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PREDICTION OF AIR DISTRIBUTION IN MULTIROOM BUILDINGS

by

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ABSTRACT

A method for predicting room air distribution in a multiroom building is described and some case study results are shown in this paper. Air movements and transient room temperatures in a building are simultaneously calculated by the computer simulation program called "PSSP". The PSSP can calculate various types of air movements, such as natural ventilation, infiltration and forced ventilation.

Airflows through openings and cracks at envelopes and interroom walls are defined as nonlinear equations of the air volume and pressure difference. Air distribution in a building is conducted by solving volume balance equations of the inflows and outflows in rooms. These calculation procedures are described in detail. Air movements, room air temperatures, surface temperatures and air-conditioning loads are connected with each other. The methods to express these complex relations are described, too.

Air distribution is important and essential for thermal performance of buildings, especially for solar houses. The case study results on a solar house executed by the PSSP are shown, and the effectiveness of the methods for predicting room air distribution are discussed.

INTRODUCTION

In the field of building thermal simulation, developments of computers and advances of various analysis methods make it possible to predict the thermal performance of a multiroom building including its air distribution through rooms. These predicting methods require accurate expressions of the actual thermal dynamics of room air and wall surface as well as the interrelationships among them. Most of the simulations obtain their results to solve algebraic equations of room air temperatures, surface temperatures and heat supplies to room air. But these temperatures and supplies are not simply independent but closely connected with each other, therefore the equations should be solved simultaneously.

The Passive System Simulation Program called "PSSP" [1],[2] is a computer simulation program to predict thermal performance including air distribution of buildings especially passive solar houses, and its latest version for multiroom buildings "PSSP/MV2" is now available. It consists of a set of the successive state transition equations which are solved simultaneously and express thermal dynamics of room air temperatures and surface

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temperatures. The air distribution in a building, it is the consequence of the room air movements, is expressed as a part of the successive state transition equation of room air.

It is necessary to solve the Navier-Stokes' equation in turbulent field under unsteady or variable boundary conditions. It may be impossible, however, with the present technical level. Therefore, the following assumptions are adopted; Air temperature distribution in a room is uniform, and supplied heat to the room air diffuses instantly.

AIR MOVEMENTS IN BUILDINGS

Prediction of air movements and distribution in a building requires to examine the fluid mechanics of ventilation and infiltration. Both ventilation and infiltration have the same meaning in building thermal performance, and the difference between them is whether to be intentional or not. Ventilation is defined in this paper as a general concept of all kinds of air movements in a building. Natural ventilation airflows are classified into two types according to their fluid mechanism, one is through openings and the other is through cracks. As an airflow is caused by pressure difference which is generated by wind and thermal force, the situation of an opening and a crack in a building (at an envelope or at an interroom wall) is important.

Airflows through Openings

At an envelope, the air flows into room i from the outdoor space through an opening. The pressure difference across the opening, the airflow velocity and the airflow volume through the opening are shown as follows.

$$V_{in} = 3600 A v_{in} = 3600 A \alpha_{in} \sqrt{2g/\gamma_o P_{in}} \quad (1)$$

$$P_{in} = C \frac{\gamma_o}{2g} v_o^2 + h(\gamma_i - \gamma_o) - p_i \quad (2)$$

where

A = area of the opening (m^2)
C = wind pressure coefficient
g = gravity, 9.8m/s^2
h = height from standard pressure level (m)
P = pressure difference (kg/m^2)
p_i = reference pressure of room i (kg/m^2)
V = airflow volume (m^3/h)
v = airflow velocity (m/s)
v_o = outdoor wind velocity (m/s)
γ_i = air density of room i (kg/m^3)
γ_o = air density of the outdoors (kg/m^3)
α = airflow coefficient of the opening

and suffix in means inflowing from the outdoors to room i. Eq.(1) is derived from an assumption that the airflow is an orifice flow. The airflow coefficient α is the ratio of actual velocity to theoretical velocity, and it depends on the configuration of the opening. It is about 0.7 for a normal window.

Eq.(2) consists of wind and stack or thermal pressure. The wind pressure coefficient C is the conversion ratio of wind energy from dynamic pressure to static pressure, and it is variable according to an angle, S, between the outdoor wind direction and envelope orientation. The PSSP assumes the value of C as a linear function of S(deg.), it passes on the points of C=0.75 (S=0), C=0.75 (S=30), C=0 (S=75), C=-0.4 (S=90) and C=-0.4 (S=180). And this relation has been adopted to the HASP/ACLD [3],[4] which is the most prevalent dynamic heat load calculation computer program in Japan.

Air density, γ, is related to its temperature and barometric pressure, but simply expressed here by using absolute air temperature, T, as $\gamma = 353/T$.

It is well known that outdoor wind velocity is different according to height. If the height of an opening were extremely different from that of the observatory, it would be necessary to correct the wind velocity by using a power low (usually its power index is 1/4). The height of an opening in Eq.(2) is normally presented as the central height of the opening. If the difference of stack forces at the bottom and top of an opening were not neglected, Eq.(2) should be integrated along height.

In case that relative pressure of the outdoors is lower than that of the indoors, room air flows out to the outdoors. In this case, Eqs.(1) and (2) are rewritten as follows.

$$V_{out} = 3600Av_{out} = 3600A\alpha_{out}\sqrt{2g/\gamma_i P_{out}} \qquad (3)$$

$$P_{out} = -C\frac{\gamma_o}{2g}v_o^2 + h(\gamma_o - \gamma_i) + p_i \qquad (4)$$

where suffix out means outflowing from room i.

To combine Eqs.(1),(2),(3) and (4), the differences of air density and the airflow coefficients between the indoors and the outdoors are ignored, and the equations are reduced to as follows.

$$V = 3600A\alpha\sqrt{2g/\gamma_o}\frac{P}{|P|}\sqrt{|P|} \qquad (5)$$

$$P = C\frac{\gamma_o}{2g}v_o^2 + h(\gamma_i - \gamma_o) - p_i \qquad (6)$$

The absolute value of P in Eq.(5) makes it possible to define airflow direction, as the plus value of V means inflowing and the minus means outflowing. And it also prevents computer execution from the error occurred in root calculation where P should be equal or greater than zero.

At an opening of the interroom wall adjacent to room m, the pressure difference across the opening is only connected with stack force.

$$P = h(\gamma_i - \gamma_m) + p_m - p_i \qquad (7)$$

where

p_m = reference pressure of adjacent room m (kg/m^2)
γ_m = air density of adjacent room m (kg/m^3)

Airflows through Cracks

Pressure difference across a crack is as same as that of an opening, Eq.(6) is at an envelope and Eq.(7) is at an interroom wall. But the airflow mechanism through a crack has intermediate characteristics between an orifice flow and a capillary flow. Therefore the airflow volume, V, through a crack is expressed by using its pressure difference, P, as

$$V = sL\frac{P}{|P|}|P|^{1/n} \tag{8}$$

where

s = airflow coefficient of the crack
L = crack length (m)
n = power index

A lot of experiments were carried out to estimate s and n, and the results are summed in [5]. The power index n is an inherent constant according to the configuration, material and width of a crack, but the PSSP adopts 1.5 as the value of n in common to every crack in a building. On the other hand, each airflow coefficient is given as a simulation data, but its approximate value in default of the data is 40 for inner door, 13 for wooden sash and 3 for aluminum sash.

