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Hybrid Simulation of a Ventilation System based on Natural Convection

Part 2——Experiment with a Sloped Roof Model and Estimation of Ventilation Rate

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Key Words: Hybrid Simulation, Natural Ventilation System, Convective Heat Transfer Coefficient, Flow Pattern, Ventilation Rate, Sloped Roof Model, Rayleigh Number

Synopsis: Hybrid simulation was executed with a sloped roof model in the water tank using the method described in the accompannied paper. The values of convective heat transfer coefficient obtained along the inside surface of different parts of the room enclosure were obtained and the results were compared with those of a flat roof model. Ventilation rate and air change rate of the room were estimated using the results obtained from the measurement of water flow velocity combined with estimated pressure difference between inside and outside.

Nomenclature

- A : area [m²]
- b, c : constant [−]
- d : height of vertical plate, length of side of square horizontal plate [m]
- g : gravity acceleration constant 9.8 [m/s²]
- h_i : height of the part i of the room space from the middle of the door ($i=1, 2, \cdots, 32$) [m]
- I : direct incident solar radiation [kcal/m²·h]
- m : air change rate [h⁻¹]
- n : time step [h]
- Nu : Nusselt number ($=\alpha d/\lambda$) [−]
- q : rate of heat flow [kcal/m²·h]
- $q_{ci}(n)$: rate of convective heat flow from the i part of the inside surface at time n [kcal/m²·h]
- Q : rate of ventilation or air movement [m³/h]
- Ra : Rayleigh number [$=g\beta\Delta\theta l^3/(\nu a)$] [−]
- T_i : inside absolute temperature [°K]
- T_o : outside absolute temperature [°K]
- u : velocity [m/s]
- v_i : volume of the i part of the room space ($i=1, 2, \cdots, 32$) [m³]
- V : room volume [m³]
- z : height from neutral zone [m]
- α : convective heat transfer coefficient

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Fig. 1 Acrylic model with sloped roof

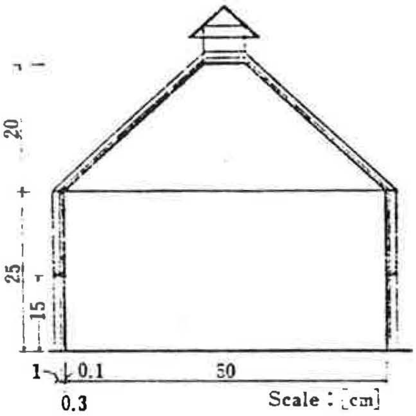


Fig. 2 Section of the experiment model with sloped roof

[kcal/m²·h·°C]

δ_1 : 1 mm distance between first measurement point and surface [m]

δ_2 : 2 mm distance between second measurement point and surface [m]

δ_m : 1 mm width of the acrylic layer [m]

ξ : pressure loss coefficient [−]

Δp : pressure drop between outside and inside [kg/m²]

γ : specific weight [kg/m³]

γ_i : specific weight of the i part of the inside room space ($i=1, 2, \cdots, 32$) [kg/m³]

γ_o : specific weight of the outside air [kg/m³]

λ : thermal conductivity [kcal/m²·h·°C]

suffix; m: model, w: water

ν : coefficient of kinematic viscosity [m²/h]

ψ : slope tilt angle [°]

θ_0 : temperature of the room-side surface of the wall or roof [°C]

θ_1, θ_2 : temperatures of the points at 1 mm distance and 2 mm distance from the wall or roof [°C]

θ_m : internal surface temperature of the double layer wall or roof of the model [°C]

θ_r : inside room temperature [°C]

Introduction

A method of hybrid simulation was proposed in the accompanied paper to clarify the nature of natural ventilation caused by the natural convection along the inside surface of higher temperature in a buliding with a flat roof[1]. Water was used as fluid media and acrylic plate as a material for the scaled down model. Hybrid simulation implies simultaneously performed simulation of calculation and experiment where calculations are made for the heat transfer of the building, and at the same time, the model experiment is conducted to determine the values of convective heat transfer coefficient along the inside surfaces. The results of both experiment and calculation are thus combined on the real time basis.

In order to study the effect of roof shape on the natural ventilation system, an attempt was made with a sloped roof model, again using water as fluid media. Convective heat transfer coefficients were obtained by hybrid simulation method.

The water flow pattern inside the model space could be observed to help to understand the nature of natural ventilation system.

It was then attempted to estimate the rate of ventilation under given conditions by measuring the water flow veiocity at the outlet opening and by calculating the pressure difference between segmented inside volumes and outside.

1. Sloped roof model with vertical walls

The second experiment was made using an acrylic model with a sloped roof with a scale of 1/5.18 (**Figs. 1, 2**). The total height of the model was 45 cm. Same as in the case of the flat roof model, the calculation was made for a presupposed building with 20 cm thick concrete walls and roof. **Fig. 3** shows the flow chart explaining the whole process of hybrid simulation. Here some improvements

