}}(T_{c1} - T_{c0})$
Space below slab (Operating)	$c_b \rho_b z_b \frac{\partial T_b}{\partial t} = c_a \rho_a \frac{V_s}{A}(T_r - T_b) + h_{sc}(T_{c0} - T_b) + q_b$
Ditto (Not operating)	$c_b \rho_b z_b \frac{\partial T_b}{\partial t} = h_{sc}(T_{c0} - T_b) + q_b$

$$v_m = (1.45\ g\ \beta\ (z_v/2)\ |T_{ct} - T_m| / 0.52)^{1/2} \quad (6)$$

Heat Flow Balance With D1D Model

The heat flow rates through the floor panel and slab in the case of steady state measurements were re-calculated by D1D model with the convective heat transfer coefficient equations presented in Figure 12. Figure 13 shows the re-calculation result with less errors than Figure 8.

Convective Heat Transfer Coefficient Ratio

The result of heat flow calculation with 1D model is much affected by the 1D convective heat transfer coefficient. The ratio of it to the convective heat transfer coefficient is called "Convective heat transfer coefficient ratio" (γ), with which the heat flow balance in the plenum is expressed as

$$\gamma\ \ h_{cp}\,(T_{p0}''-T_a'') + \gamma\ \ h_{cc}\,(T_{ct}''-T_a'') + Q_o + Q_s = 0 \quad (7)$$

where the convective heat transfer coefficients on the floor panel underside and slab upper surface are calculated from the space-averages of airflow velocity, airflow temperature and surface temperature.

The measured steady state temperatures of the floor panel underside and slab upper surface were given to 1D model to calculate the 1D convective heat transfer coefficients and the convective heat transfer coefficient ratios. Figure 14 shows that the calculated 1D convective heat transfer coefficients increase with the supply air volume flow rate. Figure 15 shows that the calculated convective heat transfer coefficient ratios are almost constant as far as the supply air volume flow rate ranges from 70 to 130 m^3/(h outlet), regardless of the temperature differential. This means that, with a certain convective heat transfer coefficient ratio, 1D model can predict the heat flow rates through all the plenum inner surfaces with 2% error. The convective heat transfer coefficient ratio can also be computed by D1D model.

Transition Calculation With 1D Model

1D model can easily simulate temperatures in transition, using the convective heat transfer coefficient ratio calculated from the steady state measurement results as well as the convective heat transfer coefficient equations presented in Figure 12. The measured transition temperatures of the floor panel upper surface and slab underside were given to 1D model, which worked out the outlet air temperature, as shown in Figure 16, in a reasonable accordance with the measured values.

CONCLUSIONS

The forced convection heat transfer coefficient on a surface facing a parallel surface 150 mm away followed Jurges's equation for the velocity at the middle between the surfaces more than 2 m/s. For the velocity below it, the forced convection heat transfer coefficient was expressed well by an equation involving Archimedes number.

"1-dimensional model" and "Developed 1-dimensional model" were proposed. The former can calculate space-average temperatures and heat flow rates in transition, while the latter can express horizontal temperature variations in a steady state. The above-mentioned convective heat transfer coefficient equations were applied to these models, which calculated temperatures and heat flow rates in a reasonable accordance with the measured values.

REFERENCES

[1] The Japan Society of Mechanical Engineers, "Heat transfer handbook", 1986.

[2] The Society of Heating, Air-Conditioning and Sanitary Engineers of Japan, "SHASE Handbook", 1995

[3] Watanabe, K., "Architectural planning fundamentals", 1965

[4] American Society of Heating, Refrigerating and Air-Conditioning Engineers, "ASHRAE Handbook Fundamentals", 1997

[5] The Chartered Institution of Building Services Engineers, "CIBSE Guide Volume C", 1986

[6] Barnard, N., "Dynamic Energy storage in the building Fabric", TR 9/94 a technical report from BSRIA, The Building Services Research and Information Association, 1995

[7] Holmes, M. J. and Wilson, A.., "Assessment of the performance of Ventilated floor thermal storage systems", ASHRAE Transitions Vol.102, Part 1, pp.698-707, 1996

