**STUDIES ON METHODS OF REDUCING HEAT LOSSES FROM
FLAT PLATE SOLAR COLLECTORS**

**Phase 2. Annual Progress Report,
February 1, 1976—January 1, 1977**

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MASTER

ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Division of Solar Energy

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STUDIES ON METHODS OF
REDUCING HEAT LOSSES FROM FLAT
PLATE SOLAR COLLECTORS
PHASE 2

Annual Progress Report
for Period Feb. 1, 1976-Jan. 1, 1977

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ABSTRACT

Improvements to flat plate solar collectors for heating and cooling of buildings are being investigated through two parallel studies. The first study, which deals with the free convective heat loss from V-corrugated absorber plate to a plane glass cover, has shown that, for the same average spacing, the free convective heat loss is greater for a V-corrugated absorber plate than for a plane absorber plate. However, provided the average spacing is large enough, the amount of increase is slight. The second study, which deals with the free convective heat loss in a honeycomb solar collector in which the honeycomb consists of a set of horizontal partitions, or slits, has shown that provided the solar collector is tilted to near vertical, such a honeycomb gives better free convective loss suppression than does a square-celled honeycomb having the same amount of material.

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1. SUMMARY

This report describes the results of two studies, both of which are related to methods for improving the efficiency of flat plate solar collectors.

The first study deals with the free convective heat loss from a V-corrugated absorber plate to a plane transparent (glass) cover. Measurements carried out in the University of Waterloo Natural Convection Apparatus give Nusselt numbers as a function of Rayleigh number for angles of inclination of the plate from the horizontal of 0, 30, 45 and 60 degree. The values of the depth ratio (the ratio of the average depth of the air layer to the depth of the V's) investigated were 1, 2.5 and 4; the angle of opening of the V's was 60°. It is concluded that the heat transfer is greater than that for two parallel plane plates having the same average spacing, by up to 50% for a depth ratio of unity. However, for the larger values of the depth ratio, the increase in heat transfer is slight.

The second study deals with the free convective heat transfer occurring in a honeycomb solar collector in which the honeycomb consists simply of a set of horizontal partitions (or slits) oriented normal to the plate. Measurements were carried out on this problem on the University of Waterloo Natural Convection Apparatus. Aspect ratios (plate spacing to distance between partitions) of 5 and 10 have been investigated, as have slit materials both opaque and transparent to thermal radiation. A significant conclusion is that for angles of inclination of the plate

from the horizontal ranging from 70 to 90 degree, the horizontal slit honeycomb gives superior convection suppression to a square-celled honeycomb, for the same amount of honeycomb wall material.

2. INTRODUCTION

Methods must be found for reducing heat losses from the conventional flat plate solar collector, if it is to perform efficiently in solar heating and cooling systems for buildings. In particular some additional mechanism or mechanisms must be incorporated into its design in order to substantially decrease losses from the collector top. These mechanisms must be economic, durable and practical. More important, they must not substantially decrease the solar radiation input to the collector plate.

The development of new such mechanisms, and the study of critical problems associated with mechanisms already suggested, is the broad objective of the research described in this report.

This broad objective is being pursued through two parallel studies:

(i) a study of free convective heat loss across the inclined air layers which exists in a flat plate solar collector having a V-corrugated absorber plate.

(ii) a study of free convective heat loss across the inclined air layer which exists in a flat plate solar collector incorporating a honeycomb of large plan aspect ratio.

The background and specific objectives of these individual studies are as follows:

Study 1: Natural Convection across an Inclined Air Layer Confined Between one Flat Plate and one V-corrugated plate.

A definitive study on the free convective heat loss across an

inclined air layer confined between two flat parallel plates was carried out at the University of Waterloo in the Phase 1 study [1]* and the resulting correlation equations [2,3] are now being used by many workers for characterizing the heat loss across the air gaps in flat plate solar collectors. The question now arises: is this equation still valid in the case where one or both of the bounding surfaces is not flat, but contains various protrusions (such as tubes), or if it is in some way roughened or corrugated?