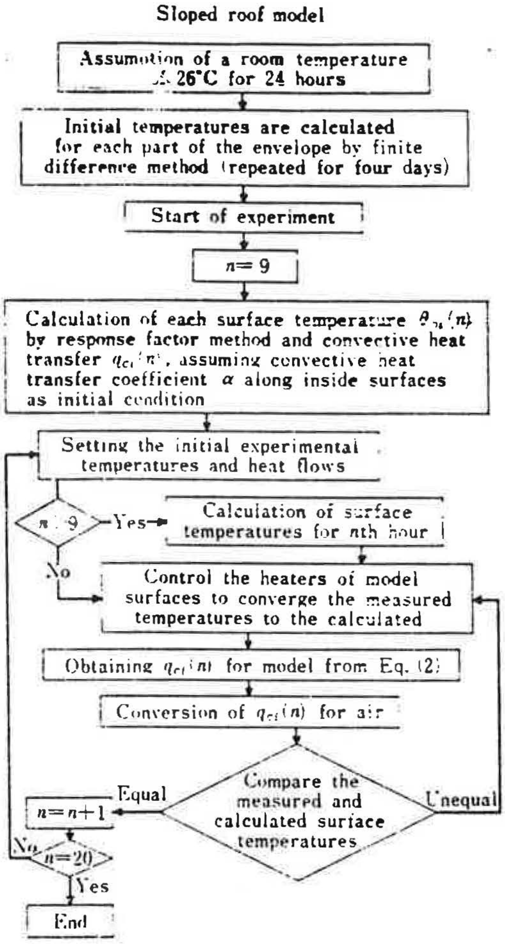


Fig. 3 Flow chart of the experiment program

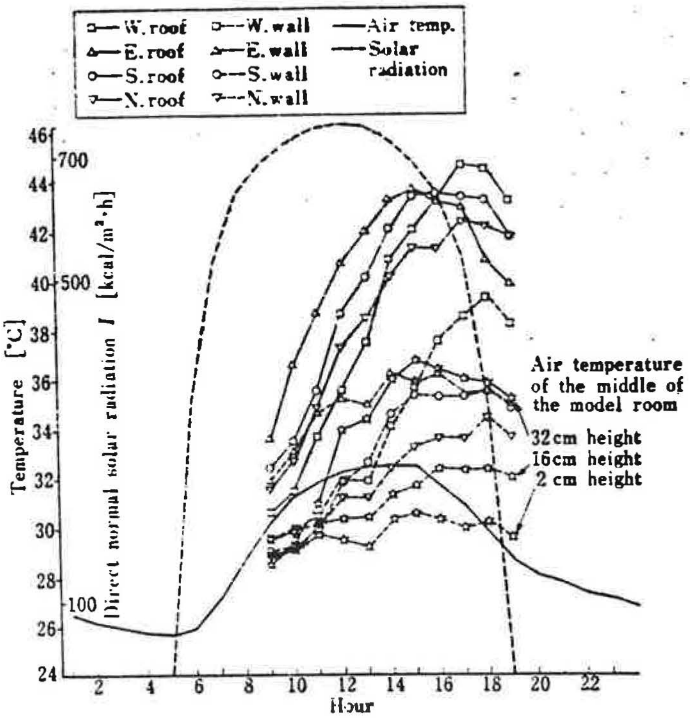


Fig. 4 Results of the measured inside air and surface temperatures of the sloped roof model against given outside air temperature and direct normal solar radiation

were made in the simulation process. The value of convective heat flow at the inside surface obtained from the experiment was directly put into the program instead of having the room temperature converged in the process previously conducted. This could make the convergence time shorter and the operation simpler. Inside surface temperatures in all walls and the sloped roof could be set to the values calculated on-line within a range of ±1°C. The conditions and the results of the experiment for a summer day in Tokyo from 9 to 19 o'clock are shown in **Fig. 4**.

2. Convective heat transfer coefficient

The heat transfer coefficient was again obtained by three main methods, as in the case of the flat roof model. The first method is based on *Ra* numbers already found by other researchers in similar cases for free convection. The Nusselt number may be found from the following equation[2].

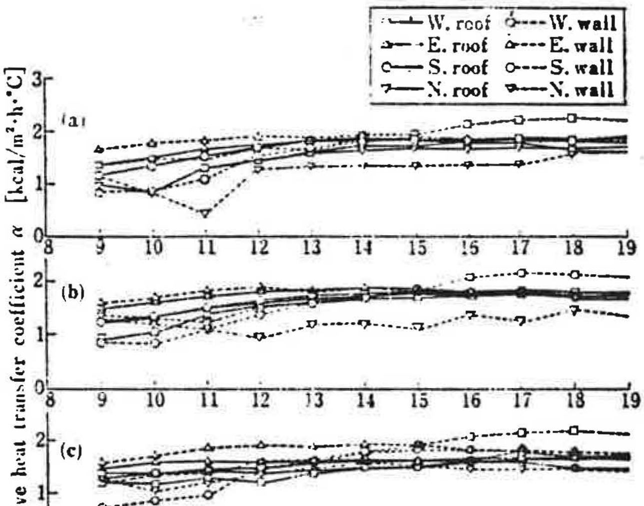


Fig. 5 Convective heat transfer coefficient obtained for free convection of the sloped roof model against, **(a)** 5 cm apart from surface, **(b)** 10 cm apart from surface, **(c)** middle of the room, converted to the values of air

$$Nu=c(Ra\cos\psi)^b \quad \cdots\cdots(1)$$

Here for the sloped surface of the ceiling for tilt angle of $\psi=30°$, it is accepted that $c=0.56$, and $b=1/4$. The results on the convective heat transfer coefficients are shown in **Fig. 5**.

The second method uses estimated heat conduction across acrylic plate from temperature measurement, to be equal to the convective heat flow at the surface as in the following equation referring to **Fig. 6**.

$$q_{ri}(n)=(\lambda_m/\delta_m)(\theta_m-\theta_0) \qquad \cdots\cdots(2)$$

The third method uses the heat flux at the conduction region within the boundary layer using equation (3), referring to **Fig. 6**.

$$q_{oi}(n)=(\lambda_w/\delta_1)(\theta_0-\theta_1) \qquad \cdots\cdots(3)$$

In this case the rate of conductive heat flow is obtained from temperature measurement at two places of each wall or roof surface: 1 mm distacne from the surface and 2 mm distance from the surface.