NOMENCLATURE

A	= floor area [m^2]
Ar	= Archimedes number [-]
C	= constant [-]
c	= specific heat capacity [J/(kg K)]
g	= gravitational acceleration [m/s^2]
h_c	= convective heat transfer coefficient [W/(K m^2)]
h_r	= radiative heat transfer coefficient [W/(K m^2)]
h_s	= surface heat transfer coefficient [W/(K m^2)]
n	= constant [-]
Q_o	= heat flow rate due to air temperature differential between entry and exit of plenum [W]
Q_s	= heat flow rate through plenum side surfaces [W]
q	= heat flux [W/m^2]
T	= temperature [C]
T_o	= air temperature at exit of plenum [C]
T_s	= air temperature at entry of plenum [C]
t	= time [s]
V_s	= supply air volume flow rate [m^3/s]
v	= airflow velocity [m/s]
x	= horizontal distance [m]
z	= vertical distance or height [m]
β	= coefficient of thermal expansion [K^{-1}]
γ	= convective heat transfer coefficient ratio [-]
ε	= emmisivity [-]
Δ	= difference between values
λ	= thermal conductivity [W/(K m)]
ρ	= density [kg/m^3]
σ	= Stefan-Boltzmann constant [W/(m^2 K^4)]

SUBSCRIPTS

a	= air
b	= space below slab
c	= concrete slab
m	= at middle of floor plenum void height
p	= floor panel
r	= space above floor panel
t	= upper surface
v	= floor plenum void
0	= underside
'	= of 1-dimensional model
"	= space-average

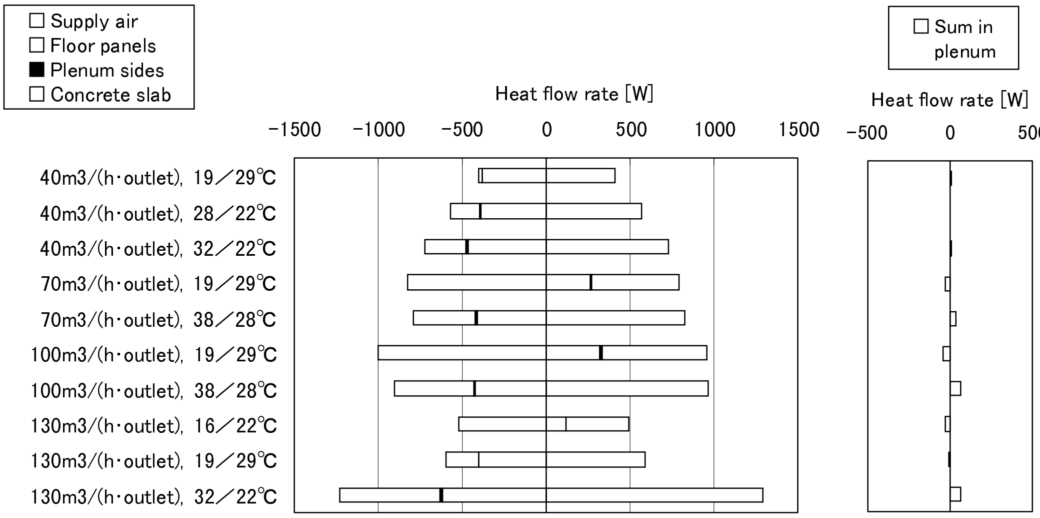

Figure 13 Heat flow balance in floor plenum calculated with Developed 1-dimensional model

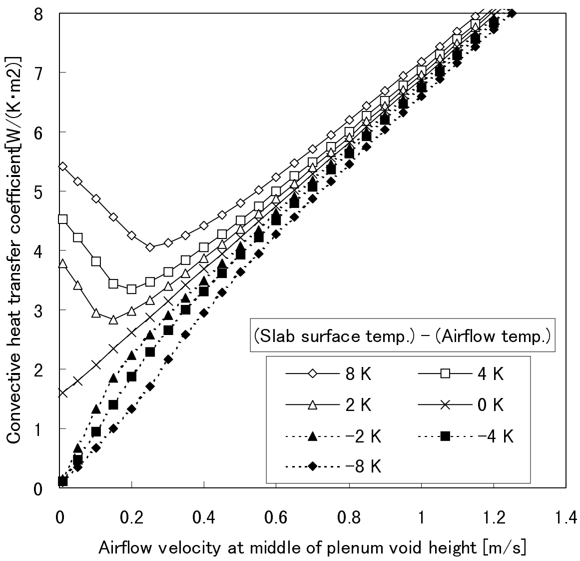

Figure 12 Connection between natural and forced convection heat transfer coefficients