As a first step in answering this question, the combination of the V-corrugated plate and the flat plate is being studied. Such a combination is of special interest since it occurs in the design of an efficient solar air heater developed in Australia [4,5]. In this air heater the absorber plate is V-corrugated so as to give two beneficial effects. First, the V-corrugating of the absorber plate improves its directional selectivity - particularly if the plate is slightly spectrally selective at the start [6]. Second, it exposes more surface area to the flowing air behind the plate, thereby enhancing the forced convective heat exchange between the circulating air and the absorber plate.

If the absorber plate is selective, the dominant heat loss is by free convection and hence the proper evaluation of this air heater requires an accurate knowledge of the free convective loss in this geometry. The combination of a V-corrugated plate and a flat plate is also of special interest in light of Phase 1 recent studies [1,7] indicating

*Numbers in square brackets denote reference listed in Section 6.

that V-corrugating the inner transparent cover of a two-cover collector will increase that cover's solar transmittance. In this instance it is the upper plate rather than the lower one which is V-corrugated.

The objective of this study, therefore, is to generate a correlation equation for the free convective loss across this type of air layer, covering the range of variables applicable to solar collectors. A secondary objective is to compare this correlation equation with that applying to an air layer bounded by two plane surfaces, so as to quantitatively indicate the effect of protusion and other forms of large-scale roughening of the absorber plate on the free convective loss.

Study 2: Natural Convection Across an Inclined Air Layer Constrained by a Honeycomb of Large Plan Aspect Ratio.

A number of recent studies [8,9] have demonstrated the improvement in solar collector efficiency obtained by incorporating a transparent honeycomb between the absorber plate and glass cover in a conventional one glass cover flat plate collector. As was indicated in an early analytical study [10], the improvement is due both to reduction in radiant losses by the honeycomb, and due to suppression of the free convective loss by the honeycomb. In the Phase 1 study at the University of Waterloo [2,11], the suppression of the free convection loss by a square-celled honeycomb was studied in detail, and a correlation equation for the free convective loss covering the range of parameters suitable for solar collectors was obtained. The purpose of the present study

is to extend the phase 1 study so as to include rectangular honeycombs with the long sides in the horizontal direction, the limiting case of which is a set of horizontal partitions, or slits.

The following sections of this report give the results obtained to date on these two studies; Study 1 is contained in Section 3, Study 2 is contained in Section 4.

3. FREE CONVECTION ACROSS
INCLINED AIR LAYERS BOUNDED BY ONE
FLAT SURFACE AND ONE V-CORRUGATED SURFACE

3.1 Description of Problem

Figure 3-1 shows a sketch of the problem considered. The air layer is assumed to be very extensive in the direction normal to the plane of the drawing. The flat lower surface is assumed to be inclined at some angle θ from the horizontal. The angle of opening of the V's was chosen at 60 degrees since Australian air-heater design studies had indicated this angle to be optimal when one takes into account radiant heat transfer, forced convective heat transfer and material and manufacturing considerations. The width of the layer, W , is assumed to be much greater than the spacing, L . The ratio of L to the depth of the V's, H , is termed the depth ratio: $A = L/H$. The lower flat surface is maintained at a uniform temperature T_h , greater than that of the upper V-corrugated surface, which is uniform in temperature at T_c , by an amount $\Delta T = T_h - T_c$.

As far as heat transfer across the air layer is concerned, the physical problem shown in Figure 1, is identical to that where the lower surface is V-corrugated and at T_h and the upper one is flat and at $T_c < T_h$. This latter is the physical situation actually occurring in the flat plate solar collector having a V-corrugated absorber plate and a flat glass cover. That the two situations are identical (called reciprocity) can be shown by a number of procedures. (One involves writing down the fundamental mass, momentum and energy equations and

Figure 3-1

Sketch of the inclined air layer bound by a V-corrugated plate and a flat plate.

judiciously dedimensionalizing them so as to show that the flow in both situations must satisfy identical equations.) It is planned to demonstrate the reciprocity experimentally during the course of the study.

In the case of the present experimental study on the University of Waterloo Natural Convection Apparatus, the arrangement shown in Figure 3.1 was chosen since it is more easily incorporated into the Apparatus.

