University of
Waterloo Research Institute

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

ANNUAL PROGRESS REPORT

K. G. T. HOLLANDS

MASTER

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FLAT PLATE SOLAR COLLECTORS

PROGRESS REPORT

for Period August 1, 1974 to July 31, 1975

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September 1975

Prepared For

THE U.S. ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION

UNDER CONTRACT NO. E(11-1) 2597

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ABSTRACT

Improvements to flat plate solar collectors for heating and cooling of buildings are being evaluated by three parallel studies. The first study has demonstrated that suppression of convective heat transfer by honeycomb panels is possible even when the panel is inclined from the horizontal. The second study has resulted in a precise, simple equation for the free convective heat transfer through an inclined air layer. The third study has theoretically demonstrated that if an inner cover is V-corrugated, its solar transmittance may be enhanced.

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SUMMARY

The objective of this research project is to examine ways of reducing heat losses from flat-plate solar collectors so that they may perform efficiently in solar heating and cooling systems for buildings. This broad objective is being pursued through three specific parallel studies: (I) an experimental and theoretical study on suppression of free convective heat transfer by inclined honeycomb panels; (II) a definitive study of natural convection heat transfer across inclined air layers; and (III) a study of a new innovation in flat-plate solar collectors, the V-corrugated inner cover.

The main objective of the honeycomb study is to establish whether in fact effective convection suppression is possible when the honeycomb is in the inclined position. It has been found that in fact it is possible. In particular, a square-celled honeycomb of aspect ratio (cell height to width ratio) of 5 to 1, when inclined at 45° has been demonstrated experimentally to suppress convective heat transfer up to a critical Rayleigh number of 850,000. (Without a honeycomb the critical Rayleigh number is 2,400). Measurements are reported on honeycombs having aspect ratios of 2, 3, 4 and 5. Plots, suitable for design, giving critical Rayleigh number as a function of aspect ratio and angle of inclination are contained in this report.

The theoretical study involves computer simulation of the flow in a honeycomb cell. They indicate that at angles near vertical, very considerable heat transfer is carried by the base flow. Infact, the "critical" Rayleigh number (corresponding to the point of significant departure of the Nusselt number from unity) is established by the base flow, rather than an instability, which is the case in the horizontal position.

The definitive study of the free convective heat transfer across inclined air layers has been concluded, for angles up to about 70° , and a single correlation equation obtained. Consideration has been given to recommendations for optimum width of air gap in solar collectors. It is concluded that establishing the proper gap-width will involve a systems study as well as purely a free convection study. However, the design encompassing a certain gap-width giving a "local" minimum in convective heat transfer appears very promising.

Theoretical calculations, based on a Monte Carlo radiation analysis, has demonstrated that a V-corrugated inner cover, may, under certain circumstances produce a higher transmittance of solar radiation than a flat inner cover. In particular very high transmittances ($>99\%$) are predicted for a V-corrugated teflon film inner cover. Experiments to confirm this are underway.

STATEMENT OF
OBJECTIVES

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 the 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 objective of the present research project.

The broad objective is to be pursued through three parallel studies:

(I) a study of suppression of free convective heat transfer across inclined honeycomb panels;

(II) a definitive study on free convective heat transfer through inclined air layers occurring in flat plate collectors;

(III) a study of a new device for minimizing top losses, - the V-corrugated inner cover.

The specific objectives of these individual studies are as follows:

Study 1: Natural convection heat losses on tilted honeycomb collectors.

That honeycombs suppress free convective heat transfer in the

horizontal position has already been demonstrated; moreover the conditions under which suppression is achieved in this situation are now largely defined. However, very little is known about these phenomena for the inclined position. The phenomena are being examined both theoretically and experimentally. The experimental study, carried out on the University of Waterloo Natural Convection Apparatus, is to determine if virtual suppression of convective heat transfer is possible in the inclined condition, and if so, to try to define the conditions under which virtual suppression takes place. The theoretical study is to examine stability of the base flow in the honeycomb cell, so as to point the way to the proper interpretation of the experimental results.

