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# EXPERIMENTAL STUDY OF NATURAL CONVECTION HEAT TRANSFER THROUGH AN APERTURE IN PASSIVE SOLAR HEATED BUILDINGS*

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## ABSTRACT

The objective of this study is to obtain correlations between natural convection heat transfer through an aperture and temperature difference between the two rooms. A one-fifth similitude model of a two-room building is used. The model is filled with Freon[3] gas to satisfy similarity of the experiment to full-scale conditions in air. The experimental apparatus and experimental techniques are explained. Experimental results are presented in terms of Grashof, Nusselt, and Prandtl numbers.

The effects of the height, the width, and the vertical position of the apertures are investigated, as is the effect of the room volume.

## 1. INTRODUCTION

In passive solar heated buildings, especially in multi-room buildings, it is desirable to distribute heat that is gained in south-side rooms to north-side rooms by natural means insofar as possible. Coupling rooms by natural convection is one of the simplest and most effective and practical ways to distribute this heat. To design energy efficient and comfortable passive solar buildings, it is important to understand the mechanisms of natural convection. Because natural convection through an aperture is affected by its geometry, it is necessary to investigate the correlation between heat transfer and the geometry of many different apertures.

Convective heat transfer through an aperture was studied by Brown and Solvasoni[1] in 1962 using a real scale model with air as a fluid. They derived a theoretical correlation between the Grashof, Nusselt, and Prandtl numbers. They did the experiments for several rectangular openings in the vertical partition over the range of the Grashof number from $10^6$-$10^8$ based on opening height as the characteristic length. Their experimental results agreed fairly well with their theoretical correlation.

Nansteel and Greif[2] studied the natural convection in enclosures fitted with a vertical adiabatic partition. The experiments were carried out with water for Rayleigh numbers over the range $10^{10}$-$10^{11}$ based on the distance between the hot plate and the cold plate as the characteristic length. They used the cross-cavity heat transfer averaged over the whole partition for the calculation of the Nusselt numbers. Their results show relatively little dependence of the rate of heat transfer on different widths of openings. It was also shown that the heat transfer decreased with decreasing the aperture height. However, it seems that the heat transfer per area of the openings increases with decreasing the aperture height ratio.

Weber[3] studied the subject experimentally at Los Alamos using a one-fifth similitude model filled with Freon 12. The experiments were carried out for different heights and widths of the doorways keeping the area of the doorways constant to eliminate the effects of area changes. Assuming that the effect of the width is small enough to ignore, the effect of the height was obtained. He also studied the effect of the width in other experiments and showed that the effect is very small compared with the height of the doorway.

The objective of this study is to obtain correlations of heat transfer by natural convection under steady state conditions using an improved apparatus and for not only doorways, but also for other kinds of apertures, for example, a window between rooms. To reduce costs, the experiments are done using a one-fifth similitude model filled with Freon 12.

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## 2. THEORY

The following correlation was derived by Brown and Solvason, assuming a zero-viscosity incompressible fluid and steady-state flow:

$$\frac{Nu}{Pr} = \frac{C}{3} Gr^{0.5} \quad ,$$

where Nu is the Nusselt number defined as

$$Nu = \frac{Q_{cv}}{A\Delta T} \cdot \frac{H}{k} \quad ,$$

where $Q_{cv}$ is the natural convection heat flow rate through an aperture (W), A is the area of the aperture ($m^2$), $\Delta T$ is the temperature difference between two rooms (K), H is the height of the aperture (m), and k is the thermal conductivity of the fluid (W/mK).

Gr is the Grashof number defined as

$$Gr = \frac{g H^3 \Delta T}{\nu^2 \bar{T}} \quad ,$$

where g is the gravity acceleration ($m/s^2$), $\bar{T}$ is the average temperature of the fluid (K), and $\nu$ is the kinematic viscosity ($m^2/s$).

Pr is the Prandtl number defined as

$$Pr = C_p \mu / k \quad ,$$

where $C_p$ is the specific heat (J/kg K), $\mu$ is the absolute viscosity (N $s/m^2$), and C is the discharge coefficient of the aperture.[4]

To satisfy the similarity of the model, Gr and Pr must be equal to the values for a full-scale building.