3.2 Previous Studies

A full review of previous studies relevant to an inclined air layer bounded by two parallel flat plates is given in [2,3]. For the case where one of the plates is V-corrugated, the only relevant study to our knowledge is that of Chinnappa [12]. In this study, the angle of opening of the V's was 60 degrees, and heating was from below, with the V-corrugated plate being the lower plate. The angle of inclination of the flat plate from horizontal was zero degree. Since the air pressure was not varied (as is the case in the University of Waterloo Natural Convection Apparatus) only a narrow range in Rayleigh number was covered for any particular set-up. No effect of spacing ratio A , was observed, and heat transfer results were found to be intermediate between those for an air layer bounded between two parallel flat surface inclined at 0 degree and 60 degree from the horizontal.

3.3 Description of Experimental Apparatus

The University of Waterloo Natural Convection Apparatus which was used in this experiment is fully described elsewhere [2,11,13,14,15].

In the present study, the upper flat copper plate of the apparatus (the cooled plate) was replaced by a V-corrugated copper plate. This V-corrugated plate was constructed by folding a .032 inch thick copper sheet into the prescribed form and soldering tubes to its rear (top-side) in alternate V-grooves. These tubes were attached to headers at each end, and in the experiments, cool circulating water from a constant temperature bath is circulated through these tubes. Energy balance and fin calculations indicated that the plate would remain essentially isothermal (to within about .1F) at the heat flux rates experienced in the apparatus. Copper-constantan thermocouples were attached to representative points in the central area of the plate so as to determine the temperature difference between upper and lower plates, in the usual way for the apparatus. Comparisons of the readings of the various thermocouples verified the near-isothermality of the V-corrugated plate. The peripheral dimensions of the V-corrugated plate was the same as that of the lower plate- namely 22 inch by 24 inch. This gives a value for W , (Figure 3-1) of 24 inch. The height of the corrugations, H , was made .5 inch; a second V-corrugated plate was also made having an $H = 1.5$ inch and a sheet thickness of .064 inch.

The heat flux across the air layer was measured at the lower plate using the heat flux meter and heater plate method, in the usual way for the apparatus. This measurement samples the heat flux in the central 5 inch by 5 inch area of the plates and is thought to be representative of the average heat flux over the entire area of plates

having a large value of W , such as in solar collectors.

The air layer was sealed around the periphery of the two plates by stretching aluminum foil between the plates and giving it a good thermal contact at each plate. This foil is of sufficient thickness (.005 inch) to ensure a linear temperature rise from one plate to the other, thereby ensuring no net loss of heat from the air layer to the surrounds.

3.4 Dimensional Analysis

For the present set of experiments the Nusselt number is a function of the following dimensionless groups:

$$Nu = Nu (Ra, A, \theta) \quad (3-1)$$

where Ra is the Rayleigh number defined by:

$$Ra = \frac{g\beta\Delta TL^3}{\nu\lambda} \quad (3-2)$$

where g = acceleration of gravity and β the thermal expansion coefficient, ν the kinematic viscosity and λ the thermal diffusivity of air. The depth ratio $A = L/H$. The angle of inclination θ is as defined in Figure 3-1. The Nusselt number is defined by:

$$Nu = \frac{qL}{k\Delta T} \quad (3-3)$$

where q is the heat flux across the air layer (based on unit area of flat surface on the lower plate), exclusive of radiative exchange, and k is the thermal conductivity of air. Fluid properties are evaluated at the arithmetic mean of the two plate temperatures.

Other dimensionless groups of interest are: the Prandtl number, Pr , which is fixed at that for air at moderate temperatures, $Pr \approx .71$, and the angle of opening of the V's, which is fixed at 60 degree.

The aspect ratio W/L is assumed to be sufficiently large in the experiments that the effects of the periphery are not felt at the central area where the heat flux is measured. Thus the results are expected to be applicable to that situation existing in the limit of large W/L . Our experience gained on studies having two flat surfaces [2,14] have shown that this expectation is satisfied provided $\theta \leq 60$ degree. For the range $60 < \theta \leq 90$ degree the effects of the periphery are felt in the central area. Consequently measurements were not taken for $\theta > 60$ degree.