Air Volume Balances in Rooms

Ventilation is classified into natural ventilation and forced (or mechanical) ventilation. The former is caused by wind pressure and stack pressure across openings and cracks as was presented previously, and the latter is generated by air-handling units and electric fan ventilators. If the airflow volume of forced ventilation is given as a simulation data, the total amount of airflow volume in a room is expressed as a function of reference room pressures. For example, there is a building which has I of rooms, and there are J of airflows in room i. The total airflow volume amount of room i is given to sum up the volume of each airflow j.

$$F_i(p_1,\cdots,p_i\cdots,p_I) = \sum_j^J V_{ij} \tag{9}$$

where each airflow volume, V_{ij}, is as follows.

natural ventilation

$$V_{ij} = 3600A_{ij}\alpha_{ij}\sqrt{2g/\gamma_o}\frac{P_{ij}}{|P_{ij}|}\sqrt{|P_{ij}|} \quad \text{(for openings)} \tag{10a}$$

$$V_{ij} = s_{ij}L_{ij}\frac{P_{ij}}{|P_{ij}|}|P_{ij}|^{1/1.5} \quad \text{(for cracks)} \tag{10b}$$

forced ventilation

$$V_{ij} = \text{constant} \tag{10c}$$

and each pressure difference is

$$P_{ij} = C_{ij}\frac{\gamma_o}{2g}v_o^2 + h_{ij}(\gamma_i - \gamma_o) - p_i \quad \text{(at envelopes)} \tag{11a}$$

$$P_{ij} = h_{ij}(\gamma_i - \gamma_m) + p_m - p_i \quad \text{(at interroom walls)} \tag{11b}$$

As the inflowing volume and the outflowing volume in each room should be balanced, the total amount of airflow volume in every room should be zero.

$$F_1 = \cdots = F_i = \cdots = F_I = 0 \qquad (12a)$$

by using a vector $\mathfrak{F}$ and $\mathfrak{P}$, Eq.(12a) is rewritten into

$$\mathfrak{F}(\ \mathfrak{P}\ ) = 0 \qquad (12b)$$

where

$$\mathfrak{F} = (F_1\ ,\ \cdots\ ,\ F_i\ \cdots\ ,\ F_I)$$
$$\mathfrak{P} = (p_1\ ,\ \cdots\ ,\ p_i\ \cdots\ ,\ p_I)$$

$\mathfrak{F}$ is a nonlinear function of multitude variables, the concept of a Newton's method is applied to the solution of Eq.(12b)

$$\mathfrak{P}_{k+1} = \mathfrak{P}_k - \{\mathfrak{F}'(\mathfrak{P}_k)\}^{-1}\mathfrak{F}(\mathfrak{P}_k) \qquad (13a)$$

As Eq.(13a) contains an inverse matrix of partial differential matrix, $\mathfrak{F}'(\mathfrak{P}_k)$, the next transformation gives more simple procedure to solve linear simultaneous equations instead of matrix inversion.

$$\mathfrak{F}'(\mathfrak{P}_k)(\mathfrak{P}_k - \mathfrak{P}_{k+1}) = \mathfrak{F}(\mathfrak{P}_k) \qquad (13b)$$

The dimension of partial differential matrix, $\mathfrak{F}'$, is I by I, and its each element is given numerically as

$$\frac{\partial F_i(\mathfrak{P}_k)}{\partial p_{i,k}} = \frac{F_i(p_1,\cdots,p_i+e,\cdots,p_I) - F_i(p_1,\cdots,p_i-e,\cdots,p_I)}{2e} \qquad (14)$$

where the increment, e, is arbitrary, but it affects the computing time and the accuracy of the solution.

Many examples show that a Newton's method is not always convergent. The possibility of convergence depends on the nature of its function and the adequateness of its initial values. The function $\mathfrak{F}$ is mathematically an odd function. It has a nature that it diverges taking plus and minus values alternately, if the one step prior values of room reference pressures are inadequate. Therefore the next procedure keeps the room reference pressures within a range where Eq.(13b) is convergent.

$$\mathfrak{P}_{k=2n} = (\mathfrak{P}_{k-1} + \mathfrak{P}_{k-2})/2 \quad (n\text{:natural number}) \qquad (15)$$

SUCCESSIVE STATE TRANSITION EQUATIONS

The PSSP consists of the successive state transition equations of surface temperatures and room air temperatures expressed finally as linear simultaneous equations. The temperatures of every surface and every room air in a building at every time step can be acquired by solving these equations.

Heat Balances at Surfaces

A heat balance equation of surface j at time n is given by

$$CV_{j,n} + NSR_{j,n} + NLR_{j,n} + CD_{j,n} = 0 \qquad (16)$$

where

CD = conductive heat flux (W/m^2)

CV = convective heat flux (W/m^2)
NSR = net shortwave radiation (W/m^2)
NLR = net longwave radiation (W/m^2)

Net radiation is classified into shortwave and longwave by 3 μm wave length which is the critical wave length of solar transmission through normal pane. It is possible to take latent heat flux into account in Eq.(16) at outside wall surfaces, and its procedure applying the Lewis' law is described in the reference [1]. On the other hand, humidity absorption at inside surfaces is urgent but still left unresolved for the PSSP.

Convective heat flux at outside surface $\bar{j}$ is given by

$$CV_{\bar{j},n} = \alpha_{c,\bar{j},n}(T_{o,n} - T_{\bar{j},n}) \tag{17}$$

where

$T_{\bar{j}}$ = temperature of surface $\bar{j}$ (K)
T_o = outdoor air temperature (K)
$\alpha_{c,\bar{j}}$ = convective heat transfer coefficient between outdoor air and outdoor surface $\bar{j}$ (W/m^2K)

There are many proposals to estimate the outside convective heat transfer coefficient, the PSSP adopts the next experimental equations using the ambient air velocity adjacent to the surface.

$$\left.\begin{array}{lll} \text{for walls [6]} & & \\ \alpha_{c,\bar{j},n} = 4.7 + 7.6v_{\bar{j},n} & & \\ \quad v_{\bar{j},n} = 0.25v_{o,n} & v_{o,n} > 2 & \text{(for windward)} \\ \quad v_{\bar{j},n} = 0.5 & v_{o,n} \leq 2 & \text{(for windward)} \\ \quad v_{\bar{j},n} = 0.3 + 0.05v_{o,n} & & \text{(for leeward)} \end{array}\right\} \tag{18a}$$

$$\left.\begin{array}{l} \text{for roofs [7]} \\ \alpha_{c,\bar{j},n} = 8.72 + 2.33v_{\bar{j},n}, \\ \quad v_{\bar{j},n} = 0.3 + 0.05v_{o,n} \end{array}\right\} \tag{18b}$$

where

$v_{\bar{j}}$ = ambient air velocity adjacent to surface $\bar{j}$ (m/s)
v_o = outdoor wind velocity (m/s)

Net shortwave radiation at outside surface $\bar{j}$ is given by

$$NSR_{\bar{j},n} = a_{\bar{j}}\{\xi_{\bar{j},n}DN_n \cos\theta_{\bar{j},n} + F_{\bar{j}s}SH_n + (1 - F_{\bar{j}s})\rho_g TH_n\} \tag{19}$$

where

$a_{\bar{j}}$ = solar absorptance of surface $\bar{j}$
DN = direct solar radiation incident upon normal surface (W/m^2)
$F_{\bar{j}s}$ = shape factor of surface $\bar{j}$ viewing the sky
SH = diffuse solar radiation incident upon horizontal surface (W/m^2)
TH = global solar radiation (W/m^2)

$\theta_{\bar{j}}$ = incident angle of direct solar radiation upon surface $\bar{j}$ (deg.)
$\xi_{\bar{j}}$ = sunlit area ratio of surface $\bar{j}$
ρ_g = solar reflectance of the ground surface

Effects of solar shading such as overhangs and eaves are taken into account in terms of the sunlit area ratio, ξ, and the shape factor viewing the sky, F_{js}.

