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A Laboratory Experiment on Natural Ventilation of a Roof Cavity

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Synopsis: A laboratory experiment was carried out on the natural ventilation characteristics of a roof cavity focusing on the lessening of solar heat transfer through the roof. Experimental outcomes resulted from measurements in a cavity model heated at the top side in order to mimic a solar radiation incident on the roof. Inlet and outlet openings were narrowed to simulate obstacles along the cavity and were then grouped into three cases based on the obstacle placement. Two inclination angles at 20 and 30 degrees were examined. The cavity surface temperatures, velocity and temperature profiles in the airflow were measured and heat transported by the airflow was examined. The results showed that the heat-dissipation performance of natural ventilation was greatly influenced by the opening size. It was found that average velocity reached to 0.25 m/s in the experiment. The corresponding maximum flow rate was 7.7 L/s with a turn of over of 3.0 times per minute. The highest found Reynolds number was 2009.

Keywords: Roof Cavity, Natural Ventilation, Opening Slit Size, Opening Configuration

1. Introduction

Among building-planning, methods of climate control are one of the fastest growing branches of new energy demand [1]. By today's technology, climate control can be accomplished simply by using mechanical measures. However, these measures have endangered economic and environmental principles. Therefore, investigations of passive climate control technologies for buildings are of continued interest [2-4].

By far the most effective measure for structural climate control is to keep solar heat out. It is more viable to prevent heat transfer through building claddings than to evacuate the penetrated heat by way of cooling installations. When it comes to a factory structure which is widely spread and has a relatively low height, the heat transfer characteristics of the roof have an exclusive influence on the thermal environment as well as the thermal load beneath it. In particular, solar irradiation on the roof outweighs the cooling load substantially, making the work space below unbearably hot, which in turn causes a reduction in work efficiency and precision. Therefore, in order to improve roof performance there needs to be a viable method for the achievement of factory working-space climate control.

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2. Background

Using a rooftop as a solar collector is a widely-known method today. Khedari *et al.* [5] conducted research on the use of a roof solar collector (RSC) to enhance natural ventilation of the space beneath. Some configurations of roof solar collectors were investigated for residential applications. They found that the RSC made from gypsum board has a better performance than plywood in resisting the heat loss. Moreover, in order to optimize the natural ventilation in the length of the RSC, it should be shorter to the order of 1 m. Khedari *et al.* [6] also investigated the effect of RSC airspace size on the air flow rate. A comparison between four types of RSC showed that with equal restrictions, airspace with a thickness of 140 mm enabled a higher induced air-flow rate compared to air space with a thickness of 80 mm. Hirunlabh *et al.* [7] studied four new RSC configurations using a validated numerical model. The analysis showed that the highest air volume flow rate was 0.0206 m^3/s per 1 m^2 of RSC when using the new RSC configuration.

In a numerical study of sun-shading on walls and roofs by Akasaka *et al.* [8], it was found that cavity ventilation can lessen rooftop heat gain by 50%, and wall heat gain by 40%, with the exception of north oriented walls.

In a simulation study of solar-assisted ventilation by Adam *et al.* [9], the ventilation effect was calculated by dividing a cavity into blocks along its flow, and then seeking a simultaneous balance of heat and buoyancy. It was pointed out that the height of a heat-absorber should be limited when indoor air temperature is lower than outdoor temperatures.

Most research on naturally-ventilated rooftops is aimed at relatively short roofs. With a longer roof, there is a sizeable vertical difference between the inlet and the outlet. This allows for a considerable buoyant force to act on the air in the cavity during solar irradiation transference. This buoyancy can be utilized favorably in order to dissipate irradiated solar heat outdoors by way of natural ventilation. But to the best of the authors' knowledge, there is no data existing on the characteristics of natural air flow through an elongated roof cavity, which is the focus of this paper.

The ultimate intention of the present study is to figure out how to reduce heat penetration to lower surfaces through ventilation of the cavity air. Lowering the temperature of the lower surface decreases the heat transferred further down to the occupied zone beneath it.

In this study, the characteristics of natural ventilation through a roof-cavity were examined experimentally using a heated, inclined duct, which was open to the outside at the upper and lower ends, inside a laboratory.

In practical applications, inlet or outlet openings of a cavity are usually covered with guards such as a grid or a net in order to protect the cavity from rainwater, insects, birds or falling leaves. Structural elements in a cavity, rafters for example, also increase resistance against the airflow. To cope with this fact, the present study examined the effect of restrictions at the openings of the roof-cavity.

