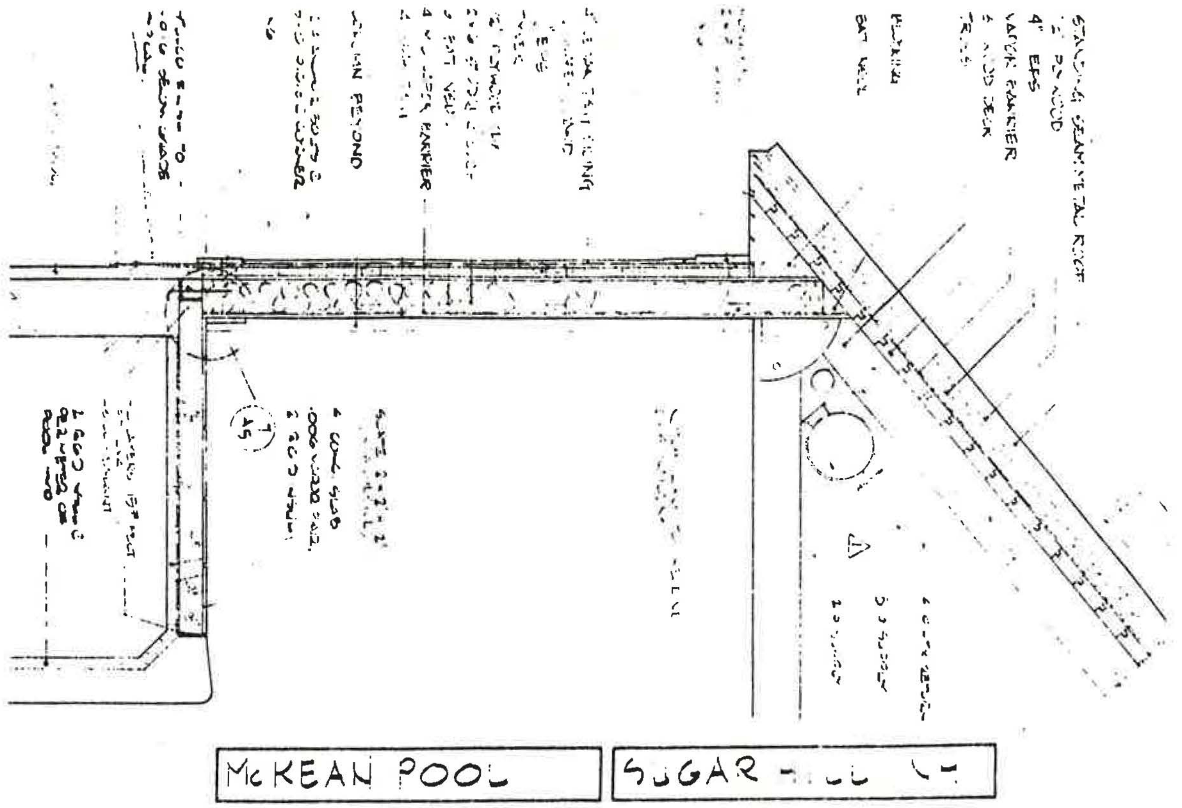