Net longwave radiation at an outside surface is defined as the equation of its surface temperature, and an assumption that the ground surface is approximately the black body and its temperature is nearly equal to the outdoor air temperature, conducts the following equation.

$$NLR_{\bar{j},n} = \varepsilon_{\bar{j}}\{F_{\bar{j}s}AH_n + (1 - F_{\bar{j}s})\sigma T_{o,n}^4 - \sigma T_{\bar{j},n}^4\} \tag{20a}$$

where

AH = atmospheric radiation incident upon horizontal surface (W/m^2)
$\varepsilon_{\bar{j}}$ = longwave emittance or absorptance of surface $\bar{j}$
σ = Stefan-Boltzmann's constant, 5.67×10^{-8}W/m^{2}K^4

Eq.(20a) is a nonlinear equation of the fourth degree of $T_{\bar{j}}$. But the following conversion makes it possible to linearize the equation.

$$NLR_{\bar{j},n} = \varepsilon_{\bar{j}}F_{\bar{j}s}(AH_n - \sigma T_{o,n}^4) + \alpha_{r,\bar{j},n}(T_{o,n} - T_{\bar{j},n}) \tag{20b}$$

$$\alpha_{r,\bar{j},n} \fallingdotseq 4\varepsilon_j\sigma\{(T_{\bar{j},n-1} + T_{o,n})/2\}^3 \tag{21}$$

where

$\alpha_{r,\bar{j}}$ = radiative heat transfer coefficient between outdoor air and surface $\bar{j}$ (W/m^{2}K)

As the outside surface temperature at time n, $T_{\bar{j},n}$ is an unknown variable to be solved, so the value at time n-1 is substituted into Eq.(21). As long as there is slight difference among $T_{\bar{j},n}$, $T_{\bar{j},n-1}$ and $T_{o,n}$, this approximation is accurate.

As well as an outside surface, convective heat flux at inside surface j is given by

$$CV_{j,n} = \alpha_{c,j,n}(T_{i,n} - T_{j,n}) \tag{22}$$

where

$\alpha_{c,j}$ = convective heat transfer coefficient between room air and inside surface $\bar{j}$ (W/m^{2}K)

There are many factors to define the convective heat transfer coefficient at an inside surface, such as ambient airflow velocity near the surface, temperature difference between the surface and the room air and direction of convective heat flux. In a forced ventilated room (in an air-conditioned room), the PSSP adopts 7.3W/m^{2}K as $\alpha_{c,j,n}$ in common that is derived from a Jurges's equation [8] assuming 0.5m/s of ambient airflow velocity near surfaces. In natural ventilation, the next relation is supposed.

$$\alpha_{c,j,n} = \nu|T_{i,n-1} - T_{j,n-1}|^n \tag{23}$$

where

ν = direction-dependent coefficient

The power index n and the coefficient ν are given according to McAdams [9] as n=0.25 in common, ν=1.8 for horizontal heat flux, ν=2.5 for upward and ν=1.3 for downward respectively. The room air temperature and the surface temperature in Eq.(23) should be better to be the values at time n, but they are unknown values at time n, so the values at time n-1 are substituted.

To estimate radiative heat inter-exchange (multi-reflection and absorption) among inside surfaces, a concept of the absorption factor proposed by Gebhart [10] is introduced. The shortwave absorption factor, which denotes the ratio of the emitted shortwave radiation from surface l to the absorbed one at surface j, is given by

$$\gamma_{lj} = F_{lj}a_j + \sum_{k}^{J} F_{lk}\rho_k\gamma_{kj} \tag{24}$$

where

F_{lj} = shape factor of surface l viewing surface j
γ_{lj} = shortwave absorption factor from surface l to surface j

and suffix k means an arbitrary surface in the room where there are J of surfaces.

There are J of surfaces and F kinds of fenestration in a room. The direct solar radiation transmits through fenestration f and incidents upon surface l. The sunlit area of surface l absorbs a part of it and reflects the rest according to its solar absorptance and the reflectance, and surface j absorbs the reflected from surface l directly and by way of other surfaces. Besides, there are other shortwave radiations ought to be absorbed at surface j, transmitted diffuse solar radiation through fenestration f and illumination from surface k. The net shortwave radiation at inside surface j is given by

$$NSR_{j,n} = \sum_{f}^{F}\tau_f DN_n \sum_{l}^{J}\xi_{fl,n}(a_j\delta_{jl} + \gamma_{lj}\rho_l S_l/S_j)\cos\theta_{l,n} + \sum_{f}^{F}\tau_f F_{fs} SH_n \gamma_{fj} S_f/S_j + \sum_{k}^{J} SL_{k,n}\gamma_{kj}S_k/S_j \tag{25a}$$

where

F_{fs} = shape factor of fenestration f viewing the sky
S_j = area of surface j (m^2)
SL_k = shortwave radiation of illumination and so on from surface k (W/m^2)
δ_{jl} = Kronecker's delta, if $j=l$ $\delta_{jl}=1$, if $j\neq l$ $\delta_{jl}=0$
ξ_{fl} = sunlit area ratio of surface l according to the transmitted solar radiation through fenestration f
τ_f = solar transmittance of fenestration f

By using a theorem of reciprocality such as $a_l S_l \gamma_{lj} = a_j S_j \gamma_{jl}$, Eq.(25a) can be rewritten to

$$NSR_{j,n} = \sum_{f}^{F}\tau_f DN_n \sum_{l}^{J}\xi_{fl,n}a_j(\delta_{jl} + \gamma_{jl}\rho_l/a_l)\cos\theta_{l,n} + \sum_{f}^{F}\tau F_{fs} SH_n \gamma_{jf} a_j/a_f + \sum_{k}^{J} SL_{k,n}\gamma_{jk}a_j/a_k \tag{25b}$$

In the same manner, the longwave absorption factor is defined as

$$\beta_{lj} = F_{lj}\varepsilon_j + \sum_k F_{lk}(1-\varepsilon_k)\beta_{kj} \tag{26}$$

where

β_{lj} = longwave absorption factor from surface l to surface j

Surface j absorbs the longwave radiation emitted from surface l, which is proportional to the fourth power of its surface temperature, and that of illumination, human bodies and so on from surface k. And surface j itself also emits longwave radiation according to its surface temperature.

$$NLR_{j.n} = \sum_l^J \beta_{lj}\varepsilon_l\sigma T_{l.n}^4 S_l/S_j + \sum_k^J LL_{k.n}\beta_{kj}S_k/S_j - \varepsilon_j\sigma T_{j.n}^4 \tag{27a}$$

where

LL_k = longwave radiation of illumination, human bodies and so on from surface k (W/m^2)