The aims of this study were:

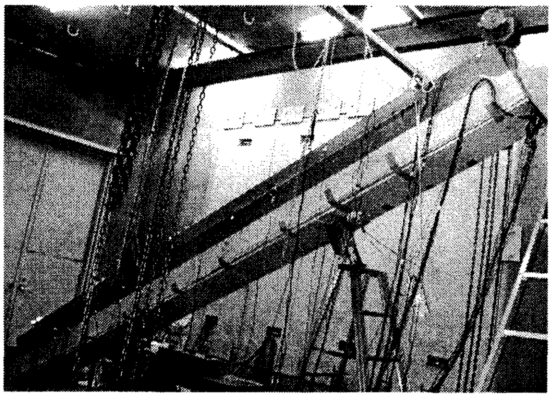
(1) To clarify the characteristics of buoyant airflow in a roof cavity,
(2) To obtain data on heat and air flow through the cavity; and
(3) To examine how restrictions at the openings influence cavity ventilation performance.

2. Experimental Arrangement

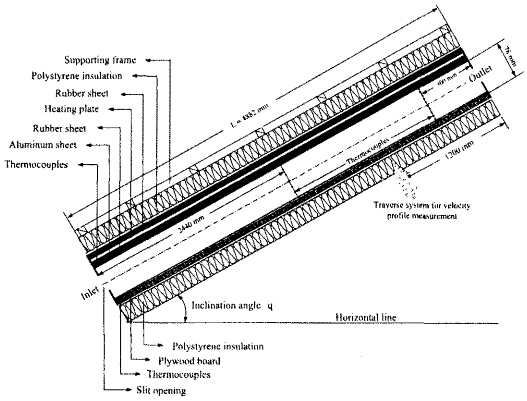
2.1 Experimental Set-Up

An experimental study was carried out in a laboratory to examine the detailed behavior of air in a roof-cavity under steady-state conditions. The experimental model is a simple rectangular duct section, which has internal dimensions with a length of 4882 mm, a width of 400 mm and a depth of 78 mm. The two smallest opposing sides were open to the laboratory as the air inlet and outlet. The other sides were constructed of plywood with a 12 mm thickness and were covered with foamed polystyrene thermal insulation boards with a 50 mm thickness. A photograph of the experimental model and a sectional diagram of the cavity are displayed in Figure 1 (a) and (b).

Six electric heating plates were mounted along the upper side of the duct along its length. The plates were connected to an AC power supply and the power input was adjusted individually for each plate. The upper and lower sides of the heating plates were covered with rubber sheets with a 5 mm thickness in order to measure the heat flux across them. The insulation board above the heater kept the generated heat located on the lower side. The lower surface of the rubber sheet was covered with aluminum foil in order to reduce radiation from the surface. 18 T-type thermocouples were inlaid between the rubber sheet and the aluminum foil to measure the surface temperature distribution along the centre-line.



(a) Experimental model



(b) Sectional diagram of cavity model

Figure 1 Cavity model

At a location of 300 mm from the outlet, 15 T-type thermocouples were arranged across the cavity to measure the centreline temperature profile of the outgoing air. The other five were placed in the middle of the cavity length in order to measure middle-temperature profiles. The ambient temperature was measured at four different heights inside the chamber where air temperatures would not be altered by warm air escaping from the cavity. All the thermocouples output signals were acquired with a multi-channel data logger with a 0.1 °C precision. The data were recorded with a PC.

A velocity profile was measured on the centreline at a distance of 1200 mm from the outlet. An anemometer probe (Model 6201, Kanomax) was attached to a traverse system, and the velocity profile was read at 19 locations spaced 4 mm apart across the cavity. It was assumed that the velocity profile was most representative at this location. The anemometer probe has a heated sphere with a 2.5 mm diameter. The accuracy was 0.02 m/s according to the manufacturer's manual. The velocity measurement results were transferred to the PC.

The examined heat productions were 150, 100, 75 and 50 W/m^2. These values were selected when considering that the top cover of the roof-cavity is black painted steel plate with a solar radiation absorptivity of 0.95.

To cope with resistance for airflow inside the cavity, the inlet and outlet openings were narrowed with aluminum plates leaving vent-type openings with widths of 10, 20, 35 and 78 mm (the last size was equal to the depth of the duct).

The whole cavity-model assembly was supported by a framework which was suspended from the laboratory ceiling. The cavity model could be shifted to whichever angle was required in the frame. The inlet of the cavity was located at 375 mm above the laboratory floor. The examined angled inclines were 20 and 30 degrees from the horizontal line.