Study 2: Natural convection in inclined air layers.

The objective of this study is to make a new set of measurements on this problem, using the University of Waterloo Natural Convection Apparatus; and to interpret the results, using recently-acquired theoretical knowledge of the fluid flow in fluid layers, so as to produce a set of reliable, accurate and simple equations for predicting free convective heat loss across parallel plates in solar collectors.

Study 3: The V-corrugated inner cover.

The V-corrugated inner-cover is structured in such a way so as to re-direct reflected solar rays towards the absorbing plate so that they are not lost. The objective of this study is to find the conditions under which the overall transmittance of a V-corrugated inner cover is greater than that of the plane cover, if such conditions exist.

PROGRESS ON OBJECTIVES

Study 1: Natural Convection Heat Losses on Tilted Honeycomb Collectors

Experimental

The method of fabricating honeycombs from polyethylene sheet has been developed. The sheet used is 1.5×10^{-3} inch thick, which is sufficiently thin to be essentially non-absorbing to long-wave ($\lambda > 3\mu\text{m}$) radiation. This was substantiated by a transmission test (direct-beam) run on a spectrophotometer, the results of which are shown in Figure 1. Except for narrow bands at $6.9 \mu\text{m}$ and at $14 \mu\text{m}$, the polyethylene sheet is relatively transparent. The loss in transmission outside of these bands is almost certainly due to reflection and scattering, rather than absorption.

Figure 2 illustrates the method used for fabricating the honeycombs. Overlapping strips are joined along thin lines using a specially developed plastic heat-sealing tool, (capable of giving heat-sealed join lines as little as .01 inch wide), so as to form a square-celled array. The present honeycomb is considered to an improvement over that used in an earlier study [1] in that the walls are thin, and pressure-sensitive tape was eliminated. These factors help to make sure that the convective heat transfer mode is decoupled from the radiative and conductive modes.

The original research proposal called for testing of polyethylene honeycombs of aspect ratios of two, three and four with two honeycombs at each aspect ratio. With permission from the project manager this was changed to:

Figure 1

Measured Direct-Beam Transmittance, τ , of Polyethylene Film
used for Fabricating Honeycombs.

Figure 2

Sketch Showing Construction of Polyethylene Film Honeycomb
used for Free Convection Studies.

2 honeycombs of aspect ratio 2

2 honeycombs of aspect ratio 3

1 honeycomb of aspect ratio 4

1 honeycomb of aspect ratio 5.

That is, the test on a second aspect ratio 4 honeycomb was changed to a test of a single honeycomb of aspect ratio 5. All tests, except that on the second honeycomb at aspect ratio 2, have been completed. The results are shown in Figures 3-7. The actual dimensions of the honeycombs are given in the figures. The lower $D \times D$ face is heated, the upper $D \times D$ face is cooled. The angle θ referred to in the Figures is the angle of inclination of the ($D \times D$) face of the honeycomb, (not of the honeycomb cell axis), as measured from the horizontal.

The plot shows the data in dimensionless form, -i.e. in terms of Nusselt number and Rayleigh number. A Nusselt number of unity indicates pure conduction and hence "virtual" suppression of convective heat transfer. (The term "virtual suppression" is used here since it is known that, in the inclined position, there is always some convection, no matter how small the Rayleigh number. However, under some circumstances, the contribution of this convection to the Nusselt number is so small that it may for engineering purposes, be neglected; this we will call "virtual suppression".) Taking, for example, the 4 to 1 aspect ratio honeycomb (Figure 6), virtual suppression of free convective heat transfer takes place up to critical Rayleigh numbers of about 4×10^5 for $\theta=0^\circ$; 3×10^5 for $\theta = 30^\circ$; 2.5×10^5 for $\theta = 45^\circ$; 2×10^5 for $\theta = 60^\circ$ and 1.5×10^5 for $\theta = 90^\circ$.