A theorem of reciplocality such as $\varepsilon_l S_{l.n}\beta_{lj.n} = \varepsilon_j S_{j.n}\beta_{jl.n}$, and an energy preservation low, $\sum_l^J \beta_{lj.n} = 1$, rewrite Eq.(27a) into

$$\begin{aligned} NLR_{j.n} &= \sum_l^J \beta_{lj}\varepsilon_l\sigma(T_{l.n}^4 - T_{j.n}^4) + \sum_k^J LL_{k.n}\beta_{jk}a_j/a_k \\ &= \sum_l^J \beta_{lj}\alpha_{r.lj.n}(T_{l.n} - T_{j.n}) + \sum_k^J LL_{k.n}\beta_{jk}a_j/a_k \end{aligned} \tag{27b}$$

where

$\alpha_{r.lj}$ = radiative heat transfer coefficient between surface l and surface j (W/m^{2}K)

$\alpha_{r.lj}$ is introduced to linearize the fourth degree relation to the single degree between the surface temperatures of l and j, but the surface temperatures at time n is unknown. So the temperatures at time n-1 is substituted.

$$\alpha_{r.lj.n} \doteqdot 4\varepsilon_j\sigma\{(T_{l.n-1}+T_{j.n-1})/2\}^3 \tag{28}$$

Both the shortwave absorption factor and longwave absorption factor defined as Eq.(24) and Eq.(26) respectively, are the inherent characteristics of radiative heat inter-exchange between surfaces in a room. They are expressed as simultaneous equations and must be solved prior to the calculation of inside surface heat balances.

Successive State Transition Equations of Surface Temperatures

To evaluate conductive heat flux through walls, the approximate inditial response [11],[12] of multi-layer wall is introduced as

$$\Phi_m(t) = A + \sum_k^K A_{m.k}\exp(-\alpha_k t) + Q_m\delta(t) \tag{29}$$

where

A = each term's coefficient of wall inditial response (W/m^{2}K)
Q = heat absorbed immediately on wall inditial response (J/m^{2}K)
t = time (h)

α = root of characteristic equation of wall heat conduction system (1/h)
δ = Dirack's delta function (1/h)
Φ = approximate inditial response

and suffix m denotes the sort of surface heat flow response as $m=j$ is inside surface heat flow by inside surface temperature excitation, $m=o$ is concerned surface flow by its opposite side surface excitation and $m=\bar{j}$ is outside surface flow by outside surface excitation. The number of the roots and the coefficients of a wall inditial response is ordinarily infinite. But the number of them in Eq.(29) is designated previously as K according to the conductive characteristics of a wall, and the error caused by its approximation is compensated in terms of Q.

Through a trapezoid hold function for sampled data and Z-transform of Eq.(29), the successive state equations of wall heat conduction is given as follows.

$$\bar{C}D_{j,n} = a_o T_{\bar{j},n} - a_j T_{j,n} + D_{j,n-1} \tag{30a}$$

$$D_{j,n-1} = b_o T_{\bar{j}n-1} - b_j T_{j,n-1} + \sum_k^K x_{j,k,n-1} \tag{30b}$$

$$x_{j,k,n-1} = \varphi_k x_{j,k,n-2} + p_k(A_{o,k}T_{\bar{j},n-2} - A_{j,k}T_{j,n-2}) + q_k(A_{o,k}T_{\bar{j},n-1} - A_{j,k}T_{j,n-1}) \tag{30c}$$

$$\left.\begin{array}{ll} \varphi = \exp(-\alpha_k\Delta), & \\ a_m = A + \sum_k^K A_{m,k}(1+q_k)+Q_m/\Delta, & b_m = \sum_k^K A_{m,k}p_k - Q_m/\Delta, \\ p_k = \varphi_k - (1-\varphi_k)/(\alpha_k\Delta), & q_k = -p_k - (1-\varphi_k) \end{array}\right\} \tag{30d}$$

where

Δ = calculating time interval (h)

The successive state transition equation of an inside surface temperature is led by substituting Eq.(30a) into Eq.(16) as

$$(a_j + \alpha_{c,j,n} + \sum_l^J \beta_{lj}\alpha_{r,lj,n})T_{j,n} - a_o T_{\bar{j},n} - \alpha_{c,j,n}T_{i,n} - \sum_l^J \beta_{lj}\alpha_{r,lj,n}T_{l,n}$$
$$= NSR_{j,n} + ALL_{j,n} + \sum_k^J LL_{k,n}\beta_{kj}\varepsilon_j/\varepsilon_k + D_{j,n-1} \tag{31}$$

where $\bar{j}$ means the opposite side surface of the wall (it may be inside surface or outside surface) against the concerning surface j.

On the other hand, if $\bar{j}$ denotes the outside surface at an envelope and j is its opposite inside surface, the successive state equation of the outside surface temperature is given by

$$(a_{\bar{j}} + \alpha_{c,\bar{j},n} + \alpha_{r,\bar{j},n})T_{\bar{j},n} - a_o T_{j,n} = (\alpha_{c,\bar{j},n} + \alpha_{r,\bar{j},n})T_{o,n}$$
$$+ NSR_{\bar{j},n} + \varepsilon_{\bar{j}}F_{\bar{j}s}(AH_n - \sigma T_{o,n}^4) + D_{\bar{j},n-1} \tag{32}$$

Successive State Transition Equations of Room air Temperatures

A heat balance equation of room air is a state equation of the room air temperature. As well as surface temperatures, a trapezoid hold function of sampled data and Z-transform of the differencial equation of a room air temperature give the successive state transition equation of room air temperature.

$$T_{i,n}/q_{i,n} - \sum_{j}^{J} \alpha_{c,j,n} S_{j,n} T_{j,n} - \sum_{m}^{I} \Lambda_{m,n} VI_{m,n} T_{m,n} = TA_{n-1} + \Lambda_{o,n} VI_{o,n} T_{o,n} + CL_{i,n} + H_{i,n} \quad (33a)$$

$$TA_{n-1} = \varphi_n T_{i,n-1}/q_n + p_n/q_n \left(\sum_{j}^{J} \alpha_{c,j,n} S_j T_{j,n-1} + \sum_{m}^{I} \Lambda_{m,n} VI_{m,n} T_{m,n-1}\right) + p_n/q_n (\Lambda_{o,n} VI_{o,n} T_{o,n-1} + CL_{n-1} + H_{n-1}) \quad (33b)$$

$$\left.\begin{aligned} &\varphi_n = \exp(-B_n \Delta / RQ_n), \quad B_n = \sum_{j}^{J} \alpha_{c,j,n} S_{j,n} + \sum_{m}^{I} \Lambda_{m,n} VI_{m,n} + \Lambda_{o,n} VI_{o,n} \\ &p_n = -\{\varphi_n - RQ_n(1-\varphi_n)/(B_n \Delta)\}/B_n, \quad q_n = -p_n + (1-\varphi_n)/B_n \end{aligned}\right\} \quad (33c)$$

where

CL_i = convective heat generated by illumination, human bodies and so on in room i (W)
H_i = supplied heat to room air i (W)
VI = amount of inflowing air volume from other room m or outdoor o (m^3/h)
Λ = volumetric air specific heat (J/m^3K)

Either room air temperature, $T_{i,n}$, or supplied heat, $H_{i,n}$, is the unknown value in Eq.(33a). If the room is air-conditioned and $T_{i,n}$ is designated, $H_{i,n}$ is the unknown value. If the supplied heat $H_{i,n}$ is designated including the case of $H_{i,n}=0$, Eq.(33a) is solved for $T_{i,n}$. The calculation methods of inflowing air volume to rooms are shown previously, where air density to estimate stack effects is described as a function of air temperature. But room air temperature at time n, $T_{i,n}$, is the unknown variables at time n. So the value at time n-1, $T_{i,n-1}$, is substituted instead of $T_{i,n}$ into the calculation of airflows through rooms.