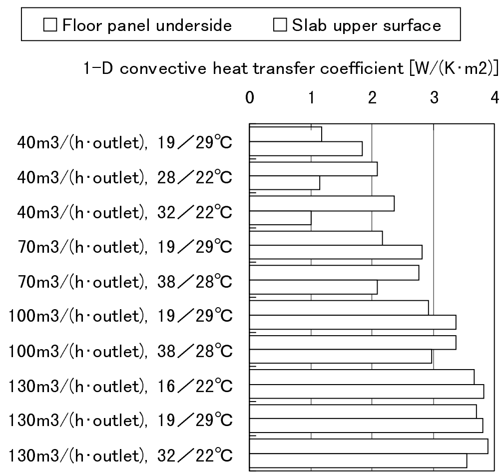

Figure 14 Convective heat transfer coefficient ratio

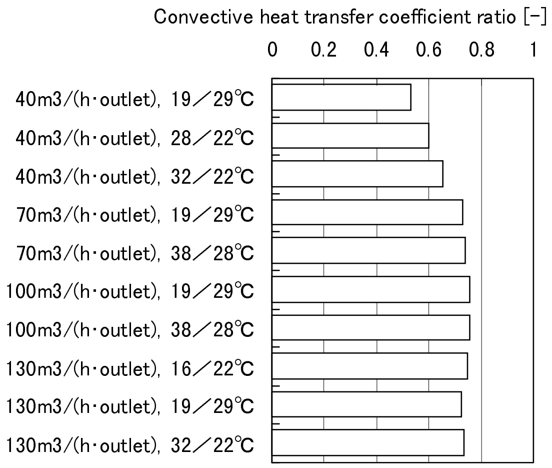

Figure 15 Convective heat transfer coefficient ratio

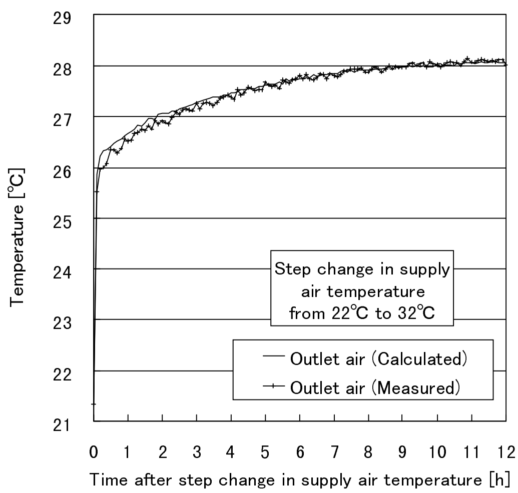

Figure 16 Outlet air temperature in transition calculated with 1-dimensinal model