The experiment was conducted in a laboratory with a length of 6.5 m, a width of 6 m, and an angled roof with a height of 4.5 m to 5.5 m. The walls and roof had a heavy thermal inertia which reduced inside temperature fluctuation caused by daily outdoor temperature changes.

2.2 Experimental Procedure

After the cavity was shifted to the required angle, experiments were carried out with different levels of heat production and different opening configurations. The different combinations of vent sizes were grouped as shown in Table 1. Case A represented a restriction at the outlet opening while the inlet opening was fully open. Case B meant that the inlet opening was restricted while the outlet opening was fully open. In Case C, both the inlet and outlet openings were restricted equally. The fully open case (restricted neither at the inlet nor outlet) served as a reference. After changing the configuration, the input power was adjusted to attain an equivalent heat flux at the six heaters. The model was allowed to run at least 20 hours until a thermally steady condition was achieved. When a steady condition was achieved, referring to the readings of the temperatures in the model, the temperatures and the velocities were recorded. Experiments with some of the configurations could not be performed because the heater temperature rose too high. When the heat flux was high, the openings were small, or the angle of the incline was gradual.

3. Experimental Results

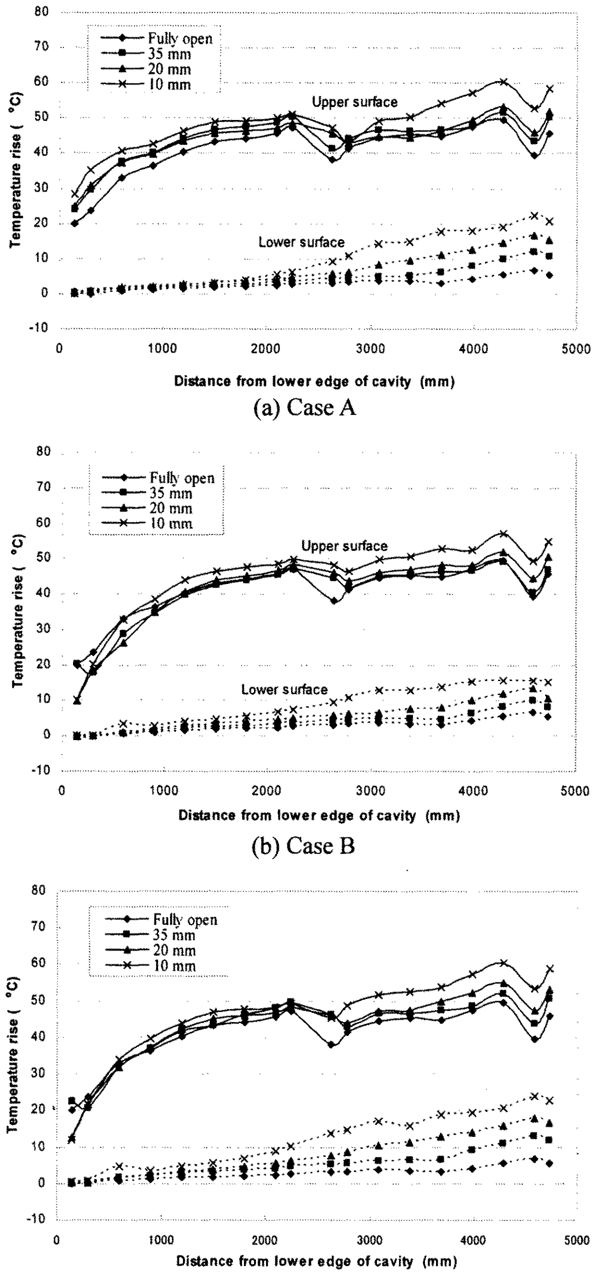
3.1 Cavity Surface Temperatures

Experiments for all conditions took several months. During these months the daily laboratory temperature changed. Thus in the following discourse, all the temperatures are referred to as the average laboratory temperature of respective measurements, and subsequent increases were discussed.

Figure 2 (a), (b), and (c) show temperature distributions along the centerline of upper and lower surfaces in the cavity for Cases A, B and C, respectively. During this experiment, the cavity was angled at 30 degrees, the heat production was set at 150 W/m^2, and the openings were changed from fully open to 35, 20 and 10 mm, respectively.

In general, temperatures at both the upper and lower surfaces increased depending on their distance from the inlet. This demonstrated that the stack effect worked well in the cavity. But the increases were not linear along the cavity length, particularly at the upper surface. It was observed that temperatures at the upper surface increased rapidly within the first 2250 mm. Beyond this point, the line of temperature increases became gradual but irregular, and there were distinct temperature drops at several positions. The drop was caused by a discontinuity of heaters at these locations.