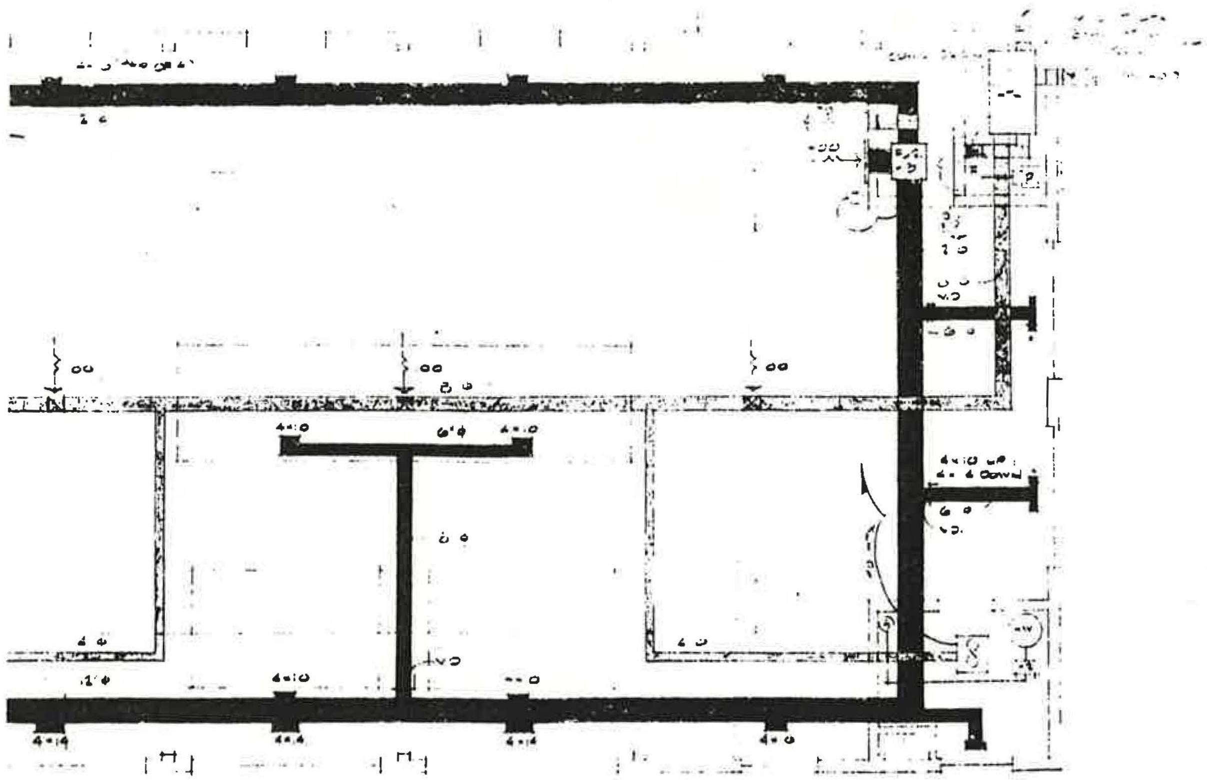