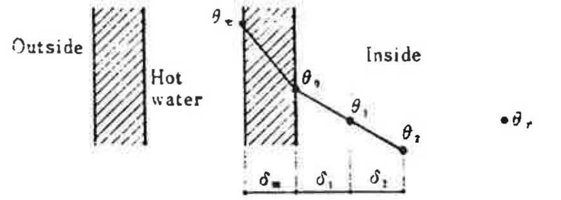


Fig. 6 Section of the wall showing the temperature measurement points

The rates of conductive heat flows thus obtained in the second and third methods were then applied to calculate the convective heat transfer coefficients against the room temperature at three points; 5 cm distance from the surface, 10 cm distance from the surface and 32 cm high above the floor in the middle of the model room. The following equations were used. For the second method

$$\alpha=(\lambda_m/\delta_m)(\theta_m-\theta_0)/(\theta_0-\theta_r) \qquad \cdots\cdots(4)$$

For the third method

$$\alpha=(\lambda_w/\delta_1)(\theta_0-\theta_1)/(\theta_0-\theta_r) \qquad \cdots\cdots(5)$$

and

$$\alpha=\{\lambda_w/(\delta_1+\delta_2)\}(\theta_0-\theta_2)/(\theta_0-\theta_r) \qquad \cdots\cdots(6)$$

The results are shown in **Figs. 7** and **8**. All the values of convective heat transfer coefficient shown in the figures are the ones converted to air. Where 1 mm distance of the conduction area was fixed more carefully, as in the case of 1 mm acrylic layer, a more stable graph of convective heat

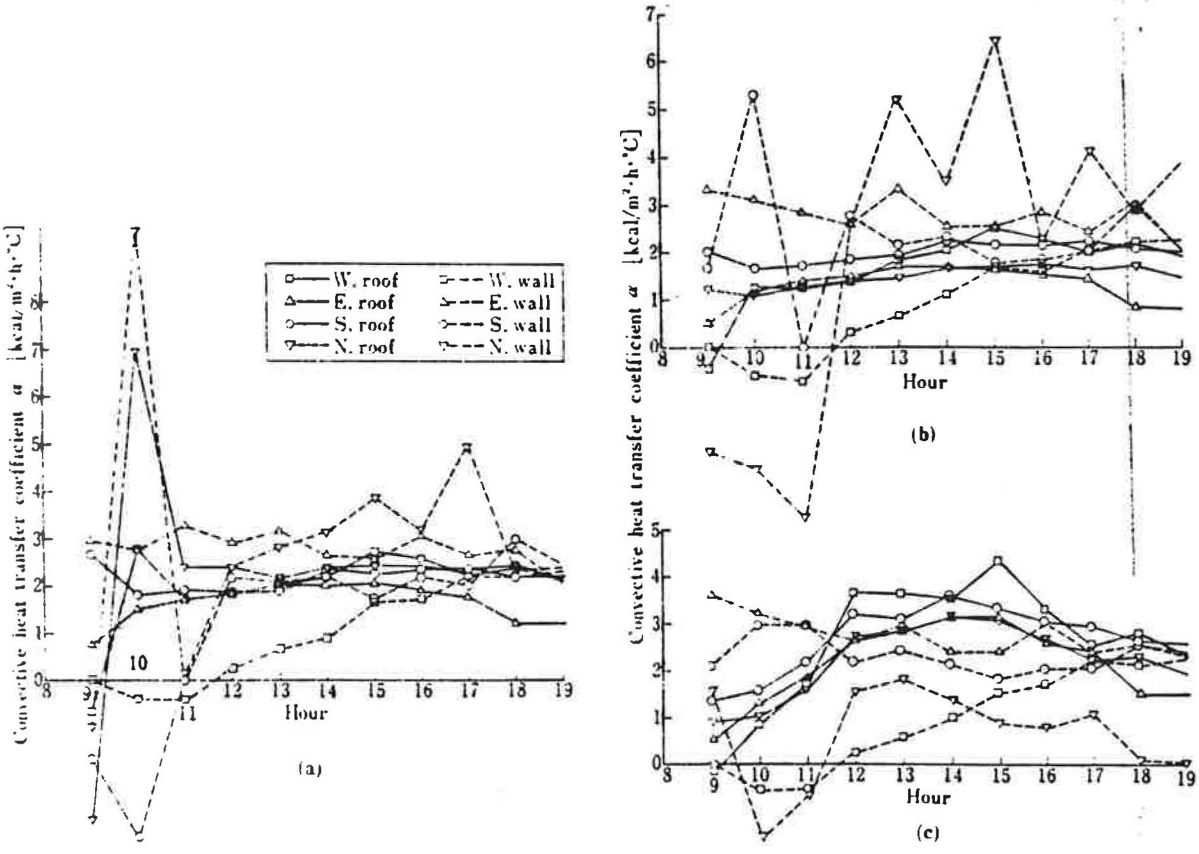


Fig. 7 Convective heat transfer coefficient obtained by conductive heat transfer across acrylic plate of the sloped roof model converted to the values for air, against (**a**) 5 cm apart from surface, (**b**) 10 cm apart from surface, (**c**) middle of the room

transfer coefficient could be obtained. The convective heat transfer coefficients are calculated assuming the thermal conductivity to be constant. Nevertheless as the thermal conductivity changes a little with the change of temperature when the surface temperature is very low, the convective heat transfer coefficient may have a maximum decrease of about 5%. This error could be considered negligible.