When the opening size was substantial, the temperature was



(a) Case A

(b) Case B

(c) Case C

Figure 2 Temperature distributions along the centerline of upper and lower surfaces of the cavity tilted at 30° and a heat production of 150 w/m^2 for various opening sizes

minimal on both the upper and lower surfaces. But when the opening was narrowed, the temperature increased in all of the cases. This was especially distinguishable in the areas closer to the outlet.

Restrictions placed at the openings greatly influenced the temperature distributions on both surfaces. The smaller the opening, the higher the temperatures. In particular the increase was sizeable on the lower surface, thus decreasing the difference in temperature between the two surfaces. In Case C, the temperatures of the upper and lower surfaces tended to approach each other when the slit size was reduced. For Case C, the average temperature of the lower surface was 2.8 °C in the fully

Table 1 Experimental Conditions

Heat production W/m^2	Inclination angle	Experimental Case	Inlet slit size mm	Outlet slit size mm
50, 75, 100, 150	20°, 30°	A	78	35
				20
				10
		B	35	78
			20	
			10	
		C	35	35
			20	20
			10	10
		Fully open	78	78

open case. This increased to 5.5°C, 7.9 °C and 11.8 °C, respectively with reduction in the slit sizes of 35, 20 and 10 mm.

3.2 Temperature Profiles in Cavity Air

Figure 3 shows the outgoing temperature profiles of the air flow, when the heat production was 150 W/m^2 and the inclination was 30 degrees.

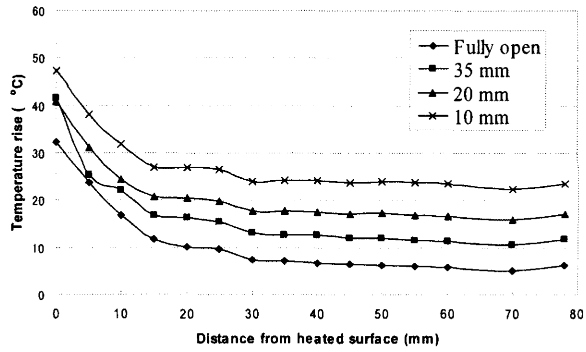
In all cases the temperature decreased rapidly as the distance from the heated surface increased. The gradient lessened as the position approached the unheated surface. This indicated that air in the closer vicinity to the heated surface had lower density than of the air in its surroundings. The temperature at the nearest position to the unheated surface increased slightly. This was due to radiation heat transfers from the upper heated surface to the lower unheated surface.

The average temperatures of the airflow were shown for all cases in Table 2. In the fully open case, 150 W/m^2 of heat input and a tilt angle of 30°, the average temperature was 10.6°C. This increased to 17.5 °C, 22.3 °C, and 30.0 °C with slit sizes of 35, 20 and 10 mm, respectively, in Case C. Generally, the average temperature was at the lowest at the fully open case, and increased in the order of Case B, Case A and Case C.

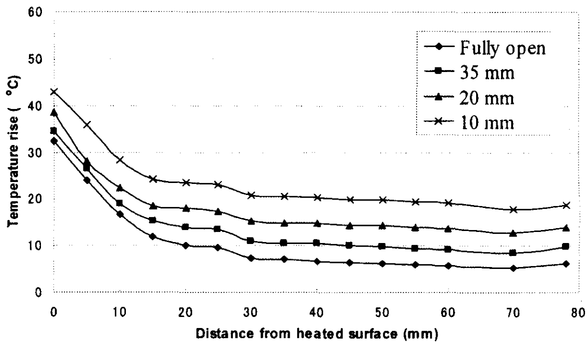
3.3 Velocity Profiles

Velocity profiles in the heat production of 150 W/m^2, tilted at a 30 degree angle were shown in Figure 4. Rapid velocity occurred in the region near the heated surface. The velocity gradually decreased as the position neared the unheated surface. The hotter air near the upper surface moved faster while the cooler air near the lower surface moved slower. The fastest velocity was found at a distance of around 8 mm from the heated surface. In all of cases, when the vents were set smaller, the lines of the velocity profile were reduced accordingly. The velocity profiles of Case A were higher than those of Case B, and the velocity profiles of Case C were the lowest among them.