Fig. 7



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THE INFLUENCE OF GEOMETRY ON NATURAL CONVECTION IN BUILDINGS

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ABSTRACT

Strong free convection airflows occur within passive solar buildings resulting from elevated temperatures of surfaces irradiated by solar energy compared with the cooler surfaces not receiving radiation. The geometry of a building has a large influence on the directions and magnitudes of natural airflows, and thus heat transfer between zones. This investigation has utilized a variety of reduced-scale building configurations to study the effects of geometry on natural convection heat transfer. Similarity between the reduced-scale model and a full-scale passive solar building is achieved by having similar geometries and by replacing air with Freon-12 gas as the model's working fluid. Filling the model with Freon-12 gas results in similarity in Prandtl numbers and Rayleigh numbers based on temperature differences in the range from 10^9 to 10^{11}. Results from four geometries are described with an emphasis placed on the effects of heat loss on zone temperature stratification shifts.

1. NOMENCLATURE

C	orifice coefficient, dimensionless
c_p	specific heat at constant pressure, J/kg K
D	zone height, m
g	acceleration of gravity, m/s^2
h	characteristic length, m
H	orifice height, m
k	thermal conductivity w/m·K
Nu	Nusselt number $[q \cdot h/(k \Delta T)]$, dimensionless
Pr	Prandtl number (ν/α), dimensionless
q	energy flux, w/m^2
Q	volumetric flow rate, m^3/s
Ra	Rayleigh number $(g \cdot \beta \cdot h^3 \cdot \Delta T/\nu \cdot \alpha)$, dimensionless
St	Stanton number $q/(\rho \cdot c_p \cdot u_c \cdot \Delta T \cdot H \cdot W)$, dimensionless
T	temperature, K
u	velocity, m/s
W	orifice width, m
x	vertical coordinate, m
α	thermal diffusivity, m^2/s
β	coefficient of thermal expansion, 1/K
Δ	difference
ρ	density, kg/m^3
ν	kinematic viscosity, m^2/s

Subscripts

c	characteristic
m	mixed-mean

Superscripts

-	average

2. INTRODUCTION

Free convection is a primary mechanism for heat transfer between different zones within passive solar buildings. Zones that are neither radiatively nor conductively coupled to a solar collection zone, (such as a sunspace, the space behind a Trombe wall, or a direct gain zone), in fact are solely dependent on natural convection for heat transfer and to insure thermal comfort within that zone. There exists a dichotomy between the importance and comprehension of this natural convection heat transfer mechanism within passive solar buildings. Unlike radiation heat transfer analysis where even complex geometries can be treated with the simple view factor approach, we find ourselves at a loss to predict the effects of even simple geometric changes on the heat transfer due to natural convection. These include such common features as apertures, elevation changes, corridors, and stairways. A review of air velocity profiles and temperature scans from but a dozen passive solar buildings (1) makes evident that geometry has a large influence on the directions and magnitudes of natural air flows, and thus the heat transfer between zones. The second law of thermodynamics predicts the natural distribution of collected solar energy from the collection zone to other cooler zones within the building. The question that is of interest here, then is "How does geometry affect this distribution?"

Previous experimental studies concerned with convection in passive solar buildings have been performed both at full scale and reduced scale. The reduced scale testing provides a high degree of control over the experimental conditions and is capable of producing repeatable results. When one is concerned with testing a number of geometries, the reduced scale experiment offers the advantages of reduced downtime and costs during reconfigurations. A reduced-scale-testing apparatus, with a scaling factor of one-eighth, was utilized for this investigation. No model is able to perfectly simulate its full-scale counterpart; however, useful information can be obtained from a model if the testing is properly conducted. One requirement in modelling is that the governing physical equations be similar in both the full-scale and the reduced-scale.

The fundamental physical processes which occur in natural convection flows are essentially identical to those occuring in other fluid flows. The basic equations, which are applicable to fluid flow, therefore can be used to analyse natural convection flows. These are the differential equations which result from a conservation of mass (continuity), conservation of momentum (Navier-Stokes), and conservation of energy (energy equation) derivations. The physical circumstances of this experiment allow one to express the governing equations in simpler forms. This investigation is not concerned with transient natural convection; thus time dependence can be eliminated from the governing equations. In natural convection, the fluid motion arises from buoyancy; thus density variations that provide motion will be taken into account, otherwise the working fluid will be considered incompressible (the Boussinesq approximation). With the above approximations considered, the governing equations are written and made dimensionless by appropriate combinations of variables (2). Two dimensionless groups result: $(g \cdot \beta \cdot h^3 \cdot \Delta T/\nu \cdot \alpha)$ and (ν/α). These quantities are known as the Rayleigh and Prandtl numbers respectively and are the two similarity parameters for natural convection. The governing equations for a full-scale and reduced-scale experiment will be identical if the Rayleigh and Prandtl numbers are similar between the two scales. For this investigation a scale reduction was achieved by selecting Freon-12, a gas with a greater density than that of air, as the working fluid. The Prandtl numbers for air and Freon-12 gas are nearly identical, which assures Prandtl number similarity.