If accurate estimation is requested, the air density at time n is calculated by using $T_{i,n-1}$, by which inflowing volume to a room is estimated and Eq.(33a) is solved as the first step. Next, the air density is calculated by using the air temperature of the first step and equations are solved for the second step. These steps are repeated in order to converge all of the room air and surface temperatures. The coefficients shown previously in Eqs.(21), (23) and (28) are corrected to the more accurate values through these steps.

The PSSP solves Eqs.(31), (32) and (33a) simultaneously, to acquire every room air and surface temperature in a building at every time step. It needs building input data to define the characteristics of a building and weather input data to execute building thermal simulations.

SIMULATION AND DISCUSSION

The PSSP is applied to thermal performance simulations of a passive solar house which has characteristic air circulation system through spaces by means of natural ventilation caused by stack effects. This air circulation system is called the Passive Air Circulation system (PAC). The main purpose of the simulations is to confirm the validity and efficiency of the PAC. Stack effect is generated by air density difference, air density is connected with its temperature, and air is warmed or cooled by its ambient surfaces. Therefore the simulations of the PAC require exact estimation of temperatures and air movements. As the PSSP consists of the successive state transition equations, which contain air and surface temperatures and airflow volume as the variables and are solved simultaneously, it could satisfy the requirements.

The Passive Air Circulation System

The basic principle of the PAC is the heat transportation through air circulation. If different air temperature spaces are connected by different height openings, the air circulates through the openings as long as the air temperature difference exists. If the warm space absorbs solar heat, the cold space is warmed indirectly by it through the air circulation. The transported heat is proportional to the airflow volume and temperature difference.

Figure 1 and Table 1 show a prototype model house of the PAC and its expected air circulation. There are two rooms in the model house, and inner spaces and outer spaces between the rooms and the outdoors. There are two different height openings at each wall between an inner and outer space, and a check damper is attached to each lower opening where only inflowing is possible from the inner space to the outer space. Ventilators at the walls of the attic and crawl space are shut to enclose warm air in winter but opened to exhaust hot air in summer. The interspace walls made of thermal insulation materials and the pair glass windows reduce the heat conduction between the rooms and the outdoors.

In the daytime in winter, the air temperatures in some outer and roof spaces become higher than those of other spaces by the solar absorption at their outside surfaces. These spaces are like solar collectors, suck cold air and exhaust warm air according the stack pressures. The generated warm air is carried to the room surfaces by air circulation through the inner, attic and crawl spaces. The north room, where no direct solar radiation incidents in winter, is warmed by the air circulation. Furthermore, the air circulation prevents the south room, where transmitted direct solar radiation incidents, from excessive temperature rising.

In the night of winter, the check dampers prevent the inner spaces from cold draft through the outer spaces. If the south room surfaces were warm in the night by the daytime solar gain, air circulation could be caused and temperatures of air and surfaces in the whole house would be equalized. This implies not only temperature rising up but also relative humidity falling down of the air in northern ares of the house, that is the coldest area in a house, so the PAC is effective to the prevention of dew condensation at inside surfaces.

In summer, the outdoor air flows into the attic and crawl space and hot air in the house is exhausted through the ventilators. In case that the temperature of the outdoors is lower than that of the indoors such as in the night, the house is cooled passively by natural ventilation.

Simulation of the Passive Air Circulation System

The prototype model house is 3.6m by 3.6m and 3.8m heigh. Each room is 2.1m high and has a pair glass window of 1.3m by 1.3m. The outside walls are made of wire mesh and mortar but the roof is 1.2cm of laminate wooden board and roof tile. The interspace walls are made of hard foam polyurethane board, and the walls and ceilings of the rooms are 2cm of plaster board but the floors are double flooring of 1.2cm of laminate wooden board. The width of the inner spaces is 10cm and that of the outer spaces is 5cm. The area of the openings and the check dampers is $0.15m^2$ and $0.05m^2$ respectively. The area of the ventilators in the attic space is $0.1m^2$, but that in the crawl space is $0.05m^2$. The flow coefficients of these openings, dampers and ventilators are assumed to be o.7 in common. The solar absorptance and longwave emittance are supposed as 0.8 and 0.9 at outside surfaces and 0.7 and 0.9 at inside surfaces respectively.

Another model house called the NONPAC is simulated in the comparison with the PAC. The NONPAC has the same structure as the PAC, but it has no openings and check dampers at the interwalls, and ventilators in the attic and crawl space are opened all the year round (there is 5cm of glass wool heat insulation layer on the plaster board of the ceiling). The NONPAC is assumed to represent an ordinary heat insulated house.

As the prototype model house is divided into 15 of rooms and spaces as shown in Fig.1 and Table 1, 15 of air temperatures, 106 of surface temperatures and 32 of airflows volume are calculated hourly by the PSSP. A typical fine day in winter (Feb.,4) and that of in summer (Aug.,3) are selected for the output days of the simulations from the Standard Weather Data (SWD) of Fukuoka, but the simulations are carried out for four days term prior to each output day. The SWDs were made for the HASP/ACLD at first, but now are utilized for various purposes. The SWD consists of hourly year round data; temperature and absolute humidity of outdoor air, direct and diffuse solar radiation, velocity and direction of wind and cloud ratio to estimate atmospheric radiation. The soil temperature in depth of 50cm is assumed as 15°C in winter and 25°C in summer by previous underground temperature simulation results using finite differencial methods [13].

The weather condition and hourly room air temperatures fluctuation in a typical fine day of winter are shown in Fig.2. The maximum air temperature of the PAC south room is 21°C and it is 3 deg. higher than that of the NONPAC. The PAC north room is 3 deg. higher than the NONPAC north room in the maximum air temperature,too. In the night, the air temperatures of both the south and north room in the PAC are 1∼1.5°C higher than those in the NONPAC. And the air temperature differences between the south and north room in the PAC are greater than those in the NONPAC. These are the effects of solar heat absorption at the outer spaces and the air circulation system in the PAC. The simulation results in summer are shown in Fig.3. These results are obtained under assumptions that the windows are opened from 8:00 a.m. to 8:00 p.m. and the solar radiation incident upon the windows is reduced to 0.2 times by means of solar shading such as eaves and outside blinds. In the night, the room air temperatures in the PAC are lower about 1°C than those in the NONPAC because of outdoor air inflowing and circulating through the inner spaces in the PAC. But in the daytime, introduction of the outdoor air in the PAC heightens surface temperatures of the inner surfaces, the room air temperatures in the PAC are consequently higher than those in the NONPAC.

Airflow and temperature distribution in the PAC are shown in Fig.4 ~ Fig.6 for further discussions. In Fig.4 (2:00 p.m. in winter), the air in the south inner space is warmed by the solar radiation and flows into the attic space. The air in the attic space flows down to the crawl space through the south, middle and east inner spaces where the air temperatures are lower than that of the attic space but higher than that of the crawl space. The air in the crawl space flows up to the attic space through the west and north inner spaces where the air temperatures are lower than that of the crawl space, and a part of the air in the west inner space is induced to the west outer space which has higher air temperature than its inner space. The south outer space induce the air in the south inner space, and the south inner space sucks not only the air in the attic space but also the cold air in the crawl space because of the strong suction power of its outer space. The air circulation at dawn in winter is shown in Fig.5. The warmest area in the house is the south room, and the air in the attic space flows down to the crawl space being gradually cooled through the east and north inner space. The air in the crawl space flows up to the attic space through the middle, south and west spaces where the air temperatures are higher than that of the attic space. On the other hand, the air circulation at dawn in summer is shown as Fig.6. As the inflowing air volume from the outdoors to the crawl space through the ventilators is $204m^3/h$ and that of the outflowing is $124m^3/h$, so $80m^3/h$ of the outdoor air is circulated in the house and exhausted finally through the ventilator in the attic space. The air in the crawl space flows up getting warm, cooling the surrounding surfaces in other words, to the attic space. A part of the attic space air flows out to the outdoors and the rest flows down to the crawl space through the north inner space.