## 3. APPARATUS

A similitude model was constructed as illustrated in Fig. 1. The model is divided into two rooms by a partition made of Styrofoam® covered with aluminum foil. In the south room an electric panel heater is installed, which consists of an aluminum plate and electric heating tapes. In the north room a panel cooler is installed, which consists of a copper plate and a copper tubing coil attached to the plate.

To make the model as thermally insulative as possible, 6-in.-thick polyurethane insulation covered with plywood and Formica® was used initially. The thermal conductances of the wall, the ceiling, and the floor are 0.162, 0.163, and 0.083 $W/m^2K$, respectively. Those values were obtained experimentally for this model.

As it is necessary to keep the ambient temperature around the model constant, the model was installed in a chamber consisting of a wood frame and plastic sheets. A heater consisting of 14 100-W electric bulbs is installed in the chamber and controlled by a thermostat. Two small fans are used to maintain a uniform temperature distribution in the chamber.

To measure temperatures inside and outside the model, 150 thermocouple wires are used. Water temperature at the inlet and outlet of the cooling panel is measured by a platinum resistance thermometer. Water flow rate of the cooler is measured by a mass flow meter. The data acquisition system consists of an HP-9830 desktop computer, an HP-9497 scanner, an HP-9866 printer, and an HP-9862 plotter.

As the model is constructed with plywood, Formica, and polyurethane, it is difficult to contain the Freon gas. To stop the leakage, modeling clay was inserted into every joint. Furthermore, the inside surface of the box was lined with Mylar® sheet and painted with epoxy paint.

When filling the model with Freon gas, the gas never stratifies under the air, even though the density of the gas is about 4 times that

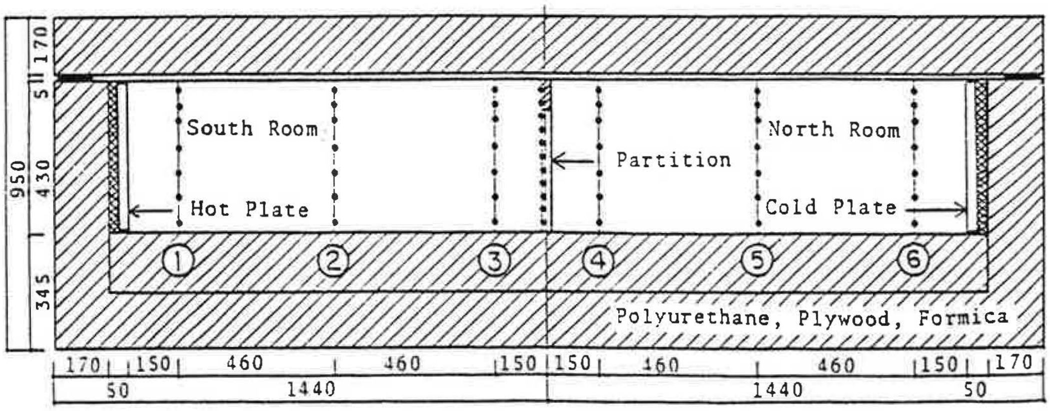


Fig. 1. Similitude model of a two-room passive solar heated building.

of air. This may be caused by the high mass diffusivity of Freon gas within air. To raise the concentration of Freon 12 in the model high enough, the gas was added continuously until the concentration reached 95%.

As it is inevitable for Freon gas to be mixed with air, gas properties must be corrected according to the concentration of the gas. Oxygen concentration is measured for each experiment and corrections are made accordingly. For example, when the concentration of Freon is 90% by volume, the correct values of Gr, Nu/Pr, and C are 0.82, 0.96, and 1.06, respectively, times the values calculated using properties of pure Freon gas. In almost all experiments, the concentration of Freon was kept between 90 and 95%.

## 4. EXPERIMENTS

The experiments were carried out for nine different geometries of apertures designated 1 through 9, as illustrated in Fig. 2.

The Geometries 1, 2, and 3 are chosen to study the effect of the height or the width of the doorways on the convective heat transfer. Geometries 4, 5, 6, and 7 are chosen to study the effect of the vertical position of the apertures; 6 has two identical apertures separated vertically. Geometries 8 and 9 are chosen to study the effect of the room volume.

For each geometry of the aperture, measurements were performed for three or five different Grashof numbers by changing the electric power to the heating plate over a range from 18W to 220W.