This work continues the series of similitude experiments utilizing Freon-12 gas to model passive solar buildings, initiated by Weber (3) and subsequently performed by Yamaguchi (4) at Los Alamos. Weber studied simple doorway geometries of various height to width ratios maintaining a constant area. The results, based on zonal-fluid temperatures, suggested weak dependence on doorway width, but strong dependence on doorway height. Yamaguchi's later work was conducted within an enclosure constructed from Formica covered polyurethane insulation. Nine different geometric configurations were chosen to study the effects of doorway heights and widths, the vertical positions of apertures, and room volumes. The results were again based on zonal fluid temperatures.

Brown and Solvason (5) in 1962 pioneered this domain of research by proposing a basic theory for natural convection across an opening. The theory, based on an approach utilizing an inviscid Bernoulli equation, suggests an equation for heat transfer through an orifice of the following form:

$$Nu = C/3 \cdot (Ra \cdot Pr)^{1/2} \quad . \qquad (1)$$

The theory compared well with their data obtained from a full-scale test apparatus with air as the working fluid. An initial assumption taken by Brown and Solvanson in developing the theory was that the core temperatures on either side of the opening were constant, that is non-stratified. More recent investigations, however, have shown the importance of temperature stratification on heat transfer through openings.

Nansteel and Greif (6) have studied natural convection in partioned enclosures within a water filled container. A liquid working fluid allows a greater reduction in the model scale, but at the expense of violating the Prandtl number similarity. The Prandtl number for air has a value of 0.71 but varies between 3 and 10 for water, depending on the temperature. Bohn and Anderson (7) have investigated the sensitivity of natural convection heat transfer within unpartitioned enclosures to the Prandtl number of the working fluid. This work was specifically performed to determine whether continued testing of water filled reduced-scale enclosures was justified for building heat transfer research. They reported a reduction in the average Nusselt number (heat transfer) of 11% when the same enclosure experiment was conducted with air as compared with water, over a Rayleigh number range from 2×10^7 to 1.2×10^8. In a related numerical problem (8) a 20% variation in Nusselt number, as the Prandtl number was varied from 0.053 to infinity, was noted. One might expect a deviation of heat transfer results for different working fluids for partitioned enclosures, as flow obstructions increase the importance of the fluid's inertial properties (9).

3. EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental apparatus (Fig. 1) consists of a rectangular enclosure constructed of 1.0 mm (0.039 in) stainless steel with the dimensions 86.4 cm x 111.8 cm x 233.7 cm (34 in x 44 in x 92 in). The enclosure has a removable top plate allowing access into the test space; a Freon-12 seal was assured by utilizing an isoprene rubber gasket between the enclosure lip and top plate. To reduce the energy losses to the environment, rigid insulation surrounded all six sides of the enclosure. The natural convection flows were driven by two opposing vertical plates, one being resistively heated, the other liquid cooled. The heated surface was constructed of two aluminum plates, 0.3175 cm (0.125 in.) thick, sandwiching ten horizontal resistance heaters. A variety of thermal boundary conditions were achieved on the heated surface, from isothermal to constant heat flux, by having a separate control for each resistance heater. The ten resistance heaters were powered by a 40 volt direct current power supply making the total power available to the heated surface near 250 watts (853.5 Btuh). A direct-current power source was chosen to avoid generating "noise" in the thermocouple circuitry. While the aspect ratio was fixed at 1/2 during this series of experiments the ratio can change by adjusting the heated surface location.

The cooled surface had a sandwich construction similar to the heated surface but with slightly thicker aluminum plates 0.4775 cm (0.188 in.). The two aluminum plates sandwiched ten individually valved 9.53 mm (3/8 in.) o.d. copper tubes. Water from a refrigerated bath was circulated through the copper tubes to establish the desired cooling surface boundary condition. The cooling capacity of the refrigerated bath exceeded the power input capabilities of the heating surface. The cooled surface was designed to be physically fixed within the enclosure.