Also shown on the figures is the free convective heat transfer which would occur across the air layer if no honeycomb is present

Figure 3

Measured Free Convective Heat Transfer Across Inclined Honeycomb, Honeycomb Aspect Ratio = 2.

Figure 4
Measured Free Convective Heat Transfer Across Inclined
Honeycomb, Honeycomb Aspect Ratio = 3.

Figure 5
Measured Free Convective Heat Transfer Across Inclined
Honeycomb, Honeycomb Aspect Ratio = 3.

Figure 6
Measured Free Convective Heat Transfer Across Inclined
Honeycomb, Honeycomb Aspect Ratio = 4.

Figure 7

Measured Free Convective Heat Transfer Across Inclined Honeycomb, Honeycomb Aspect Ratio = 5.

(curve labelled "no honeycomb"). This curve is based upon Reference [2] which resulted from Study 2 of the Research Project (See equation (1) of this report). The curve is shown as dashed for Rayleigh number $>10^5$ as use of this equation beyond this value involves extrapolation.

On comparing the honeycomb data on these Figures with the curve for no honeycomb, it is obvious that the honeycomb substantially reduces, and in some cases virtually eliminates the free convective heat transfer of the air. An interesting feature of these curves is that, in some cases, at large Rayleigh number, assuming extrapolation of Equation (1) is permitted, the honeycomb can actually increase the free convective heat transfer. This region must obviously be avoided in honeycomb design for flat plate collectors.

To establish the universality of these plots (i.e. their independence of such factors as the actual dimensions of the honeycomb cells (as opposed to the ratio of dimensions, as given by the aspect ratio), temperature difference, pressure etc), runs were made on each of two honeycombs, both having aspect ratio 3, but each having different cell dimensions. A direct comparison of the dimensionless data is shown in Figure 8. In these plots, Honeycomb No's 1 and 2 refer to those described by insets in Figures number 4 and 5 respectively. The fact that the data for both cases lie closely together indicates that the plots have some universal significance.

The close agreement shown in Figure 8 led us to conclude that it was not necessary to test two honeycombs at each proposed aspect ratio, and hence the repeat on aspect ratio of 4, was replaced by a test on a honeycomb of aspect ratio 5, as discussed earlier. It was

Figure 8

Graph Showing Comparison of Two Honeycombs, both having same Aspect Ratio, but each having Different Height. See Figures 4 and 5 for Details of Honeycombs.

also felt that the lower critical Rayleigh numbers experienced in the inclined position may necessitate honeycombs of aspect ratio greater than 4 in inclined solar collectors, and hence it was felt to be worth while to test one of aspect ratio 5.

For the horizontal layer ($\theta = 0$) it is common to define a critical Rayleigh number as being that Rayleigh number at which the Nusselt number departs from unity. In inclined layers, presumably due to the influence of the "base flow", the Nusselt number departure from unity is not as "abrupt" or "sharp" as it is in the case of the horizontal layer and hence the concept of the critical Rayleigh number, defined as such, becomes less useful. For engineering purposes it may be useful to define an "engineering" or "10%" critical Rayleigh number defined as that Rayleigh number at which Nusselt number is 10% greater than unity, or 1.1. Since the slope of graphs of Nu vs, Ra is greater at $Nu = 1.1$ than at $Nu = 1$, it is easier to determine this critical Rayleigh number from the charts. Moreover, this Rayleigh number may be useful in practice since one can design for it in the solar panel, and accept a 10% penalty on the Nusselt number, without this representing a significant penalty to the overall loss coefficient for the honeycomb. The plot of this '10%-critical Rayleigh number vs angle of inclination, θ , and aspect ratio is given in Figure 9, for aspect ratios of 3 and 4.