3.5 Range of Measurements

The approximate range of variables of interest for the design of solar collectors is as follows:

$$10^3 < Ra < 10^6$$

$$1 \leq A \leq 4$$

$$0 \leq \theta \leq 90 \text{ degree}$$

Except for the range $60 < \theta \leq 90$ alluded to above, this range was attempted to be covered in the present experiments. Measurements were taken at values for θ of 0, 30, 45 and 60 degree, and for A of 1.0, 2.5 and 4.

As is customary on the University of Waterloo Apparatus, the Rayleigh number was varied by varying the air pressure, keeping L and ΔT fixed. This permits the full variation of the Rayleigh number while minimizing the number of experimental set-ups necessary and also minimizing errors associated with stray effects. The full range in Rayleigh number could be covered with the V-corrugated plate having $H = .5'$, except, in the case $A = 1$, for the range $10^5 < Ra < 10^6$. The V-corrugated plate having $H = 1.5'$ permitted this range to be covered.

3.6 Evaluation of Nusselt Number at Low Rayleigh Number, Nu_c

At very low Ra , the heat transfer across the air layer is by conduction and radiation. To be able to separate the contribution of these components, separate calculations or measurements of the conduction component were necessary. An exact analytic solution to the conduction problem does not seem to be possible, but upper and lower bounds were established by the method of [16] and Elrod [17], and found to be:

$$A \ln \left(\frac{2A+1}{2A-1} \right) \leq Nu_c \leq \frac{3\sqrt{3}}{\pi} A \ln \left(\frac{2A+1}{2A-1} \right) \quad (3-4)$$

where Nu_c is the Nusselt number at very low Rayleigh number, where conduction dominates. Therefore, based on the average resistance to heat transfer of the two bounds:

$$Nu_c \approx 1.246 A \ln \left(\frac{2A+1}{2A-1} \right) \pm 25\% \quad (3-5)$$

Figure 3-2

Evaluation of the conduction Nusselt number, Nu_c , showing the results of analysis giving the upper and lower bounds, electrical analog experiments, and the correlation equation used. (V-corrugated plate problem).

A plot of the upper and lower bounds, based on equation (3-4), is presented in Fig. 3.2.

To establish more precisely the conduction transfer, Nu_c , an electric analog experiment was conducted using electrically conducting paper [18]. These results, converted to a Nusselt number based on the thermal properties of air, are also plotted in Fig. 3-2. An empirical relation which fits the data to within experimental accuracy is

$$Nu_c = A \ln \left(\frac{2A+1}{2A-1} \right) / [1 - 0.13/A - 0.067/A^2] \quad (3-6)$$

The guidance for the form of this equation comes from equation (3-5) and the fact that $Nu_c \rightarrow 1$ as $A \rightarrow \infty$. A comparison between the analog data points and this equation is made in Fig. 3-2.

It will be noted that as A approaches 0.5, Nu_c becomes infinitely large because the bottom of the corrugations come into contact with the flat plate. Equation (3-6) is not valid below $A = 0.61$ since the upper bound on Nu_c in equation (3-4) is exceeded.

3.7 Evaluation of Radiative Heat Transfer Correction

With the plates in the horizontal position ($\theta = 0$) and at the lowest Rayleigh number possible ($Ra \approx 10$) the heat transfer across the air gap was measured. Since this is substantially below the observed critical Rayleigh number, it is assumed that the heat flow is by radiation and conduction alone. Subtracting the conduction component, using the results of the previous section, yields the heat transfer coefficient

for radiation. Since the plate temperatures remained essentially unchanged over the entire Ra and θ range, the heat transfer by radiation can be calculated using this heat transfer coefficient for all subsequent measurements, including those where free convection is present. Therefore, the radiative transfer could be subtracted away from the total heat transfer to yield the desired Nusselt number for convection plus conduction, Nu . The radiation estimate agreed well with those obtained in the same apparatus, but based on exchange between two flat surfaces [2, 14, 15].