Gas temperatures in the model are measured at six points horizontally and seven points vertically, as illustrated in Fig. 1.

As the model has significant mass and a very low loss coefficient, it takes several days to reach a steady state condition. To speed up the process, the electric power to the heater was controlled using a special profile. For example, when it is desired to change the power from 18W to 34W, the power is set at 115W for 1 hour and then set at 34W. By this control, it was found that only 12 hours are needed to achieve reasonably steady conditions.

The heat flow rate by natural convection through the aperture, $Q_{cv}$, is calculated by the following equation:

$$Q_{cv} = Q_h - Q_{LS} - Q_p \quad ,$$

where $Q_h$ (W) is the electric power to the heating plate, and $Q_{LS}$ (W) is the heat loss of the south room through the exterior walls, the ceiling, and the floor. This heat loss is calculated by the heat conductance of each part obtained experimentally and the observed temperatures. $Q_p$ (W) is the heat flow by conduction through the partition.

## 5. RESULTS

Figure 3 shows an example of the vertical temperature profiles for Geometry 1 at seven different points, including the doorway. Temperatures are normalized by the temperature difference between the hot plate and the cold plate. This shows that the farther the point is from the aperture, the less the temperature is influenced by the flow at the aperture.

Figure 4 shows the effect of the definition of the average temperature of the room on the results for Geometry 1. The results are affected very much by the definition of the average temperature.

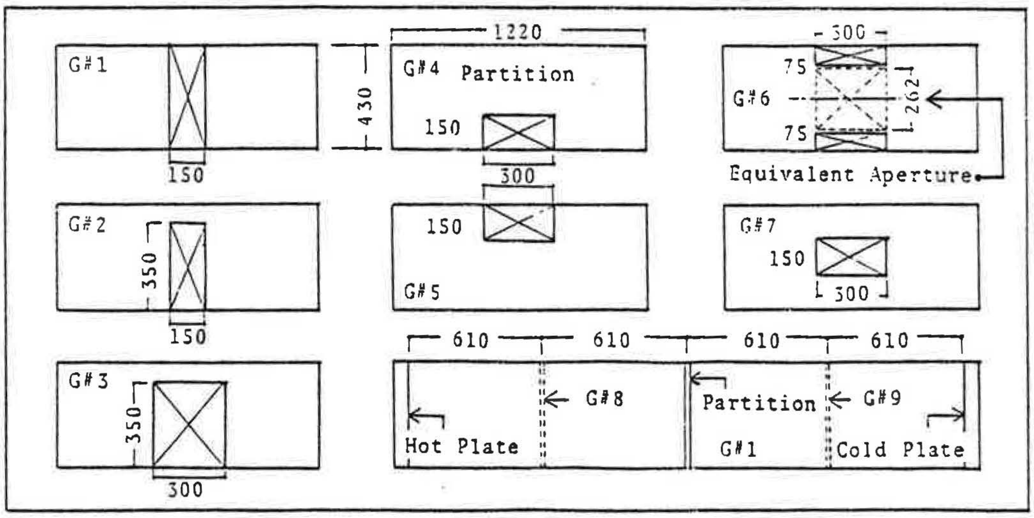


Fig. 2. Geometries of apertures used in this experiment.

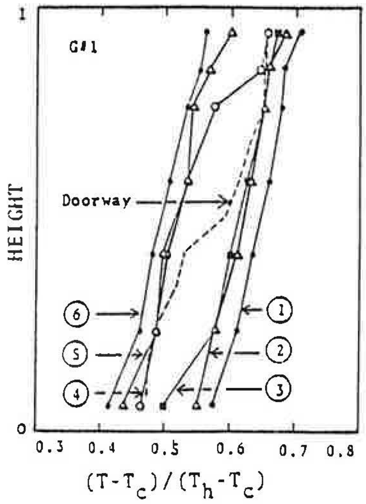


Fig. 3. Vertical temperature profile.