Table 1 Convective heat transfer coefficient equations from references

References (Authors)	Forced convection over flat plates	Natural convection over horizontal flat plates (Heat flow upwards)	Natural convection over horizontal flat plates (Heat flow downwards)
Heat transfer handbook[1]	At a distance x from the leading edge $Nu_x = 0.332\ Pr^{1/3}\ Re^{1/2}$ applicable to laminar flow if $Re<5x10^5$ $Nu_x = 0.0296\ Pr^{2/3}\ Re^{4/5}$ applicable to turbulent flow if $Re>5x10^5$	For plates in free air $Nu_m = 0.515\ (Gr\ Pr)^{1/4}$ applicable to laminar flow over plates of uniform surface temperature For plates enclosed $Nu_m = 1.65x0.0185\ (Gr\ Pr)^{2/5}$ applicable to turbulent flow if $3x10^{10}<Gr\ Pr<2x10^{11}$	For plates in free air $Nu_m = 0.6\ (Gr\ Pr)^{1/5}$ applicable to laminar flow if $10^6<Gr\ Pr<10^{11}$ For plates enclosed $Nu_m = 0.09x0.0185(Gr\ Pr)^{2/5}$ applicable to turbulent flow if $10^{11}<Gr\ Pr<3x10^{11}$
SHASE Handbook[2]	At a distance x from the leading edge $Nu_x = 0.332\ Re^{0.5}\ Pr^{1/3}$ applicable to laminar flow if $Re<5x10^5$ $Nu_x = 0.0296\ Re^{0.8}\ Pr^{0.6}$ applicable to turbulent flow if $Re>5x10^5$	For a square plate in free air $Nu_m = 0.54\ (Gr\ Pr)^{1/4}$ applicable to laminar flow if $10^5<Gr\ Pr<2x10^7$ $Nu_m = 0.14\ (Gr\ Pr)^{1/3}$ applicable to turbulent flow if $2x10^7<Gr\ Pr<3x10^{10}$	For a square plate in free air $Nu_m = 0.27\ (Gr\ Pr)^{1/4}$ applicable to laminar flow if $3x10^5<Gr\ Pr<3x10^{10}$
Architectural planning fundamentals[3]	**Jurges's equation** $h_c = 5.8+3.95\ v$ applicable to normal roughness, v<5m/s $h_c = 7.14\ v^{0.78}$ applicable to normal roughness, v>5m/s	For plates in free air M.ten Bosch's equation $h_c = 3.26\ (\Delta T)^{1/4}$ Griffiths & Davis's equation $h_c = 2.63\ (\Delta T)^{1/4}$ Jakob & Howkins's equation $h_c = 2.50\ (\Delta T)^{1/4}$	For plates in free air M.ten Bosch's equation $h_c = 1.74\ (\Delta T)^{1/4}$ Griffiths & Davis's equation $h_c = 1.33\ (\Delta T)^{1/4}$ Jakob & Howkins's equation $h_c = 1.31\ (\Delta T)^{1/4}$
ASHRAE Handbook Fundamentals 1997 [4]	For vertical plane surface, v<5m/s $h_c = 5.62+3.9\ v$ For vertical plane surface, v of 5 to 30m/s $h_c = 7.2\ v^{0.78}$	For small plates, laminar range $h_c = 1.32\ (\Delta T/L)^{0.25}$ For small plates, turbulent range $h_c = 1.52\ (\Delta T)^{0.33}$ L：plate length [m]	For small plates $h_c = 0.59\ (\Delta T/L)^{0.25}$
CIBSE Guide Volume C 1986 [5]	At a distance x from the leading edge $Nu_x = 0.33\ Pr^{0.33}\ (Re)_x^{0.5}$ applicable to laminar flow if $(Re)_x<10^5$ $Nu_x = 0.029\ Pr^{0.33}\ (Re)_x^{0.8}$ applicable to turbulent flow if $(Re)_x>10^5$	For freely exposed flat plates $h_c = 1.4\ (\Delta T/D)^{0.25}$ applicable to laminar flow if $1.4x10^5<(Gr)_D<3x10^7$ $h_c = 1.7\ (\Delta T)^{0.33}$ applicable to turbulent flow if $3x10^7<(Gr)_D<3x10^{10}$ D：characteristic plate dimension [m]	For freely exposed flat plates $h_c = 0.64\ (\Delta T/D)^{0.25}$ applicable to laminar flow if $3x10^5<(Gr)_D<3x10^{10}$
Dynamic energy storage in the building fabric (Barnard) [6]	$h_c = \rho_a\ v\ c_a\ f/(2\ Pr^{2/3})$ where f：friction coefficient (=0.0185)	$h_c = 1.7\ (\Delta T)^{0.33}$ derived from CIBSE Guide	For an enclosed surface $h_c = 0$
Assessment of the performance of ventilated floor thermal storage systems (Holmes & Wilson)[7]	For turbulent flow between two parallel plates $h_c = 2.6\ v^{0.8}/d^{0.2}$ where v：average air velocity [m/s] d：distance between the plates [m]		