3.8 Results

The experimental determinations of Nu are plotted against $Ra \cos \theta$ in Figs. 3-3 to 3-6. Figure 3-3 presents the results for $A = 1$ for the plate with smaller corrugations ($H = 0.5$ inch) at the four angles of inclination. Figure 3-4 is also for $A = 1$ but the plate has the deeper corrugations permitting Ra values up to nearly 10^6 . Results from the plates with different H 's agree to within 3% for $Ra > 2 \times 10^4$ but differ by up to 12% for $10^3 < Ra_L < 10^4$, the data for the larger H being higher. It is thought that, in this latter range of Ra , the spacing at the larger H is not sufficiently small to make edge conditions entirely negligible. Hence the data for $H = .5$ is recommended. (It is estimated that the maximum experimental error in Nu is $\pm 4\%$ and Ra may be in error by up to $\pm 2\%$).

For the two other aspect ratios ($A = 2.5$ and $A = 4.0$) the data are plotted in Figs. 3-5 and 3-6, for four angles of inclination.

Figure 3-3

Measurements of Nusselt number as a function of $Ra \cos \theta$ for $A = 1$, $H = .5$ inch. (V-corrugated plate problem).

Figure 3-4

Measurements of Nusselt number as a function of $Ra \cos \theta$ for $A = 1$, $H = 1.25$ inch. (V-corrugated plate problem).

Figure 3-5

Measurements of Nusselt number as a function of $Ra \cos \theta$ for $A = 2.5$ (V-corrugated plate problem).

$\circ$ { V-CORRUGATED UPPER PLATE,
 FLAT LOWER PLATE,
 EXPERIMENTAL POINTS
 — CORRELATION EQUATION FOR PLANE AIR LAYER [2]

Figure 3-6

Measurements of Nusselt number as a function of $Ra \cos \theta$ for $A = 4$, (V-corrugated plate problem).

3.9 Discussion of Results

It is of interest to observe how well the correlation equation for Nu developed previously [2], based on two flat plates compares with the present measurements for the same spacing. The solid lines in Figs. 3-3 to 3-6 are plotted from this equation. It is seen from the Figures that the conduction Nusselt number is higher by 24%, 7% and 4% for $A = 1$, 2.5, and 4.0 respectively. The Rayleigh numbers at which convection begins (for $\theta = 0^\circ$) or starts to play a significant role (for $\theta > 0^\circ$) are about the same for both the flat plate equations and the present data. There is also reasonable agreement in the convection-dominant zone up to $Ra \approx 4 \times 10^4$ for $A = 4.0$ and $Ra \approx 10^4$ for $A = 2.5$, and in a very narrow band about $Ra = 8 \times 10^3$ for $A = 1.0$. At still higher Ra , the data fall progressively higher than the corresponding flat-plate data. It is speculated that this is caused by cold jets of air departing from the tips between the V-grooves and penetrating the "boundary layer" on the opposite plate. If this is the mechanism, then it would be expected that the Nusselt numbers at much higher Rayleigh numbers would not continue to depart from the solid lines, but rather approach from above asymptotes parallel to those for the parallel flat plates.

3.10 Future Work

The remaining work in this study is the development of empirical equations, to fit the present data, which are suitable for design. Equations for the conduction regime and the low Ra_L regime have been

obtained. Following the method of Raithby and Hollands [19], the high-Ra asymptotes have also been predicted. It remains to find suitable combinations and additions to these equations to fit the entire body of data.

4. FREE CONVECTION

ACROSS AIR LAYERS CONSTRAINED BY

A HONEYCOMB OF LARGE PLAN ASPECT RATIO

4.1 Problem Background and Description

The method used to specify the various dimensions and aspect ratios will first be given. Referring to Figure 4-1, the honeycomb panel is assumed to be facing south. The dimension of the honeycomb cell in the east-west direction is L_{EW} , in the north-south direction L_{NS} , and the depth of the honeycomb panel, L . The angle of tilt of the panel from horizontal is θ . The two important aspect ratios namely A_E and A_P , are defined by:

$$A_E = L/L_{NS}$$

$$A_P = L_{EW}/L_{NS}.$$

From a purely free convective point of view, rectangular honeycombs may not be desirable for a horizontal or near-horizontal (say up to $\theta \approx 15^\circ$) collector inclination, as may be used for summer cooling in the southern U.S. This is because there is an equal tendency for free convective cells to be formed in any direction - therefore a laterally symmetric cell shape such as a square or hexagonal cell is called for.