A total of 140 individually calibrated copper/constantan, 40 gauge, teflon coated thermocouples were utilized throughout the apparatus for temperature measurements. One hundred and eleven thermocouples were dedicated to measurements within the enclosure while the remaining 39 were dedicated to measuring the heat loss from the enclosure. Enclosure heat loss was calculated from differential temperature measurements across a finite width of insulation within the rigid insulation surrounding the enclosure. Of the 111 enclosure thermocouples 25 were utilized to monitor the temperature profiles on the heated and cooled surfaces. The remaining thermocouples measured gas temperatures or enclosure-surface temperatures at selected locations. In addition to the thermocouples, a differential temperature measuring circuit was constructed based on two platinum resistance temperature devices to measure the water temperature difference between the two legs of the cooling plate loop. Cooling water mass flow rate was directly measured with a vibrating U-shaped tube flow meter. A positive seal on the thermocouple wires entering the enclosure was assured by passing the wires through oil filled p-traps.

Four different partition configurations were studied during this series of experiments. The four configurations are shown in Fig. 2 below. The vertical partitions consisted of 10.16 cm (4 in.) polystyrene insulation covered with aluminum tape. These partitions could be considered adiabatic as the conduction through the partition was negligible in comparison with the energy transfer through the opening. The elevated sections of configurations 3 and 4 were constructed in a similar manner with polystyrene insulation and aluminum tape. The aluminum tape was installed to reduce the emissivity of the enclosure surfaces. The configurations 1 and 2 were used to establish a data base for comparison with previously reported experimental results. Configurations 3 and 4 investigate the effect of elevation changes. Configuration 3 models a sunspace situated lower than a cooler zone connected by a simple doorway. Configuration 4 is the inverse arrangement modelling a sunspace situated higher than a northern zone.

The following procedure was maintained in this series of experiments. Once the enclosure was configured it was charged with Freon-12 until a concentration greater than 97.5% was reached. The container was then pressurized to 0.004 Pa. (1 in. water) and adjustments were made in the heater circuits, cooling water flow rate and cooling water temperature. The establishment of steady state conditions was determined by a temperature history plot for selected thermocouples throughout the enclosure and its surrounding insulation. Typically 60 hours were required to reach equilibrium after a boundary condition change. A complete scan of all instruments

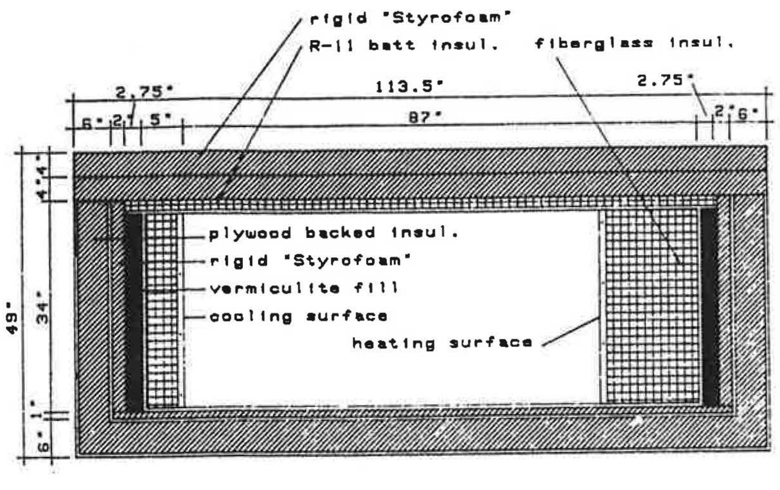

Fig. 1. Longitudinal section of apparatus.

was taken and the measured values recorded once steady state conditions had been reached. The recorded data were then averaged over a ten minute sampling period. Another data scan was performed for the same steady state conditions 24 hours later. Generally the difference between the recorded data for the two scans was less than a few percent; the greatest variation occuring in the cooling water power absorbed. When two data scans had been completed for one state, adjustments were made to the heater circuits and cooling loop to establish a new equilibrium state. This procedure was repeated 12 times for each configuration, approximately 8 states having constant-heat-flux boundary conditions and 4 states having isothermal boundary conditions. For this experimental series, the cooling surface remained in a constant flux mode. The cooling tubes or the resistance heaters were inoperative below the elevated floor level in configurations 3 or 4 respectively.