Comparison on the values of the convective heat transfer coefficients by above three methods gives a fairly good coincidence.

Comparing the sloped roof model with the flat roof model in the previous report, it is noted that the convective heat transfer coefficients along the wall surfaces turned out to be almost similar for both the flat roof and the sloped roof models, though the coefficients of the sloped roof were found to be a little higher than those of the flat roof.

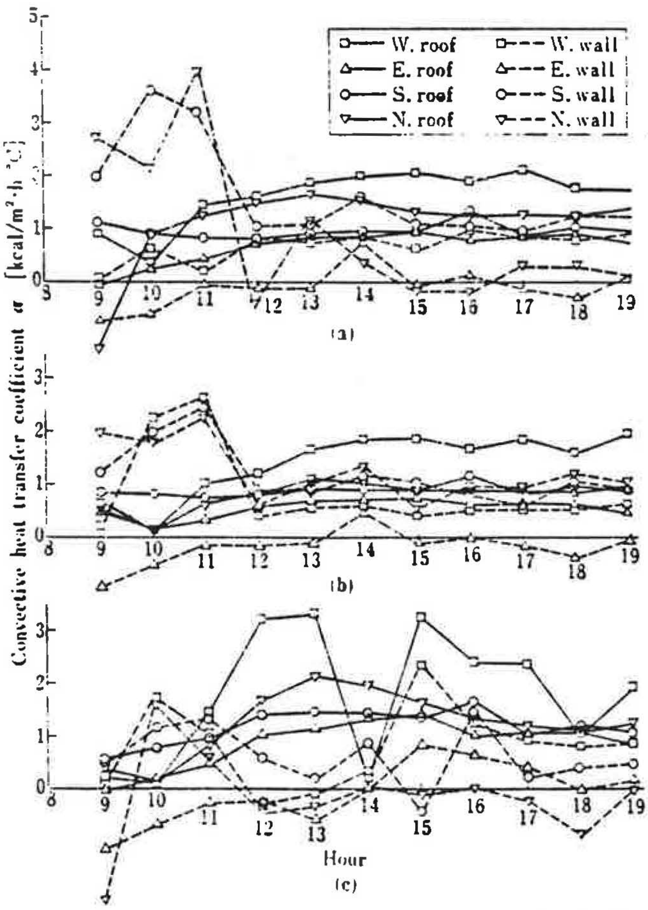


Fig. 8 Convective heat transfer coefficient obtained from thermal conductance of water through 2 mm distance from the surface of the sloped roof model converted to the values of air, against, (**a**) 5 cm from surface, (**b**) 10 cm from surface, (**c**) middle of the room

3. Flow pattern

Figure 9 indicates the flow pattern inside the model sketched from visual observation of the experiment, where red cubic plastic particles of 1 mm side length and the specific weight of 1.01 were used in order to keep the particles floating in the water. To compensate for the small difference of the specific weight, salt was added to the water.

As can be seen in the figures, the flow pattern changes with time as a result of the change in intensity of solar radiation and solar position. As the north and the east walls are warmed by the early morning sunshine, the heated air runs up along the inside surface of walls especially with higher speed along the corners and vertical edges of the walls and inclined edges of the roof, reaching the top opening.

Consequently, a circular current starts to occur in the lower part of the room by the entrance of cold water through the doors and in the direction of the cold to the warm wall [**Fig. 9** (**a**)]. At 11 o'clock [**Fig. 9** (**b**)], as the sunshine becomes stronger on the roof and inclines towards the south, the height of circular current increases.

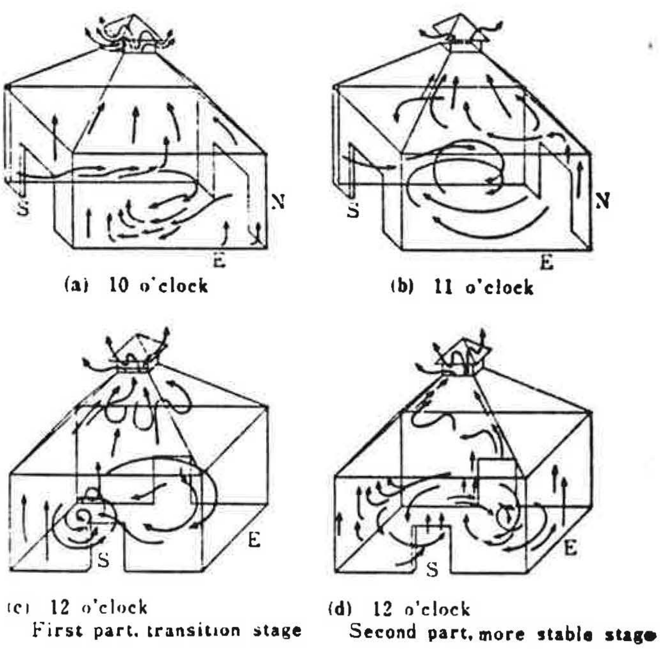


Fig. 9 Flow pattern of the water inside the model

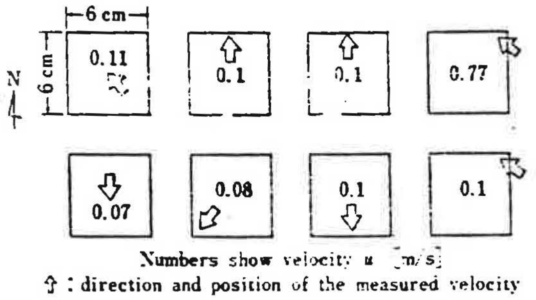


Fig. 10 Average values of the velocity of water at the top opening measured at 9 and 10 o'clock in different locations and directions

The air in the middle of the room which has become warm, rises up towards the top opening. At 12 o'clock the west roof and wall are warmed, but the south roof is still warmer, and an opposite current occurs in the transition stage [**Fig. 9 (c)**]. This causes a vortex when meeting with the main current ending with a more stable stage [**Fig. 9 (d)**].