For moderate collector tilts (range $\theta = 15^\circ$ to $\theta \approx 45^\circ$), the first incipient fluid motion is rolls with axis in the northsouth upslope direction [20]. Therefore it would appear that for this inclination,

$$A_E = A_{\text{ELEVATION}} = L / L_{NS}$$

$$A_P = A_{\text{PLAN}} = L_{EW} / L_{NS}$$

Figure 4-1

Sketch showing pertinent dimensions of a rectangular-celled honeycomb.

honeycombs with $A_p < 1$ are called for. However, since there is still some tendency for rolls with axis in the east-west direction, A_p can not be made equal to zero. From the point of view of minimizing the quantity of honeycomb wall material per unit area (and hence minimizing the honeycomb cost, and maximizing its solar transmittance) it would appear that the optimum shape of cell would be a rectangle with L_{EW} of sufficient size to "just" suppress the incipient north-south rolls, and L_{NS} of sufficient size to "just" suppress the incipient east-west rolls. That is, there will be an "optimum" value of the plan aspect ratio, A_p , which will be called $A_{p,opt}$. Since the relative stability of the two rolls is dependent upon the angle of tilt, θ , $A_{p,opt}$ is a function of θ . (At $\theta = 0$, the tendency is the same for both directions; therefore $A_{p,opt}$ for this angle is unity). The exact nature of the function $A_{p,opt}(\theta)$ is not known. However, gaining of this knowledge is of some importance to the design of honeycomb solar collectors.

For angles of tilt near vertical ($70 \leq \theta \leq 90^\circ$), a reversal of the two rolls takes place. For these angles the flow situation is more unstable to east-west rolls than to north-south rolls. Therefore $A_{p,opt} > 1$. In fact at $\theta = 90^\circ$, there are theoretical grounds for believing that there is no tendency for north-south rolls so that $A_{p,opt} = \infty$. In this case the honeycomb becomes simply a set of horizontal slits - rather like a venetian blind.

This range of angle of inclination is of some importance since it corresponds to near optimal for winter heating in northern United States and Canada. For these inclinations, then, what is needed is not

strictly a honeycomb, but a set of horizontal slits.* Consequently this is the geometry, being investigated in the present study.

4.2 Previous Studies**

Catton and co-workers [21,22], have performed analytical and experimental investigations of flow in rectangular slots similar to that enclosed by horizontal slits. However, a sufficient range of aspect ratios is not covered and the Prandtl number is not that of interest. Some work has been carried out by Probert and Ward [23] on what is in fact the effect of horizontal slits in a vertical air gap. Positive results (i.e. suppression of free convective heat transfer) was found but the results are not in sufficient detail to be used for design.

J.E. Hart ²⁴ has examined in some detail a geometry which, although treated as a problem in geophysics, is very close to a single cell in a set of horizontal slits with $\theta = 90$. A number of workers, e.g., references [25,26,27] have examined a single cell with $\theta = 90$ by numerical modelling and experiments; however, the side-wall boundary conditions are not those applying in solar collector. Charters and Koutsouheras [28] report interesting numerical modelling of the problem covering all θ but have not carried out experiments. In summary, the problem has been examined but there is not sufficient information to design with confidence, nor to seek optimized designs. Good experimental data is particularly lacking.

*For the purposes of this report we will consider a set of horizontal slits to be a special case of a honeycomb.

**The previous studies noted in this section will not include the relevant literature corresponding to the problem where $\theta=0$. This problem is reviewed in [32].

Very recently Arnold et al [29] reported a set of experiments on rectangular honeycombs having approximate values for A_p of 2, 4 and 8 and for A_E of 4. The fluid was an opaque liquid and the wall conductivity was low. The radiant coupling, present in honeycombs with air, was not accounted for and critical Rayleigh numbers were much lower than would be expected in solar collectors for the same aspect ratio. An important conclusion was that provided A_p is greater or equal to 4, it has no effect on the heat transfer. This conclusion may be limited to the value of A_E investigated, as well as to the values of the other pertinent groups.