4. TEST RESULTS

Typically results for enclosure heat transfer experiments report a relationship between the Nusselt and Rayleigh or Grashof numbers in graphical or numerical form. These nondimensional numbers require a characteristic temperature difference, length, and energy flux quantities. Past investigators have utilized either average zone-to-zone fluid temperature differences or average surface-to-surface temperature differences as a characteristic temperature difference. The energy flux has been that which is transferred from the heating surface or that which passes through the

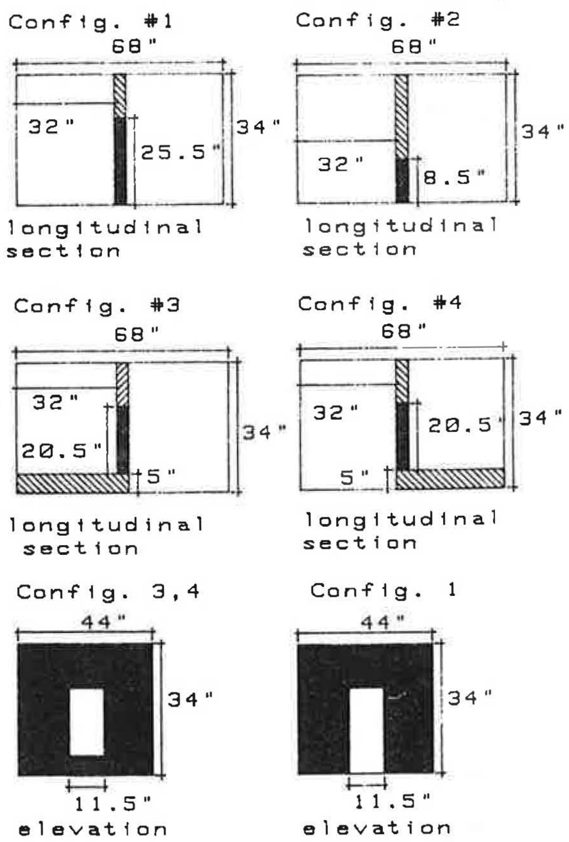

Fig. 2. The configurations.

opening. Conventional characteristic lengths have included the enclosure height, the enclosure length and/or the aperture height. This investigation has been partially concerned with understanding the effects of zonal-heat loss to the ambient on the core region temperature stratification. Thus a portion of the experiments were characterized by having a substantial heat loss through the enclosure surfaces other than the cooling and heating surfaces, and for these cases, characteristic quantities differ from those conventionally chosen.

Evidence for the argument that the core temperature profiles are influenced by the heat losses to ambient is given in Fig. 3. Two cool zone vertical temperature profiles are shown, where the different profiles occur at various horizontal positions in the cool zone. The high loss case was a situation where 63.6% of the energy transfer from the gas side of the heated plate was lost to the environment out the enclosure surfaces. In the low loss case this energy loss was 27.6%. For an adiabatic enclosure