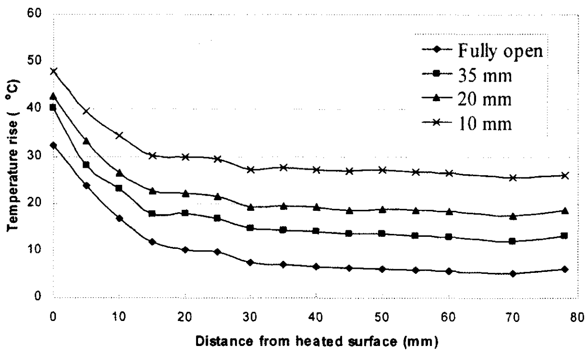
Table 3 showed average air velocities in the cavity for all the cases. The highest average velocity was 0.25 m/s in the measurement with a heat production of 150 W/m^2, both openings were fully opened and at an angle of 30°. There was a clear relation between the air velocity and the vent sizes. The reduction of the opening size subsequently reduced the air velocity. In Case A and Case C, the air velocity tended to decrease when the heat production decreased. However in Case B, a straight order rule of thumb seemed hard to define.



Case A



Case B



Case C

Figure 3 Outgoing temperatures distributions across the cavity gap tilted at 30° and a heat production of 150 W/m^2

It has been speculated that a secondary counter flow took place at the top opening because the outflow was excessive there, but there was not enough air supplied from the bottom opening. So the results of the present experiment can not be applicable for a roof cavity with a large outlet and a small inlet.

3.4 Flow Rates

The flow rates through the Case C cavity at the two inclination angles are shown in Table 4. These rates were calculated from the velocity profiles and the sectional area of the cavity, with an assumed two-dimensional flow. The decrease in the inclination angle affected the decrease in flow rates.

The flow rates were strongly influenced by the level of heat production. The stronger the heat production, the higher the flow rate. The maximum flow rate was 7.7 L/s in the experiment with a heat production of 150 W/m^2, a tilting angle of 30 degrees and two unrestricted openings. This means that the cavity was ventilated 3.0 times per minute. The Reynolds numbers were calculated from the average velocity and the equivalent diameter of the square duct, both of which are shown in Table 4. The largest Reynolds number for 2009 was found in the above experiment. The Reynolds numbers indicated that the flows were in a laminar flow range in all the experiments.

3.5 Accuracy Examination

Table 5 shows the portions of the heat balance in Case C. The produced heat was expected to dissipate in the following three ways:
(a) Transport by ventilation.
(b) Transfer through the top structure.
(c) Transfer through the lower structure.

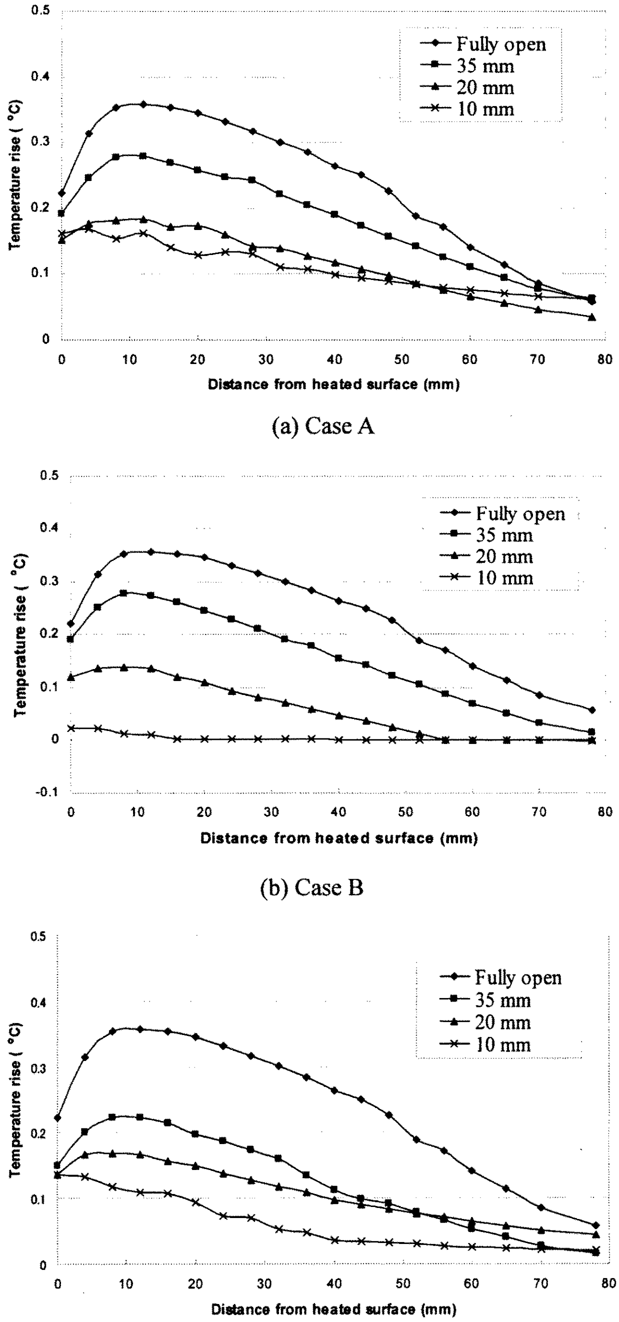
Heat production and transfer through the upper and lower structures were calculated on the surface from the entrance to the temperature profile measurement position of 4.582 m. Heat loss through the upper and lower structures was calculated from the measured temperature differences and their overall heat transfer coefficients. Heat transportation through ventilation was calculated from temperature and velocity profiles, assuming that the velocity profile was kept to the temperature-profile measurement area. Heat loss through the side walls was considered inconsequential because the amount was so minute.