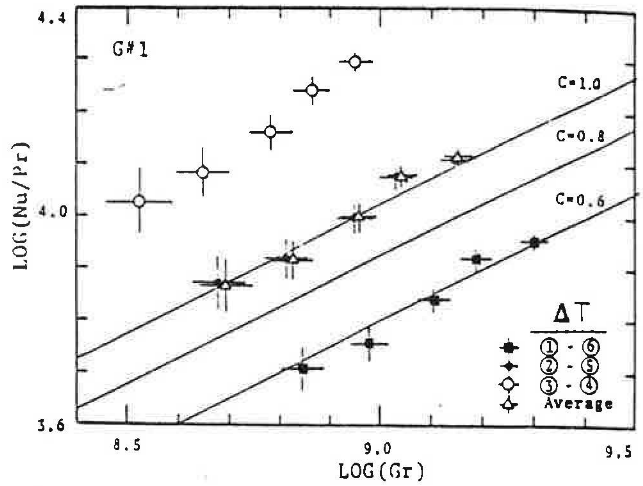


Fig. 4. Correlations between Gr and Nu/Pr for the different definitions of temperature difference.

In this paper, the average temperature at the points ① and ⑥ are used as the temperature of the south room and the north room, respectively. Figure 5 shows the results for the Geometries 1 through 7. The abscissa is log (Gr) and the ordinate is log (Nu/Pr). Theoretical correlations for discharge coefficients of 1.0, 0.8, and 0.6 are also shown. As the characteristic length, the aperture height is used except for Geometry 6, which has two identical apertures separated vertically. For Geometry 6, the equivalent single aperture shown in Fig. 2 was considered, and the height and the area of the single aperture were used for the calculation of Gr and Nu/Pr. The equivalent single aperture is that which gives the same flow rate as the original aperture assuming that the velocity profile is proportional to the square root of the distance from the neutral point. In Fig. 6, every condition is the same as in Fig. 5 except for the definition of the average temperature of each room. In this figure, temperatures above and/or below the height of the apertures were eliminated for the calculation of the average temperatures. The results in Fig. 6 show less scatter than in Fig. 5, especially for Geometries 4, 5, 6, and 7, in which the aperture height is small. This means that the temperatures in the range of the height of the aperture have the most significant effect on the convective heat flow through the aperture.

In Fig. 6 the results for Geometries 1, 2, and 3 are close to each other, and it seems that the effect of the height and the width of the apertures is relatively small.

The exponents of Gr for Geometries 2 and 3 are slightly greater than that of Geometry 1. The discharge coefficients are 0.60, 0.60, and 0.48 for Geometries 1, 2, and 3, respectively. These results are not consistent with those obtained by Weber. This difference may be caused by the difference of the geometry of the enclosure itself, the geometry of the apertures, and the definition of the temperature difference between the two rooms. Weber fixed the area of the apertures to eliminate the effect of the area. In this experiment, the area of the apertures is changed with the change of the height or the width of the apertures.

In Fig. 6, the results for Geometries 4, 5, 6, and 7 are close to each other. Though the results for 6 have higher Gr, they appear consistent with other results.

Though the groups of Geometries 1 through 3 and 4 through 7 are consistent within each group, the latter seems to have higher discharge coefficients than the former. This is also inconsistent with Weber's results. This difference has not yet been investigated.

Figure 7 shows the results for Geometries 1, 8, and 9 to investigate the effect of the volume of the rooms. The aperture is the same for these cases. The partition was moved to change the ratio of the volume of the room from 1:1 to 1:3 and to 3:1. As before, temperatures near the hot plate and the cold plate are used for the room temperatures. It is clear that there is almost no effect of the volume of the rooms on the correlation of the heat transfer.

Figure 8 shows isotherms for Geometries 1, and 2. Isotherms are helpful to visualize the flow patterns in the model. The least square correlations of Gr and Nu/Pr and the discharge coefficient of the group of