Figures 10 and 11 show this information in dimensional form. Based on air at one standard atmosphere pressure and mean temperature of 100F these plots show the critical temperature difference up to which free convective currents are virtually suppressed, for various assumed

Figure 9

The Quasi or '10%' Critical Rayleigh Number as a Function of Angle of Inclination for Aspect Ratios of 3 and 4.

Figure 10

Critical Temperature Difference for Virtual Convection Suppression by Inclined Honeycombs as a Function of Angle of Inclination, θ , and Honeycomb Depth, L , for Aspect Ratio = 3.

Figure 11

Critical Temperature Difference for Virtual Convection Suppression by Inclined Honeycombs as a Function of Angle of Inclination, θ , and Honeycomb Depth, L , for Aspect Ratio = 4.

honeycomb panel depths, L and angles from horizontal θ , for aspect ratios of 3 and 4. For example a honeycomb of aspect ratio 4 inclined at 45° from horizontal and 1.5 inch deep should virtually suppress free convective heat transfer up to temperature differences of 110 F.

Some comments should be made on the use of these results for design. As has been discussed earlier, care has been taken in these experiments to eliminate the coupling between the free convective heat transfer mode and the radiative and wall conduction modes. Other studies e.g. [6] have shown that, at least in the horizontal position, wall conduction tends to stabilize the flow and hence produce higher critical Rayleigh numbers. Moreover, [4], the effect of radiative coupling is to give an apparent increase to wall conduction, thereby also stabilizing the flow. In honeycombs in actual solar collectors, it is unlikely that the radiant and conductive de-coupling will be as effectively eliminated as on the present study. Consequently, the present results should give lower bounds for real critical Rayleigh numbers, and should therefore result in conservatively designed honeycombs.

Most of the results presented in this report were reported at the International Solar Energy Society Meeting in Los Angeles in July, 1975, [5].

Theoretical

The objective of the theoretical study on the free convective effects in inclined honeycomb cells is primarily to assist in the

interpretation of the experimental results.

As has been mentioned earlier, in the inclined cell, there exist fluid motion for any finite value of the Rayleigh number. The flow which exists at low Rayleigh number we call the base flow. It was anticipated in the proposal that the contribution of this base flow to the overall heat transfer should be small. At sufficiently large Rayleigh number, the base flow should become unstable and produce cells which convect heat transport strongly, - corresponding to the appearance of the Benard cells in the horizontal position. As a result of the results produced thus far by the theoretical study of this project, it is now known that, in fact, at least in the vertical position, very significant quantities of heat are being transported by the base flow. The gaining of this fact alone will assist greatly in interpreting the experimental results.

The full set of equations describing the base flow in an inclined honeycomb cell are highly complex and do not show promise of yielding an analytical solution. Therefore it was decided to model the three - dimensional base flow numerically on a computer.

The numerical scheme is written in terms of vorticity and a vector potential. This is found to be more satisfactory than a scheme using the primitive variables. Further, the use of vector potential and vorticity in the equations ensures satisfaction of the continuity condition at all times. The solutions obtained are the final steady state solutions of the unsteady flow problem with distorted time scales. The use of false transient terms reduces the equations to parabolic equations

and solution is made possible by a progression through time.

Some problems have been experienced in determining the best time step to progress forward. There are no simple ways of doing this except by repeated trials, especially at high Rayleigh number. Some problems were also experienced with computer storage, limiting the grid size to one-tenth of the cell depth. These problems are now overcome and based on comparisons with predictions of other workers, it is believed that the accuracy achieved in the stream function is quite good.

The program has been first applied to a square honeycomb cell at the vertical inclination, - that is, that geometry in which the two heated faces are vertical and the honeycomb walls are horizontal (Figure 12). The full set of three-dimensional equations were included. The first and most important result of this 3-dimensional analysis was that the dependence on the y direction (Figure 12) is small, indicating that a two-dimensional analysis of the problem should be sufficiently accurate. This is exemplified in Table 1 below which shows the stream functions for the three directions, ψ_x , ψ_y , and ψ_z for several values of the Rayleigh number.