4.3 Description of Apparatus

The University of Waterloo Natural Convection Apparatus is being used to measure free convective heat transfer across a horizontal slit honeycomb. The set of horizontal slits is constructed by winding polyethylene film tightly on a frame which is attached to the periphery of the copper plates. The fluid contained between the slits is air. The value of L_{EW} (Figure 4-1) is thereby fixed at 22 inch since this is the width of the copper plates. The value of L_{NS} is typically 1/2 inch so that $A_p \approx 44$. Two types of polyethylene film are being used. The first is clear polyethylene having a thickness of .0015 inch; the second is black polyethylene having a thickness of .006 inch.

4.4 Dimensional Analysis and Range of Measurements

The Nusselt number characterizing the heat transfer across a

rectangular honeycomb is a function of the following dimensionless groups:

$$Nu = Nu(Ra, Pr, A_E, A_p, \theta, C, N, H, \epsilon, \epsilon_{p1}, \epsilon_{p2})$$

The Rayleigh number Ra , has its usual meaning and is based on dimension L and temperature difference $(T_H - T_C)$, (Fig. 4-1). The Prandtl number, Pr , is .71 which is the same as that applying in solar collectors. The range of A_E of interest to solar collector design is from 3 to 10. The value of A_p is of order 40 which is considered sufficiently large to be, for practical purposes, infinite. The angle of tilt of interest, θ , (Figure 4-1) is from 0 to 90 degree. The groups C and H represent the effect of heat conduction along and across the honeycomb wall, respectively:

$$C = k_f L / (k_w \delta)$$

$$H = (k_f \delta) / (k_w L)$$

where k_f and k_w are the thermal conductivity of the fluid and wall material, respectively, and δ is the thickness of the honeycomb wall. The group N is representative of the effect of the radiant exchange across the honeycomb on the free convective exchange:

$$N = 4\sigma T_m^3 L / k_f$$

where T_m is the average temperature of the honeycomb ($T_m = (T_h + T_c)/2$) and σ is the Stefan-Boltzman constant. The quantities ϵ , ϵ_{p1} and ϵ_{p2} represent the emissivities of the honeycomb horizontal partitions (polyethylene film) and the top and bottom plates bounding the honeycomb respectively.

Edwards and Sun [30] have shown that the effect of ϵ , C and N

can be approximately combined into a single group, C^* given by:

$$\frac{1}{C^*} = \frac{1}{C} + \frac{\epsilon}{4-\epsilon} \cdot \frac{N}{A^2}$$

Earlier studies carried out in Phase 1 of this contract [1], on square-celled honeycombs, used clear polyethelene of thickness .0015 inch and emissivity $\epsilon = .13$. Polished copper plates were used for the bounding surfaces giving $\epsilon_{p1} = \epsilon_{p2} = .06$. In the present investigation it is planned to investigate the effect of changing ϵ , ϵ_{p1} and ϵ_{p2} . Thus as well as the clear polyethelene film, a black polyethelene film is also being investigated having wall thickness .006 inch and emissivity $\epsilon \sim .91$. Also to be investigated, although not yet available for report are measurements with the copper plates painted black so as to give ϵ_{p1} or $\epsilon_{p2} \sim .9$. Runs with blackened copper plates should be more representative of the situation in solar collectors, although the effect of ϵ_{p1} and ϵ_{p2} on the convective heat transfer is expected to be slight.

Table 4-1 gives the values of the important parameters for the three experimental set ups for which results are available to date.

Table 4-1

Values of Parameters Tested

Exp set No.	Film Thickness inch.	L inch.	L_{NS}	A_E	C	N	ϵ	C^*
1	0.0015	2.5	0.5	5	260.1	15.67	0.13	40.16
2	0.0015	2.5	0.5	10	520.2	15.67	0.13	43.52
3	0.006	5.0	0.5	5	65.0	15.67	0.92	4.57

The values of ϵ_{p1} and ϵ_{p2} were .06 for all runs and H was of order 10^{-6} . Since horizontal slits are expected to be most valuable for near vertical solar collector orientations, the angles chosen for θ were 60° , 75° and 90° . In addition, runs were made at $\theta = 0^\circ$ for comparison with available stability theories.