Heat dissipation by ventilation was remarkably effective. But a large portion of the produced heat was transferred through the top structure. The bottom structure released a small portion of the heat. The deviation between the produced heat and the total of three heat dissipations was allotted to the error, which was shown in the last column of Table 5. A positive error meant that heat production was larger than total heat losses, and vice-versa. The errors were between -36 and 18% and the average was -15%. Error proportion was smaller in the larger heat production experiment than in the smaller heat production experiment. The examination of heat balance in the cavity indicated that high accuracy was limited to high heat production and small opening restriction. In the experiments with opposite conditions, the accuracy was smaller.

4. Discussions

In natural ventilation through a roof cavity, the velocity might be slow and it seemed to belong to laminar flow judging from the Reynolds Number (See Table 4). But it is mentioned that fluid may have to travel a length equal to 60 times the diameter of a pipe before the stable pattern of flow corresponding to the particular Reynolds number is established[10]. Until this distance, the flow includes turbulence, which may be brought in by the entering air, and may be caused by the flow deformation at the entrance. This suggests that the flow includes turbulence when it enters into a cavity, and it transits into laminar flow gradually along the pass.

The flow of the present experiment was supposed to include turbulence throughout its length. As a result, the viscosity and the thermal conductivity of the air, which are required to calculate the friction resistance and the air temperature in the cavity, must be treated as of a turbulent viscosity and a turbulent thermal conductivity instead of the molecular viscosity and the molecular heat conductivity.



(a) Case A

(b) Case B

(c) Case C

Figure 4 Velocity profiles of the cavity tilted at 30° and a heat production of 150 w/m^2 for various opening sizes

It was difficult to satisfy the similarity laws of flow and heat transfer simultaneously in the present experiment. So a live scale experiment was planed with a length of 4.882 m. It was intended to clarify the friction resistance of flow along the cavity wall from this experiment, provided that resistance by deformation along the flow channel could be decided from the existing publications. If the friction resistance was known, the resistance of a longer cavity could be estimated from this result.

It is indicated that the viscosity and thermal conductivity

Table 2 Average temperatures rise of air flow of three cases

Inclination	Heat Production (W/m^2)	Temperature rise (°C)											
		Case A				Case B				Case C			
		Outlet slit size (mm)				Inlet slit size (mm)				Both slit size (mm)			
		78	35	20	10	78	35	20	10	78	35	20	10
30°	150	10.6	16.2	20.7	27.2	10.6	13.9	17.9	23.4	10.6	17.5	22.3	30.0
	100	7.1	12.1	16.7	19.5	7.1	12.4	13.6	17.3	7.1	12.7	17.5	22.7
	75	6.7	9.6	13.9	15.9	6.7	8.9	11.4	14.0	6.7	10.5	14.1	17.8
	50	5.2	7.5	10.1	11.6	5.2	6.3	8.4	10.1	5.2	8.0	10.5	12.9
20°	150	12.2	17.1	21.8	-	12.2	14.9	19.0	-	12.2	18.5	-	-
	100	9.5	12.5	16.7	-	9.5	11.2	13.8	15.0	9.5	13.5	-	-
	75	7.8	10.2	13.3	-	7.8	9.2	11.2	12.3	7.8	10.6	-	-
	50	6.1	7.2	9.6	-	6.1	7.1	8.5	9.0	6.1	7.8	-	-