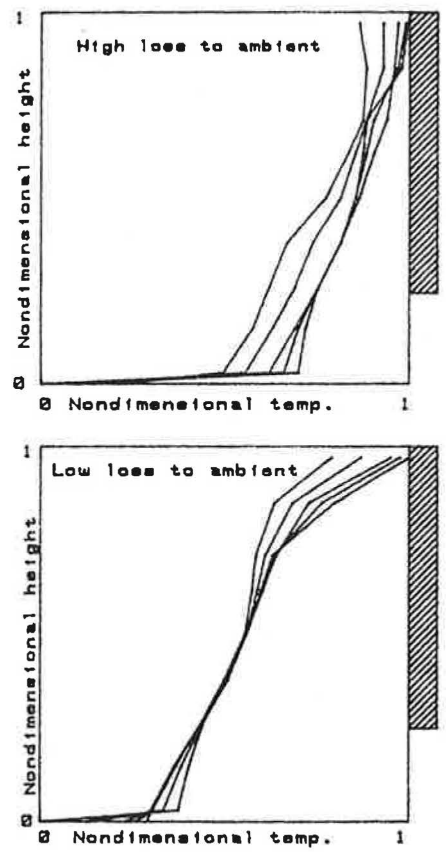

Fig. 3. Cool zone temperature profiles at various horizontal positions.

the vertical temperature profile in the zone will remanin horizontally constant, except near the cooling surface and the aperture. A progression of deviations in the temperature profiles for increasing heat loss to the ambient can be seen in Fig. 3.

The theory of Brown and Solvason for convection through rectangular openings in partitions suggest that the velocity distribution through an aperture should be

$$u = (2\cdot g\cdot\Delta T\cdot z\cdot\beta)^{1/2} \quad , \tag{2}$$

dependent on the difference in fluid density or equivalently fluid temperatures across the aperture where z is the height above the neutral plane (the plane of no horizontal flow). This expression can be integrated in the vertical direction to obtain the volumetric flow through the aperture;

$$Q = C/3\cdot W\cdot(g\cdot\Delta T\cdot H^3\cdot\beta)^{1/2} \quad . \tag{3}$$

The heat transfer rate across the aperture associated with this flow is then,

$$q = Q\cdot\rho\cdot c_p\cdot\Delta\overline{T}_m \quad , \tag{4}$$

where $\Delta\overline{T}_m$ is the mixed-mean temperature of the counter flowing streams.

Instead of selecting the Nusselt number as the non-dimensional heat transfer rate through the aperture we will employ the Stanton number defined as;

$$St = q/\rho\cdot c_p\cdot u_c\cdot\Delta T\cdot H\cdot W \quad . \tag{5}$$

This suggests different definitions for the characteristic values of temperature difference, length, and energy flux for flows through apertures. The Brown and Solvason theory is based on constant zone temperatures. The experiments are characterized by stratified zone temperatures; thus the characteristic temperature difference will differ slightly. In this case, the characteristic temperature difference is defined as the gas temperature difference between the two zones at the mid-height of the aperture. The characteristic length is defined as the orifice height and the energy flux is based on the flux at orifice. The Stanton number requires a characteristic velocity. A logical choice for the characteristic velocity comes from the Brown and Solvason theory above with z being replaced by one-half the orifice height. If this velocity is chosen, the equations (4) and (5) can be rearranged to show:

$$St = C/3\cdot\Delta\overline{T}_m\,/\Delta T_{1,2} \quad . \tag{6}$$

If the Brown and Solvason theory is rederived for a linearly stratified core with the condition that mass flow rate is independent of the stratification (1) we may write,

$$\Delta T_m/\Delta T_{1,2} = 1 + \frac{0.3\cdot H\,(b_1 + b_2)}{(a_1 - a_2)} \quad . \qquad (7)$$

where H is the aperture height and the a's and b's are the constant and linear coefficients, respectively, of the temperature profiles for zones 1 and 2. The coefficient of discharge C has a range of values from 0.6 to 1.0 for forced flow, depending on the geometry of the orifice, and a broader range for buoyancy driven flows. For isothermal zones $\Delta T_m/\Delta T_{1,2} = 1$ and thus from Equation (6) we see that the Stanton number is solely dependent on the geometry of the orifice. Stanton number results from the four experimental configurations are shown graphically in Fig. 4, where the abscissa is the temperature difference ratio $\Delta T_m/\Delta T_{1,2}$. Each data point represented by the configuration number results from a single set of heating and cooling boundary conditions. The lines shown in Fig. 4 are lines of constant discharge coefficient, and are showing the change in Stanton number for fixed temperature difference ratios. The data for configurations 1 and 3 are in good agreement with the line for a discharge coefficient of 0.6, which would be expected for these sharp edged apertures. The data for configurations 2 and 4 fall just beyond the envelope of discharge coefficients between 0.4 and 0.8. The theory of Brown and Solvason and the modification made for linear temperature profiles is based on the inviscid Bernoulli equation, and therefore does not account for viscous effects. The smaller aperture height of configuration 2 is associated with higher gas velocities, which suggests that a viscous effect may play a role in the aperture flow.