CONCLUSIONS

The main purpose of this paper is to describe simulation methods of air distribution in multiroom buildings. The details of airflow calculation procedures are shown, but the airflows are affected by air temperatures and surface temperatures. Therefore the successive state transition equations to be solved simultaneously and a computer simulation program called the PSSP are introduced. Thermal performances including air distribution of a passive solar model house are simulated by the PSSP. The model house is tinny but has complex air circulation systems, 15 of rooms and spaces, 106 of surfaces and 32 of airflows connected with each other. Through the simulations, the effectiveness of the methods to predict room air distribution is confirmed.

Air movements in rooms and buildings are so sensitive, rapid and complex that many unresolved problems are left, but it is indispensable to building thermal performance simulations. Information exchanges between the field of thermal performance and air movement are the must for the further developments of the both field researches.

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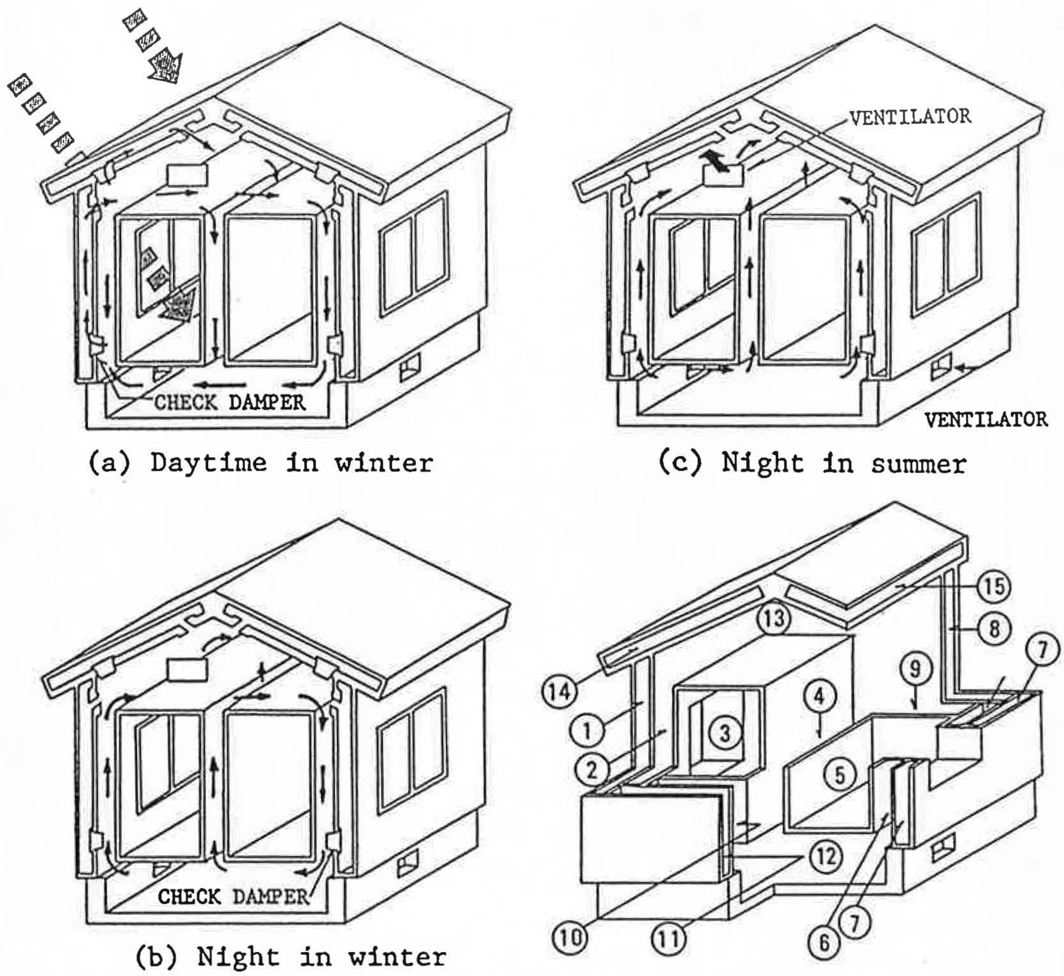