Table 3 Average velocities of air flow of three cases

Inclination	Heat Production W/m^2	Average velocity (m/s)											
		Case A				Case B				Case C			
		Outlet slit size (mm)				Inlet slit size (mm)				Both slit size (mm)			
		78	35	20	10	78	35	20	10	78	35	20	10
30°	150	0.25	0.19	0.12	0.11	0.25	0.16	0.06	0.01	0.25	0.13	0.11	0.06
	100	0.23	0.18	0.11	0.09	0.23	0.16	0.08	0.01	0.23	0.12	0.09	0.05
	75	0.21	0.14	0.07	0.07	0.21	0.16	0.09	0.02	0.21	0.11	0.08	0.05
	50	0.17	0.11	0.06	0.06	0.17	0.12	0.05	0.02	0.17	0.08	0.06	0.03
20°	150	0.21	0.17	0.12	-	0.21	0.15	0.06	-	0.21	0.12	-	-
	100	0.19	0.15	0.10	-	0.19	0.14	0.06	0.04	0.19	0.11	-	-
	75	0.15	0.10	0.08	-	0.15	0.12	0.05	0.06	0.15	0.09	-	-
	50	0.12	0.09	0.06	-	0.12	0.09	0.04	0.06	0.12	0.08	-	-

become larger than those of the molecular ones in turbulent flow. In computational fluid dynamics, the treatment of the turbulent viscosity and thermal conductivity is indicated to be calculated from the degree of the turbulence, for example in k-ε model. But this calculation method was developed for sufficiently developed turbulence, and no method has been proposed for a flow in a low Reynolds number.

One of the authors proposed a calculation method, where a cavity was divided into twenty sections along the flow to know the air temperature distribution in the cavity and to calculate the buoyant force precisely[11]. The authors intended to find the viscosity and heat conductivity, which are applicable for the above calculation method, from the present experiment.

5. Conclusions

The present results of the laboratory experiment showed that free inlets and outlets were highly recommended in order to make the cavity ventilation effective. However, even when restrictions in the openings cannot be avoided, the benefits of cavity ventilation were still able to achieve if the restrictions were not so tight.

In the experiment with a heat production of 150 W/m^2, both openings fully opened, and a tilting angle of 30 degrees, the highest air velocity reached to 0.36 m/s. The average velocity was 0.25 m/s, and the flow rate was 7.7 L/s. This corresponded to ventilation of 3.0 times per minute in the cavity. This ventilation rate was reduced to 1.6 times per minute when both openings were restricted equally to a 35 mm vent size.

When the tilting angle was decreased to 20°, with a heat production was still kept at 150 W/m^2, and both openings fully opened, the flow rate decreased to 6.6 L/s, which corresponded to a ventilation of 2.6 times per minute.

The average temperature rise of the cavity air was 10.6°C when the cavity was tilted at 30 degrees with heat production of 150 W/m^2 and the openings were kept fully open. This temperature rise increased by 6.9°C when the end openings were restricted equally to 35 mm. This inevitably affected the increment of temperature at the lower surface. Average temperature at the lower surface arose from 2.9°C of a fully open condition to 5.5°C with both opening restriction of 35 mm.

The roof cavity ventilation appeared to be effective in evacuating irradiated solar heat before transfer to the lower roof structure and further down into the space below. This suggests that natural ventilation in the roof cavity can be effectively applied to solar incidence evacuation.

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Table 4 Flow rates and Reynolds number of airflow in Case C at two inclination angles

Inclination	Heat Production	Slit size		Average Vel.	Average Temp.	Flow Rates	Reynolds Number
		Inlet	Outlet				
	W/m^2	mm	mm	m/s	°C	L/s	
30°	150	78	78	0.25	10.6	7.7	2009
		35	35	0.13	17.5	4.0	1050
		20	20	0.11	22.3	3.4	889
		10	10	0.06	30.0	2.0	511
	100	78	78	0.23	7.1	7.1	1860
		35	35	0.12	12.7	3.9	1010
		20	20	0.09	17.5	2.8	737
		10	10	0.05	22.7	1.7	436
	75	78	78	0.21	6.7	6.6	1718
		35	35	0.11	10.5	3.4	883
		20	20	0.08	14.1	2.3	612
		10	10	0.05	17.8	1.7	440
	50	78	78	0.17	5.2	5.2	1354
		35	35	0.08	8.0	2.6	677
		20	20	0.06	10.5	1.8	473
		10	10	0.03	12.9	1.0	253
20°	150	78	78	0.21	12.2	6.6	1738
		35	35	0.12	18.5	3.9	1014
		20	20	-	-	-	-
		10	10	-	-	-	-
	100	78	78	0.19	9.5	5.9	1551
		35	35	0.11	13.5	3.4	887
		20	20	-	-	-	-
		10	10	-	-	-	-
	75	78	78	0.17	7.8	5.4	1423
		35	35	0.09	10.6	2.7	709
		20	20	-	-	-	-
		10	10	-	-	-	-
	50	78	78	0.17	6.1	5.2	1372
		35	35	0.08	7.8	2.4	620
		20	20	-	-	-	-
		10	10	-	-	-	-