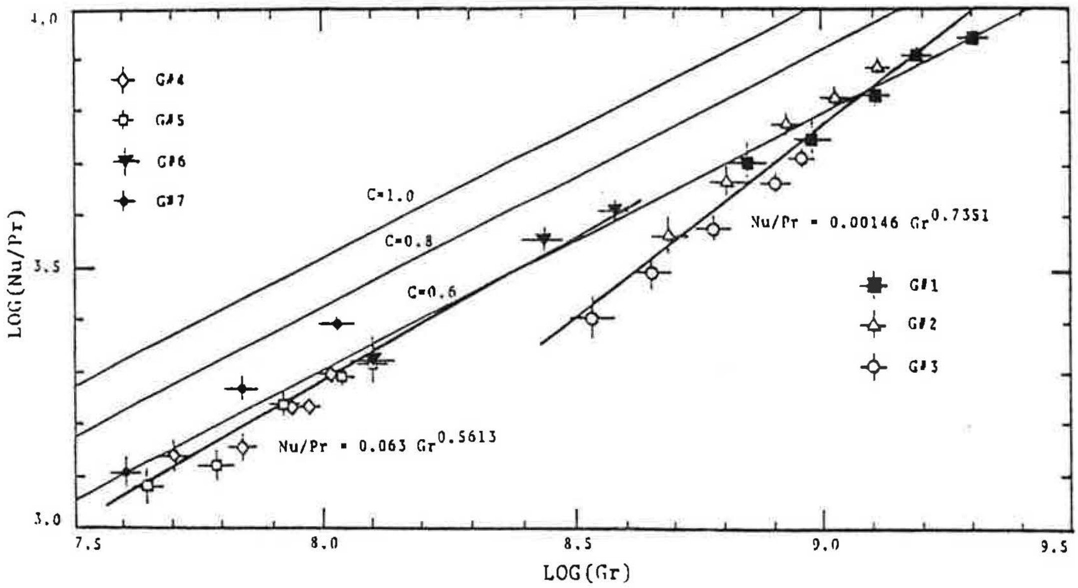


Fig. 5. Correlations between Gr and Nu/Pr for Geometries 1-7 when the temperatures near the plates are averaged for all the heights in the model.

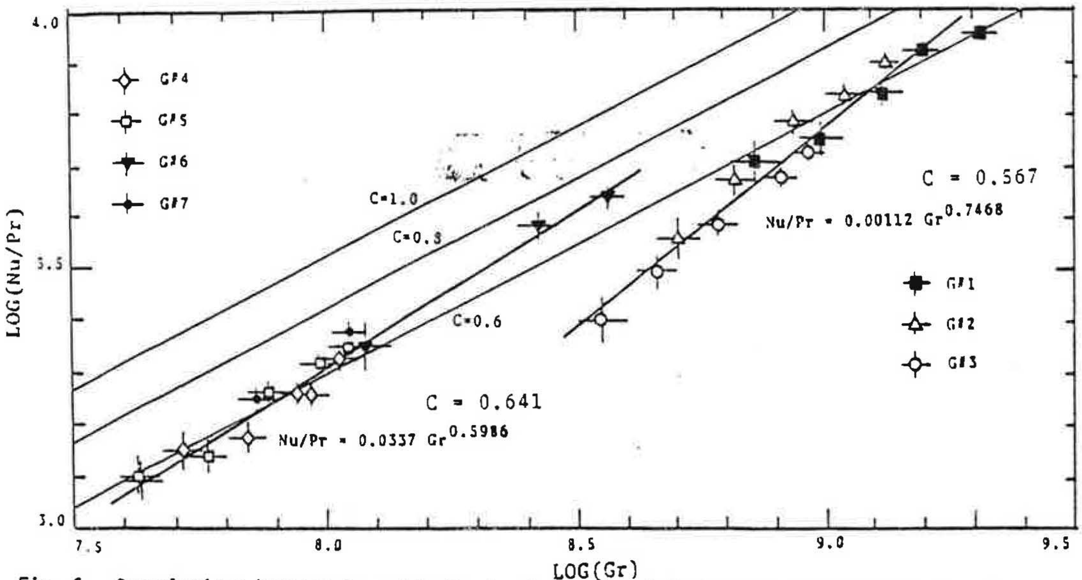


Fig. 6. Correlations between Gr and Nu/Pr for Geometries 1-7 when the temperatures near the plates are averaged for the height within the apertures.

Geometries 1, 2, and 3 or 4, 5, 6, and 7 are shown in Fig. 6. The resultant simple equation of heat transfer is

$$Q_{CV} = 3.21\ W(H\Delta T)^{3/2}$$

for Geometries 1, 2, and 3, and

$$Q_{CV} = 3.63\ W(H\Delta T)^{3/2}$$

for Geometries 4, 5, 6, and 7, where $Q_{CV}$ (Btu/h) is the rate of heat flow by convection, W (ft) is the aperture width, H (ft) is the aperture height, and $\Delta T$ (°F) is the temperature difference between the two rooms.