4. Rate of natural ventilation by stack effect in sloped roof building

To calculate the ventilation rate of the room air the driving force of the stack effect is generally expressed by[3],

$$\Delta p=\gamma\cdot 273.16\cdot z(1/T_i-1/T_o) \qquad \cdots\cdots(7)$$

Here z is the height of the top opening from the middle of the bottom opening. Using the parameter ξ the air velocity at the outlet opening can be estimated from

$$u=\sqrt{2g\Delta p/(\xi\gamma)} \qquad \cdots\cdots(8)$$

The velocity of the outlet opening air thus obtained can in turn be used to calculate the air movement rate and the air change rate respectively from the following equations,

$$Q=3\,600\cdot u\cdot A \qquad \cdots\cdots(9)$$

$$m=Q/V \qquad \cdots\cdots(10)$$

The flow velocity in the 6×6 cm opening at the top of the sloped roof was measured by a hot wire type of water velocity-meter and the average values between 9 and 10 o'clock are shown in **Fig. 10**.

4.1 Total ventilation rate of the room

Assuming $\xi=2$ taken from a case similar to this experiment[4], the velocity and the ventilation rate for the sloped roof model were calculated using the

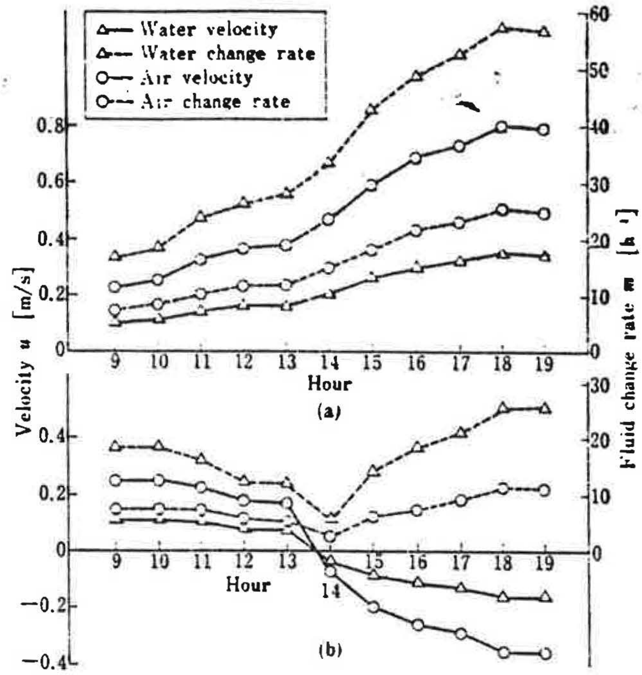


Fig. 11 Velocity and change rate of air and water, (a) at the top opening, (b) in the middle of the room (average of the values at 16 cm and 32 cm height in the middle of the model room)

temperatures of two places, (a) at the top opening and (b) in the middle of the room. The results are shown in **Fig. 11**.

The velocity obtained from the temperature of the middle of the room [**Fig. 11 (b)**] was found to become negative, which may be interpreted as reversal of the direction of air flow, around noon.

While the velocity of the top opening continuously increased until evening [**Fig. 11 (a)**. It is considered natural that the different values of ventilation rate calculated from these velocities should turn out to be different. The observations showed that the direction of flow at the top opening was all the time from inside to outside.

The reason for the difference between (a) and (b) in **Fig. 11** may be interpreted as follows. In the case of a factory, for example, where a high temperature body is placed in the room, the temperature of the middle of the room is high all day long and almost equal all over the room. Thus it is considered adequate to measure the temperatures of the middle of the room and outside air to calculate the air ventilation rate by the stack effect principle. On the other hand, in the case of natural ventilation caused by the warming up of the building enclosure, the temperature of the middle of the room is considerably lower than that of the vicinity of inside surfaces of the heated enclosure and the top roof opening. Thus it may be concluded that in such a case the temperature of the top opening could be used to calculate the velocity there, which also agrees with the measured results in **Fig. 10**, and then to find the total ventilation rate of the room.

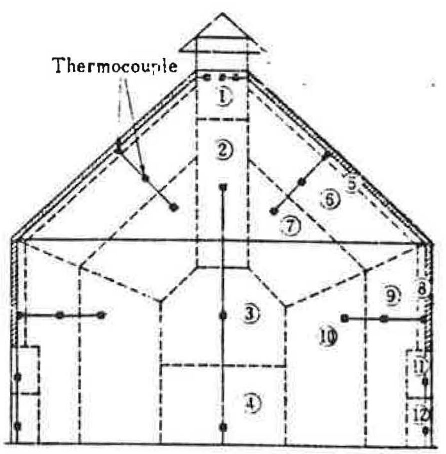
4.2 Approximate air movement rate in different parts of the room space

The above indicates that the air movement rate in the middle of the room is different from that of the enclosure vicinity and that the negative velocity of **Fig. 11** (**b**) can be explained to suggest the change of fluid movement direction in the middle of the room. This also agrees well with the results of visual observations.