4.5 Results

The results for experimental set-ups 1,2 and 3 are given in Figures 4-2, 4-3, and 4-4 respectively. In Figure 4-2 are also shown the results for a square-celled honeycomb having the same aspect ratio, based on a correlation equation developed from the Phase 1 study [1,11] using the same wall material. The square-celled honeycomb is seen to give greater convection suppression at all angles but its advantage over a set of horizontal slits decreases as the angle of inclination, θ , increase. It must be remembered that a square-celled honeycomb having $A = 5$ contains twice the amount of film material as a set of horizontal slits with $A = 5$. The two honeycomb geometries (square-celled and horizontal slits) are compared for the same amount of honeycomb material in Figure 4-3. It is seen that for $\theta < 75$ degrees the square cells still give better suppression but for $\theta \geq 75$ the set of horizontal slits gives better suppression at least around $Nu = 1.2$, which is the typical operating point in a honeycomb collector.

The shape of the curves are also different. For a square-celled honeycomb the abrupt rise in Nusselt number at a critical value was observed only at $\theta = 0$; at other values of θ a gradual rise was observed, and a precise critical value cannot be defined. In the case of the

RAYLEIGH NUMBER, Ra

Figure 4-2

Nusselt number versus Rayleigh number plots for a set of horizontal slits for experimental set up number 1 (Table 4-1).

Figure 4-3

Nusselt number versus Rayleigh number plots for experimental set up number 2, (Table 4-1).

Figure 4-4

Nusselt number versus Rayleigh number plots for experimental set up number 3 (Table 4-1).

horizontal slits an abrupt rise was observed for all $\theta \leq 75$ with Nu very nearly equal to unity (complete suppression of convection) up to a critical value. For a given set of horizontal slits, the dependence of this critical Rayleigh number on the angle of tilt has been found to be given approximately by :

$$Ra_c = \frac{Ra_c(\theta=0)}{\cos\theta}$$

The value of $Ra_c(\theta=0)$ was found to be predictable using the method described by Sun [31,32] to within about 5%, for all three experimental set-ups.

Figure 4-4 gives the results for a black polyethelene sheet with $A_E = 5$, and should be compared to Figure 4-2 which is for clear polyethelene having the same A_E . The increased wall emissivity causes a greater radiant coupling which acts to stabilize the fluid layer. Hence critical Rayleigh numbers are higher for the black film.

5. CONCLUSIONS

1. The free convective heat loss from a V-corrugated surface to a parallel plane surface exceeds that between two plane parallel surfaces, for the same average spacing and temperature difference. If the ratio of the height of the V's to the average plate spacing is one or more the amount of excess can be 50% or higher. However, if this ratio is made less than .25, the excess is relatively small (less than 10%) at least over the range of variables of interest in solar collectors ($10^4 < Ra < 10^5$).

2(a) A honeycomb consisting of a set of horizontal partitions, or slits, is effective as a convection suppressor, and such a honeycomb can be incorporated into a solar collector. If the angle of tilt of the collector is near horizontal, the honeycomb geometry is much less effective in suppressing convection currents than a square-celled honeycomb, for the same amount of honeycomb material. However, if the angle of tilt of the collector is near vertical this honeycomb geometry is slightly more effective than a square-celled honeycomb, for the same amount of honeycomb material.

2(b) A "sharp" critical Rayleigh number is observed for the honeycomb consisting of horizontal slits, even for angles of inclination, θ , up to 60 degrees. This critical Rayleigh number, Ra_c , can be scaled from the corresponding horizontal one by means of the scaling factor $\cos \theta$ - i.e. $Ra_c \cos \theta = \text{constant}$. The value of Ra_c at $\theta = 0$ can be predicted with good accuracy using the method outlined by Sun and Edwards [31,32].

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8. CONTRACT COMPLIANCE

The principal investigator, Professor K.G.T. Hollands, has spent on average 10% of his time on this project since the beginning of the current term (Jan. 31, 1976). Over the remaining 4 months he plans to spend 20% of his time.

All research reported herein complies with the contract requirements.

No technical reports, preprints or articles have so far been produced in the present phase. However, resulting from phase 1 are the papers listed as references [2], [3], [7], and [11].