## 6. CONCLUSION

Results are generally consistent with the

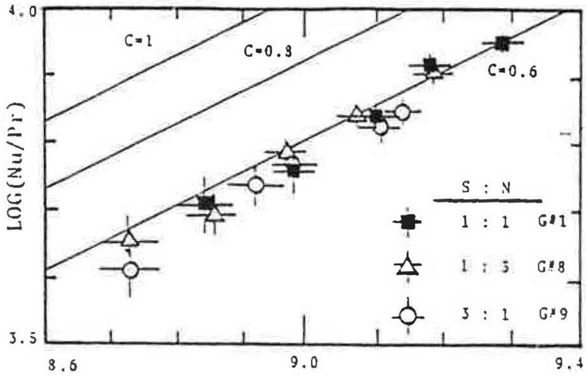


Fig. 7. Correlations between Gr and Nu/Pr for Geometries 1, 8, and 9.

previous works done by Brown and Solvason, Weber, and Grief and Nansteel except that the aperture height ratio and the aperture width ratio seem to have different effects on heat transfer from Weber's results. This inconsistency may be caused by the difference of the geometry of the model or the definition of the room temperature.

For the geometry that has two identical apertures separated vertically, the equivalent single aperture gives consistent results with other geometries. In the case tested in this experiment, dividing an aperture into two apertures at the top and the bottom resulted in 1.75 times the heat transfer for the same temperature difference and the same total aperture area.

The effect of the room volume on the correlation of the heat transfer is very small if the temperatures away from the aperture are used as the room temperatures. This experimental study will be continued in the Solar Energy Section of the Los Alamos National Laboratory.

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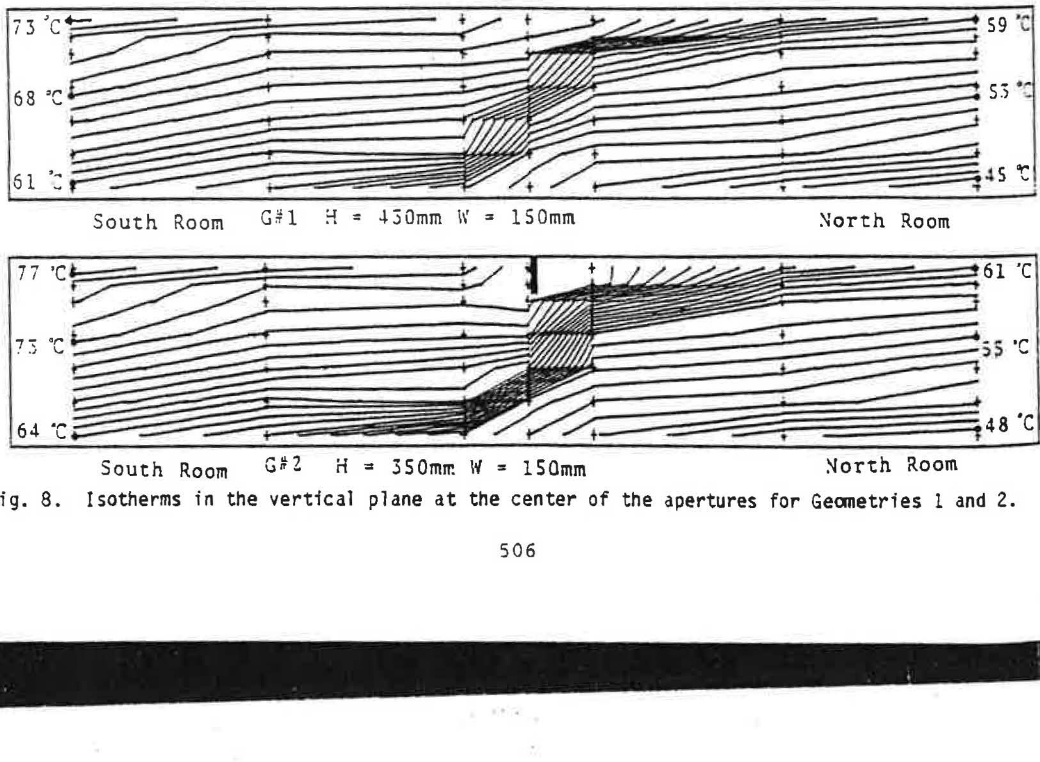


Fig. 8. Isotherms in the vertical plane at the center of the apertures for Geometries 1 and 2.