Thus in order to find the flow rate all over the room in the case of the sloped roof model, the room space was divided into 32 small volumes as shown in Fig. 12. Equation (7) was applied to calculate the pressure difference between each part of the room and the outside air. Then the velocity, air movement rate and air change rate were calculated from equations (8), (9) and (10), for each small part of the room, using v_i instead of V. The results of calculations about velocity and air movement rate are shown in **Figs. 13** and **14.**

It is obvious from the velocity curves in **Fig. 13**, that the air current direction changes from around 10 to 15 o'clock, except for the east and south roofs, east wall and the top opening. Around these, the air is constantly warmer than the outside air and rises towards the outside air through the top opening, thus deriving a cold air draft from outside through the doors to replace it. At 0.8 m height in the middle of the room for example [**Fig. 13** (**d**)], in which air does not become warmer than the outside air until 14 o'clock there is a tendency for air to go out through the doors. At the same points after 14 o'clock, the direction of velocity changes.

The zone very near the floor such as 0.1 m height in the middle of the room and the lower parts of the doors can be indicated to have an upwards current in the beginning and later on are drafted inversely downwards. While at the upper parts in the vicinity of the doors as well as at 0.1 m height in the middle of the room the opposite phenomenon occurs. In most parts of the room, as shown in **Fig. 14**, air movement rates decrease to zero from about 12 to 15 o'ciock during which their velocity directions change as shown in **Fig. 13**, and there is a pause in the transition stage.



① Mid-room 45 cm (2.54 m air)
② Mid-room 32 cm (16.6 m air)
③ Mid-room 16 cm (0.8 m air)
④ Mid-room 2 cm (0.1 m air)
⑤ 2 mm (1 cm, air) from the roof
⑥ 5 cm (26 cm, air) from the roof
⑦ 10 cm (52 cm, air) from the roof
⑧ 2 mm (1 cm, air) from the wall
⑨ 5 cm (26 cm, air) from the wall
⑩ 10 cm (52 cm, air) from the wall
⑪ Door up (north)
⑫ Door down (north)

Fig. 12 Cross section of the model showing the divisions of the room space

The graphs of **Fig. 13** and **14** show that the air current pattern is uneven within the room area and along the day. There may be a high ventilation going on at the top opening while in the middle of the room at the same time the air is still.

4.3 Average ventilation rate of the room

In order to make a rough estimation of the indoor climate of a natural ventilation system in this experiment it is necessary to find an average rate of the ventilation as for a whole room space. The following equation may be applied to estimate the average pressure difference of the room.

$$\Delta p=(1/V)\sum_{i=1}^{32} h_i(\gamma_i-\gamma_o)v_i$$
$$=(1/V)\{h_1v_1(\gamma_1-\gamma_o)+h_2v_2(\gamma_2-\gamma_o)+\cdots\cdots+h_{32}v_{32}(\gamma_{32}-\gamma_o)\} \quad \cdots\cdots(11)$$