(a) Daytime in winter

(c) Night in summer

(b) Night in winter

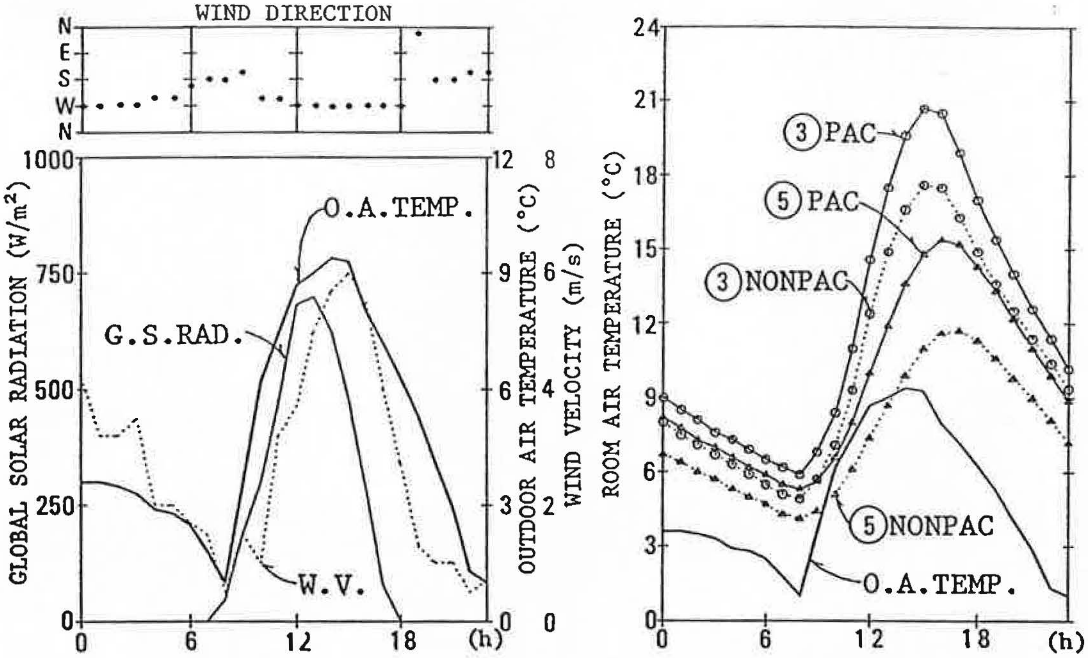
(d) Naming of the rooms and spaces

Figure 1 Airflow pattern in the Passive Air Circulation system

Table 1 Rooms and spaces in the Passive Air Circulation system

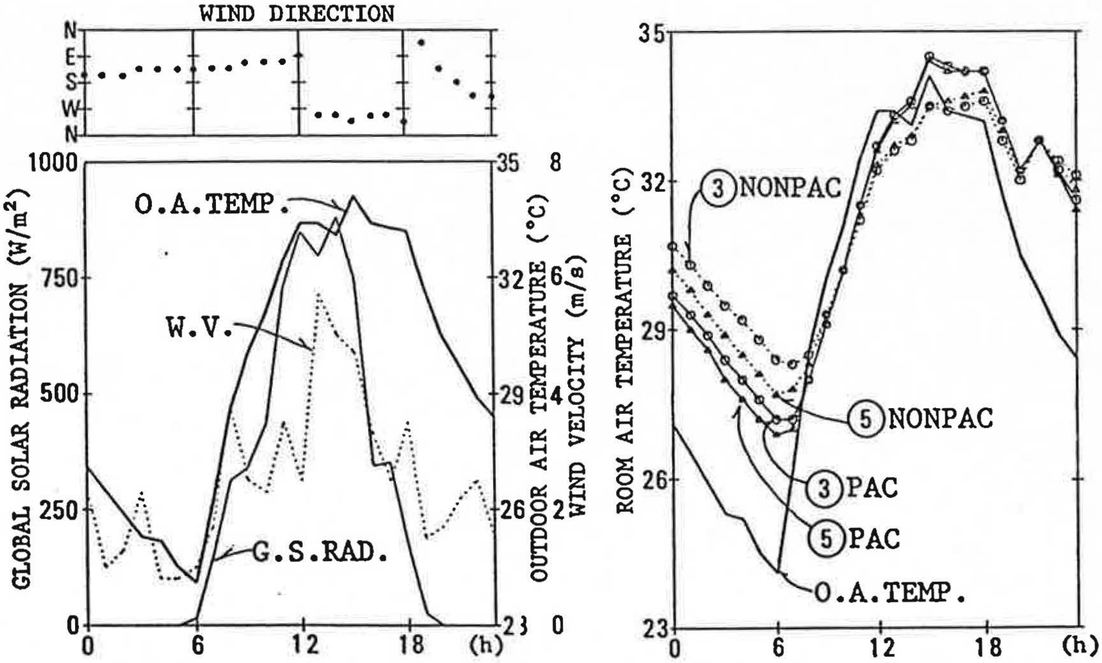
no.	name
1	south outer space
2	south inner space
3	south room
4	middle space
5	north room
6	north inner space
7	north outer space

no.	name
8	west outer space
9	west inner space
10	east inner space
11	east outer space
12	crawl space
13	attic space
14	south roof space
15	north roof space



(a) Weather condition (b) Room air temperature

Figure 2 Weather condition and room air temperature fluctuation on a typical fine day in winter (Feb. 4)



(a) Weather condition (b) Room air temperature

Figure 3 Weather condition and room air temperature fluctuation on a typical fine day in summer (Aug. 3)

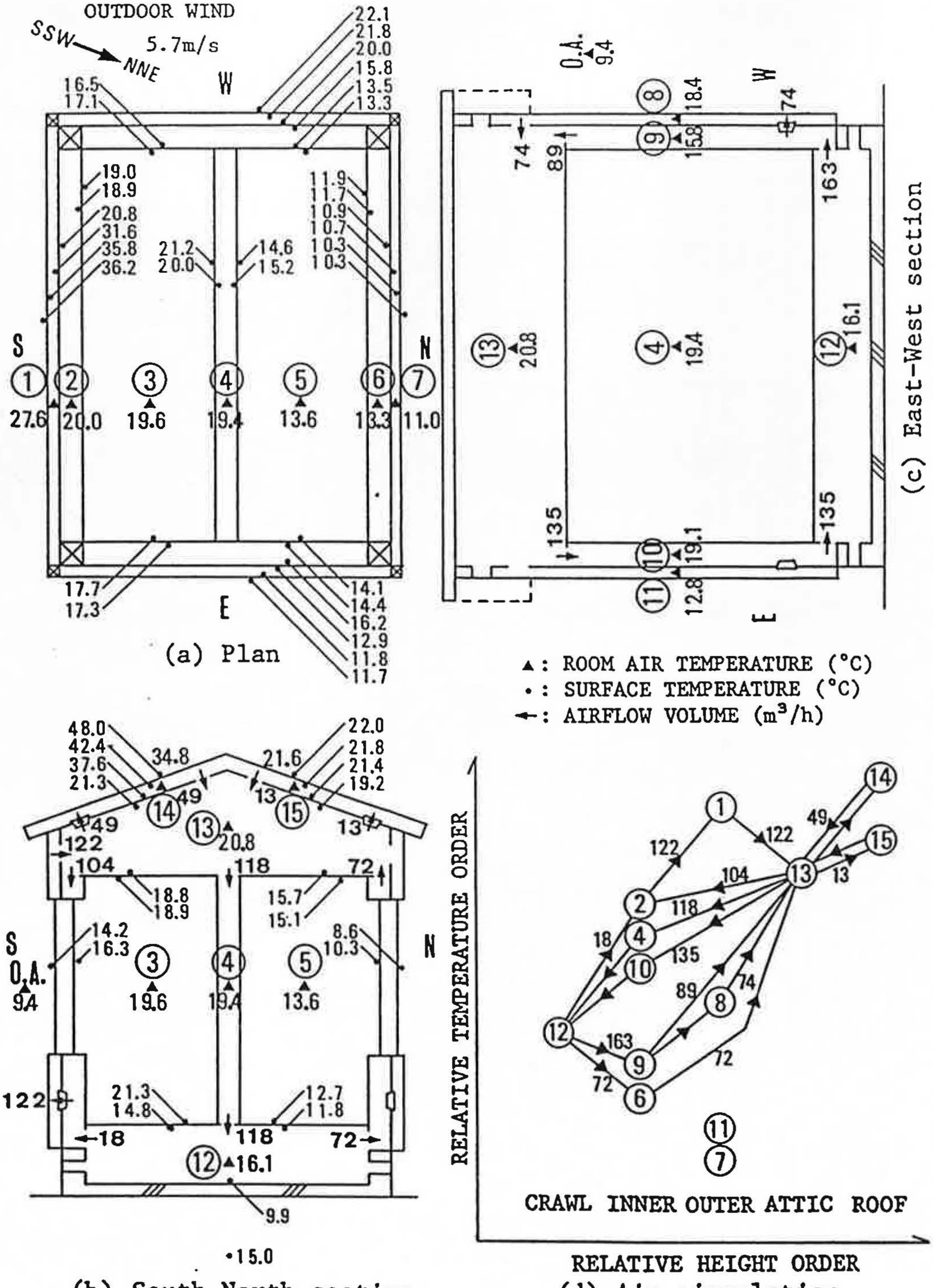


Figure 4 Airflow and temperature distribution in the afternoon of winter (Feb. 4, 2:00 p.m.)