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Table 5 Heat balance in cavity of Case C

Inclination	Heat Production	Configuration	Opening		Total heat production	Ventilation	Top side loss	Bottom side loss	Deviation	Portion of ventilation	Portion of loss through top side	Portion of loss through botom side	Portion of deviation
			Inlet	Outlet									
(°)	W/m²		mm	mm	W	W	W	W	W	%	%	%	%
30	150	Case C	78	78	275	103.0	155.9	8.6	7.5	19.3	29.2	1.6	3
			35	35		91.7	161.0	16.1	6.2	36.3	63.7	6.4	2
			20	20		96.0	162.0	23.5	-6.5	37.2	62.8	9.1	-2
			10	10		76.7	170.1	35.4	-7.2	31.1	68.9	14.3	-3
	100	Case C	78	78	184	64.7	116.9	5.5	-3.1	17.7	32.0	1.5	-2
			35	35		65.5	121.8	10.8	-14.0	35.0	65.0	5.8	-8
			20	20		64.4	129.1	18.0	-27.4	33.3	66.7	9.3	-15
			10	10		50.6	133.7	26.5	-26.8	27.4	72.6	14.4	-15
	75	Case C	78	78	138	57.6	101.9	7.1	-28.6	19.3	34.3	2.4	-21
			35	35		48.6	103.4	8.4	-22.4	32.0	68.0	5.5	-16
			20	20		44.5	105.7	13.4	-25.7	29.6	70.4	8.9	-19
			10	10		40.2	108.0	19.6	-29.8	27.1	72.9	13.2	-22
	50	Case C	78	78	92	37.0	78.3	3.2	-26.6	17.9	37.8	1.6	-29
			35	35		30.7	84.2	6.1	-28.9	26.7	73.3	5.3	-31
			20	20		26.2	83.3	10.2	-27.7	23.9	76.1	9.3	-30
			10	10		18.2	84.1	12.5	-22.7	17.8	82.2	12.2	-25
20	150	Case C	78	78	275	100.7	167.5	7.6	-0.8	36.6	60.4	3.1	-1
			35	35		93.4	175.0	17.4	-10.8	32.7	61.0	6.3	-12
			20	20		-	-	-	-	-	-	-	-
			10	10		-	-	-	-	-	-	-	-
	100	Case C	78	78	184	74.2	86.7	6.7	16.4	44.2	51.5	4.3	18
			35	35		62.7	130.4	12	-21.1	30.5	63.4	6.1	-23
			20	20		-	-	-	-	-	-	-	-
			10	10		-	-	-	-	-	-	-	-
	75	Case C	78	78	138	53.3	106	4.9	-26.2	32.6	64.2	3.3	-29
			35	35		43.1	109	7.9	-22.4	26.7	68.0	5.3	-24
			20	20		-	-	-	-	-	-	-	-
			10	10		-	-	-	-	-	-	-	-
	50	Case C	78	78	92	35.2	85.8	4.3	-33.3	28.1	68.4	3.6	-36
			35	35		28.7	84.9	5.3	-26.9	24.2	71.0	4.8	-29
			20	20		-	-	-	-	-	-	-	-
			10	10		-	-	-	-	-	-	-	-

A Laboratory Experiment on Natural Ventilation of a Roof Cavity

屋根内中空層の自然換気に関する実験室実験結果

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キーワード:屋根裏空間、自然換気、開口部のスリット、開口形状

本研究は，屋根内中空層の自然換気が屋根を通して伝達される日射熱を軽減する効果を調べるために，屋根モデルを用いた実験室実験の結果をまとめたものである。実験は，日射時に屋根内中空層へ伝達される熱を中空層上面から発生するものとした。中空層の流入および流出口は、屋根中空層に存在する障害物を模擬するため、78, 35, 20, 10mmとスリット状に狭め、その障害の位置の組み合わせによって 3 群に分けた。傾斜角 20° および 30° の 2 種と単位面積当たり発生熱量 50, 75, 100, 150 W/m²の 4 段階について測定を行い，中空層上下面温度と、中空層内気流速および温度分布を測定し、気流による熱輸送量を推定した。その結果，熱排除能力が開口寸法に強く影響されることを示した。実験中の平均流速の最高値は 0.25m/s に達した。この値は換気量に換算して 7.7L/s となり、毎分 3.0 回の換気回数に相当する。また，実験中の最大レイノルズ数は約 2009 であった。

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