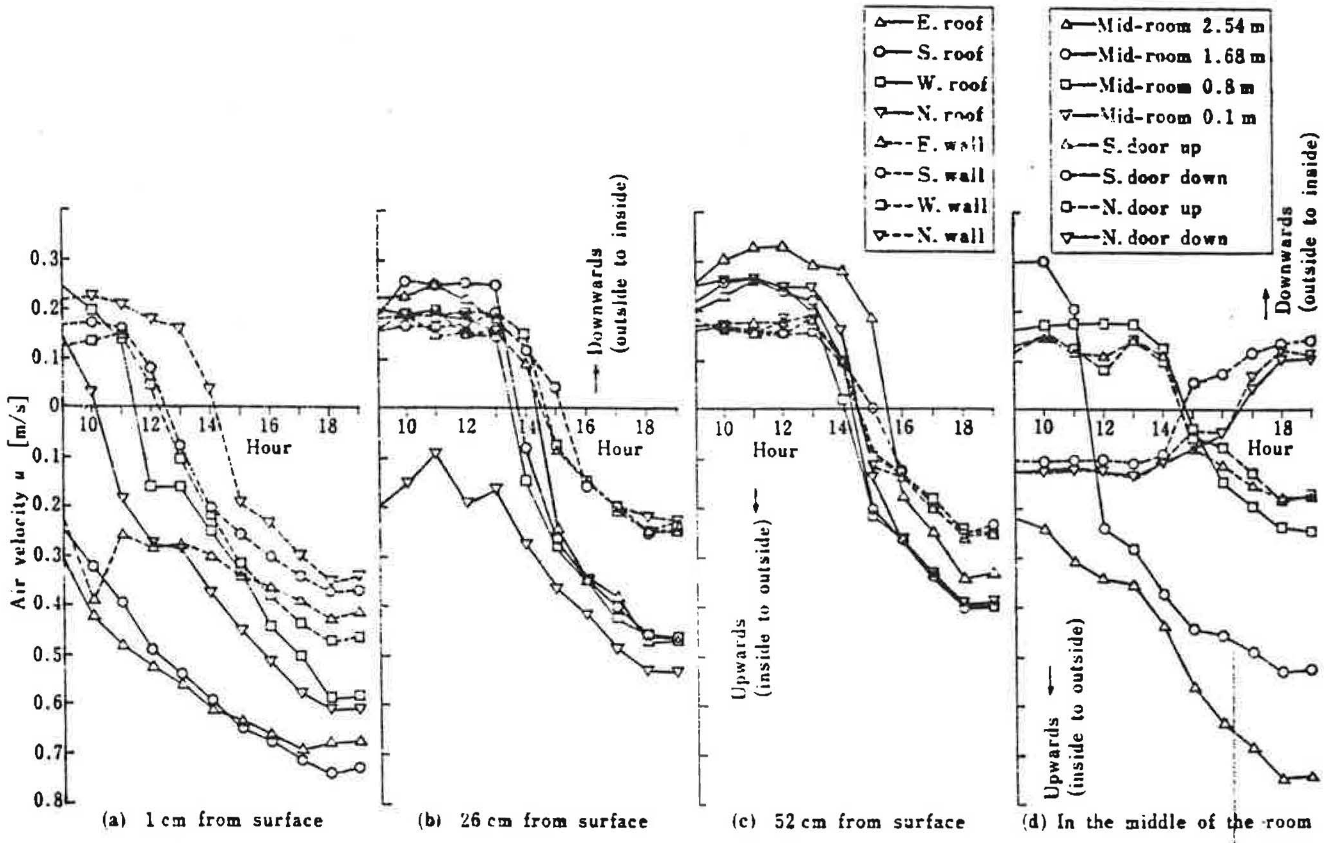


Fig. 13 Velocity, obtained for 32 small volumes of the sloped roof model room space, converted to the values of air

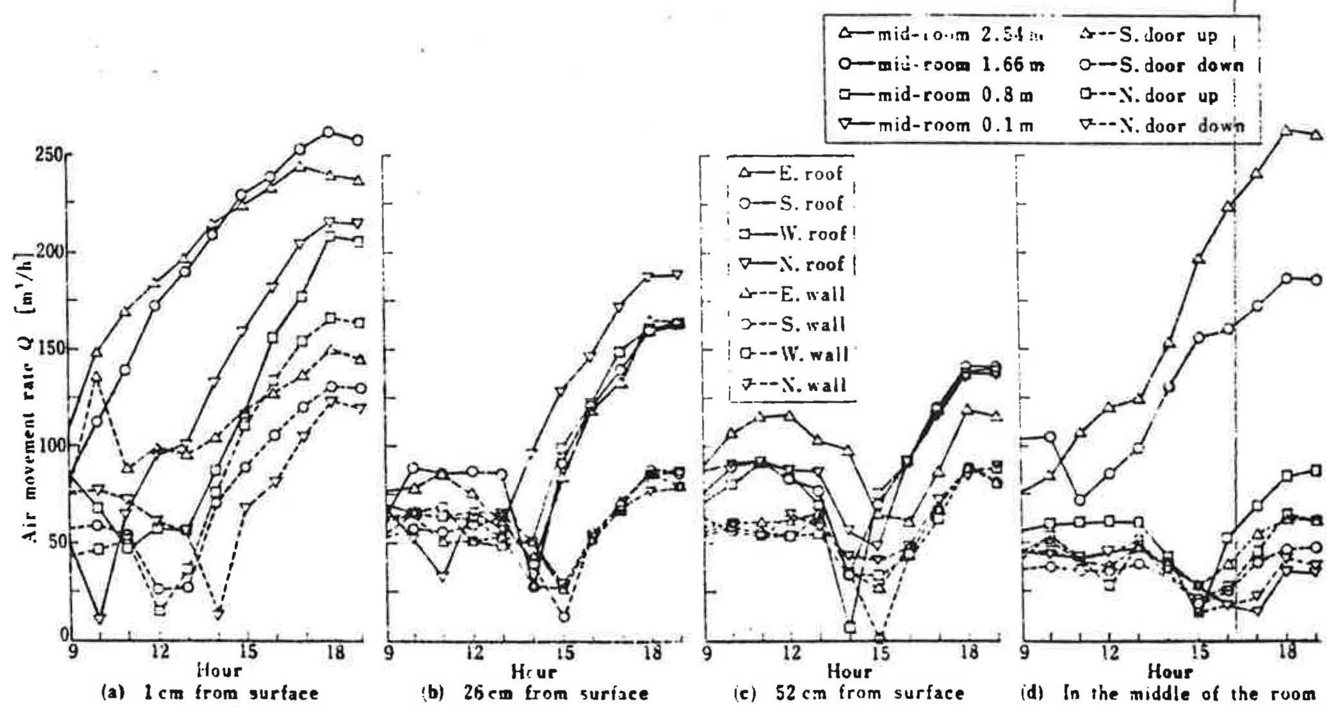


Fig. 14 Air movement (ventilation) rate obtained for 32 small volumes of the sloped roof model room space

The average values of air movement rate, air change rate and velocity were then calculated from the average pressure difference, again using equations (8), (9), and (10). The results are shown in **Fig. 15**. The air movement rate thus obtained could be considered as a representation of the ventilation all over the room space.