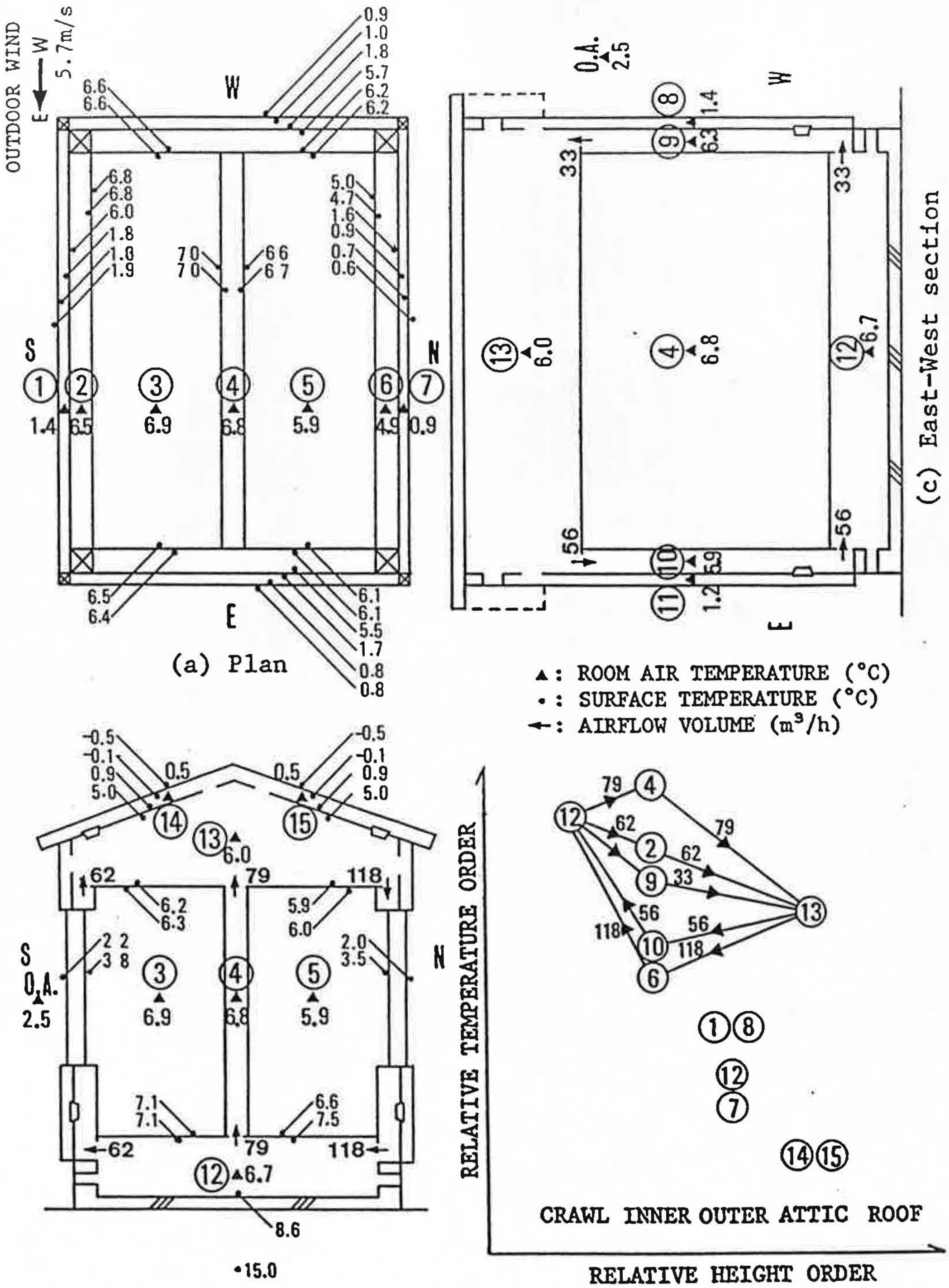


Figure 5 Airflow and temperature distribution at dawn of winter (Feb. 4, 6:00 a.m.)

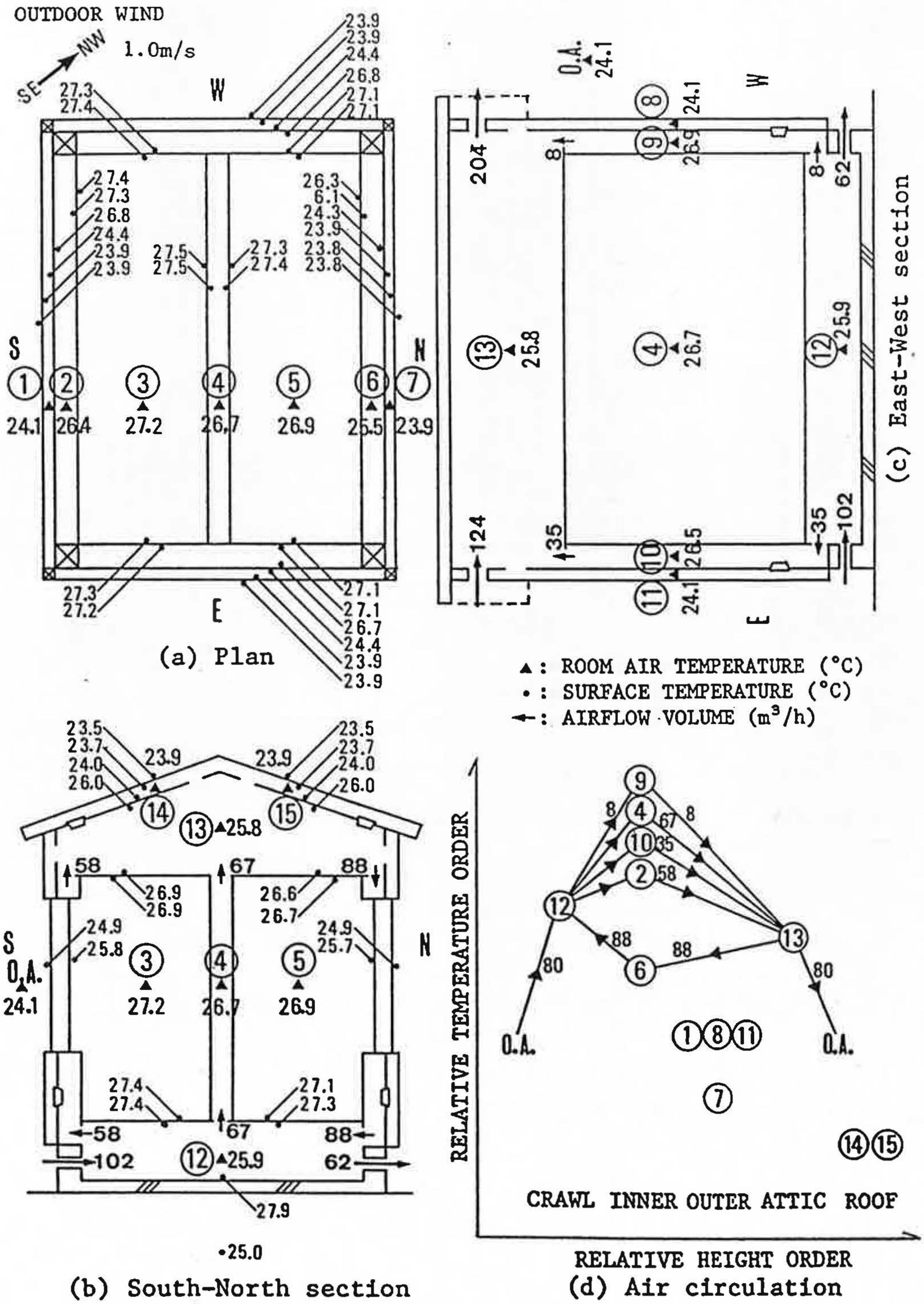


Figure 6 Airflow and temperature distribution at dawn of summer (Aug. 3, 6:00 a.m.)