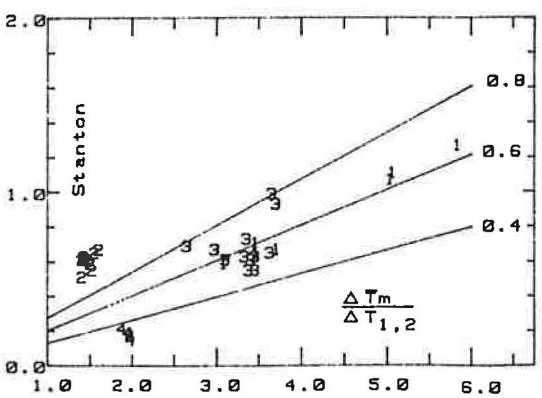

Fig. 4. Stanton number versus temperature stratification index

The heat transfer results are alternatively shown in Fig. 5 in the form of a Nusselt-Rayleigh number relation for all four configurations. The ordinate is the Nusselt number defined as below;

$$Nu = q\cdot H/\Delta T\cdot k \quad . \qquad (8)$$

The abscissa was generated with a Rayleigh number based on a heating surface-to-cooling surface height as the characteristic length.

The data in this plot are considerably scattered due in part to the different levels of heat loss permitted. Least square linear curves have been shown for each configuration. Both constant heat flux and constant wall temperature boundary condition results are shown. No significant difference was noted between the two boundary conditions. The core heat loss, however, may play a role in muting the effects of the different boundary conditions.

Now the effect of elevation change on heat transfer is considered. Based on the mean-plate height as the characteristic length, the apertures performed as reported in previous experiments, that is, heat transfer depends strongly on aperture height. Nansteel and Grief reported a Nusselt number dependence on the aperture height-enclosure height ratio;

$$Nu \propto (H/D)^{0.414} \quad . \qquad (9)$$

Based on configurations 1 and 2 in Fig. 5 the exponent in the above equation is 0.25.

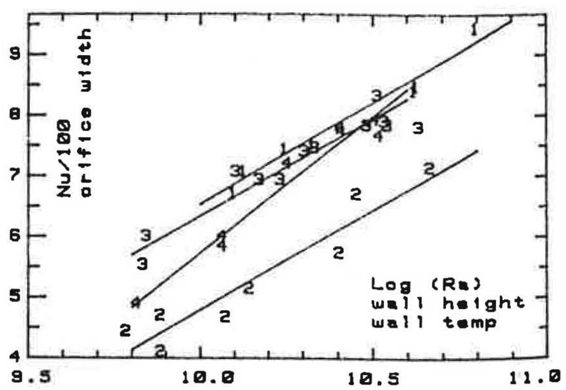

Fig. 5. Nusselt versus Rayleigh number.

With this exponent, heat transfer for configuration 3, without the elevation change is estimated to be only 3% less than with the elevation change. This increase in performance is within the error band for these results.

5. CONCLUSIONS

1. The Stanton number serves as an effective non-dimensional heat transfer variable for flow through apertures.
2. Enclosure heat loss affects the core temperature stratification and ultimately the energy transport through an aperture.
3. The level of elevation change investigated did not significantly enhance the heat transfer.
4. Core stratification plays an important role in energy transport through an aperture.

6. ACKNOWLEDGEMENTS

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