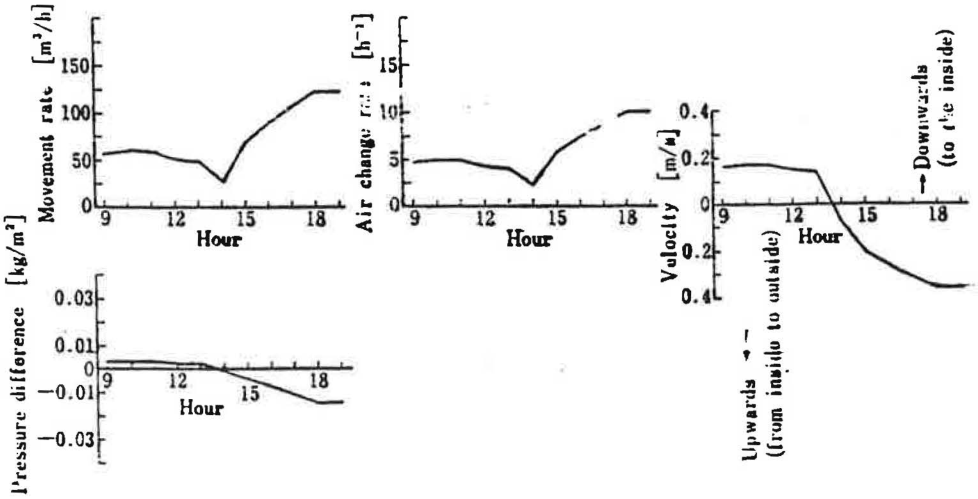


Fig. 15 Average movement rate, fluid change rate, velocity and pressure difference of the sloped roof model room space converted to the values of air

4.4 Consideration

The assumption of pressure loss coefficient ζ in **Figs. 11 to 15** may bring about a certain error in air change rate, air velocity and movement rate. This error can be considered to be very small because the coefficient is taken from a case similar to that of the model top opening, and the flow velocity measured at the model top opening agreed with the velocity calculated using the above coefficient.

Moreover, the above coefficient does not have any effect, when comparison is made between the ventilation or movement rate of different parts of the room.

Conclusion

Hybrid simulation for a natural ventilation system was applied and actual attempts were made with a scaled down model with sloped roof as well as a model with flat roof in the previous report, using water as fluid media. The major findings obtained from the experiments were as follows;

1) There was a general agreement on convective heat transfer coefficients obtained from free convection theory and those from temperature measurement within the boundary layer and across the acrylic plate. Thus the coefficients are considered reliable and useful for real buildings of a similar configuration and climatic conditions.

2) Convective heat transfer coefficients along the wall surfaces turned out to be almost similar for both the flat roof and the sloped roof models, though the coefficients of the sloped roof were found to be a little higher than those of the flat roof.

3) The room temperature differences among upper, middle and lower parts of the middle of the room were found to be greater in the case of the sloped roof model. This would indicate a higher degree of air mixture in the flat roof model, as the sloped roof surface might let the fluid slide up over the surface more easily.

4) The flow pattern of the room air changes with time, and the direction of the air circulation changes from around noon.

5) In a ventilation system based on natural convection caused by warming up of the building enclosure due to the solar radiation and the outside air temperature, it was not considered suitable to estimate the ventilation rate by the stack effect method commonly used. This is because there is a great difference between the temperatures of the middle of the room and the outlet opening. It was considered adequate for estimating the total ventilation rate of the room to calculate pressure difference between the outlet opening and outside air, using the temperature difference between the outlet opening and the outside air.

6) Dividing the room into several small volumes, the air movement rate of each division was found using the pressure difference between the small division and outside air. This may be considered useful to estimate the rate of air movement within the room space. The average values of air movement rate, velocity and air change rate made between all of the small divisions of the room were also obtained. It is interesting to see the difference between total ventilation rate and average air movement rate in this natural ventilation system.

References

1) R. Taheri, Y. Miyagawa, K. Kimura and T. Tanaka : Hybrid Simulation of a Ventilation System based on Natural Convection (Part 1) Proposal of Hybrid Simulation Process and Attempt with Flat Roof Model with Four Vertical Walls, SHASE Trans., No. 34 (1987-6), pp. 51~61
2) L. P. Holman : Heat Transfer, 5th edition, (1981), p. 282, MacGraw Hill
3) K. Kimura : Scientific Basis of Air Conditioning, (1977), pp. 175~179, Applied Science Publishers, London
4) Y. Miyagawa : Rerearch on Inside Room Environment of the Model Building of Air Membrane (No. 1) Model Experiment of the Characteristics of Exhaust Smoke Damper (in Japanese), Ohbayashi-Gumi Technical Research Institute, (1984-1), pp. 3, 9

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自然対流による換気系のハイブリッドシミュレーション

第2報——傾斜面屋根の模型による試行と換気量の推定

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キーワード：ハイブリッドシミュレーション・自然換気系・対流熱伝達率・流形・換気量・傾斜面屋根の模型・レイリー数

既報において説明した方法により，アクリル板製の傾斜面屋根の模型を水槽に入れ，ハイブリッドシミュレーションを行った．実験から室内側対流熱伝達率が求められ，既報において平らな屋根について求めた対流熱伝達率の値と比較した結果，壁についてほぼ一致し，屋根については，傾斜面のほうが水平面より大きくなることがわかった．一般的な煙突効果による換気量の計算法を日射や外気温などによる換気系に適用し，開口部温度と外気温から内外圧力差を求め，上部開口部の流速を推定したが，この値は流速の測定値とほぼ一致した．この流速から全換気量を求めた．また，外気温と室内温度による気圧差の推定から室内各部の空気流量を求めた．一方，微細粒子を入れて可視化実験を行い，流れの状況を観察した．その結果，室内各部の空気流動は時間によって変化することを認め，測定結果と照合することができた．この自然換気系では工場などの高温発熱体による自然換気の場合と全く異なり，室内各部流量の平均値と下部開口から上部開口への全換気量とは異なることを定量的に示した．

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