TABLE 1

Rayleigh Number	ψ_x	ψ_z	ψ_y
25000	0.0129	0.0109	10.303
35000	0.0112	0.0107	8.836
80000	0.0146	0.0126	12.917

At low Rayleigh numbers the temperature profile was found to be virtually linear, indicating virtual suppression of convection, as

Figure. 12
Sketch Showing Geometry Used for Theoretical Studies.

anticipated. At higher Rayleigh number the temperature profiles began to depart from linearity, indicating a convective heat transfer. The local heat flux was estimated from the local temperature gradient, determined from the temperature at the wall and the next two grid points into the fluid. The local heat flux was then integrated along the vertical wall to give the average Nusselt number.

Figure 13 shows the Nusselt number calculated as a function of the Rayleigh number, based on a purely two-dimensional flow field. The plots are compared with experimental results on square celled honeycombs for $\theta = 90^\circ$, reported earlier. The agreement is seen to be not completely satisfactory, differences of the order of 40% being observed. However, by and large the agreement on the Rayleigh number at which the Nusselt number begins to depart significantly from unity is fairly good. There exists one difference between the computer model and the experiment that could explain the difference. This is the boundary conditions of the honeycomb walls. That used in the computer program is a linear temperature rise, that demanded in the actual honeycomb is the "repetitive" class of boundary condition in which heat transfer between adjacent cells is permitted, but in such a manner that the heat gained at one point on one face exactly corresponds to the heat lost at the point on the other face immediately below it. Since this more realistic boundary condition would seem to give less heat transfer, this difference in boundary condition could explain the observed discrepancy.

It should be noted that predicted heat transfer rates are based only on base flow, and do not allow for instabilities. From this it

Figure 13

Nusselt-number-Rayleigh number Relation Predicted Theoretically by Numerical Modelling of Fluid Flow. Also shown is Corresponding Measured Data.

would seem that the heat transfer measured should also be due to base flow, and therefore that the base flow is contributing very significantly to the heat transfer, in contradiction to a premise of the proposal. Clearly this contribution of the base flow will be lowered at angles off-vertical (i.e. $\theta < 90^\circ$). However it now seems very likely that it is contributing at these angles, probably down to 45° or less. More computer runs at angles less than 90° are necessary in order to check this out.

Techniques have also been developed to determine the stability of the base flow. This technique has been applied to the horizontal cell, and its results compared to established analytical results for this geometry; the comparison was found to be perfectly satisfactory. This technique has not yet been applied to the virtual or inclined cell base flow.

Study 2: Natural Convection in Inclined Air Layers

As reported in the semi-annual progress report, this study has been completed for angles θ , measured from horizontal, for the range $0 \leq \theta < 75^\circ$. The results were reported at the ASME-AICHE National Heat Transfer Conference in San Francisco, August 1975. A copy of this paper is included in Appendix B. The recommended correlation equation is:

$$\begin{aligned} Nu = 1 + 1.44 \left[1 - \frac{1708}{Ra \cos \theta} \right]^{\bullet} \left(1 - \frac{1708 (\sin 1.8\theta)}{Ra \cos \theta} \right)^{1.6} \\ + \left[\left(\frac{Ra \cos \theta}{5830} \right)^{1/3} - 1 \right]^{\bullet} \end{aligned} \quad (1)$$

where square brackets like so: $[ ]^{\bullet}$ indicate that if the quantity inside the brackets is negative, the quantity is to be taken as zero, i.e:

$$[x]^{\bullet} = (|x| + x)/2$$

where x is any quantity.

Consideration has been given to recommendations for the optimum spacing between the absorber plate and glass cover (or between glass covers), based upon this correlation equation. A report based upon these considerations is given in Appendix B. The findings are summarized here. Figure 14 shows the recommended correlation equation plotted in dimensional terms. It shows the expected free convective heat transfer coefficient, h , as a function of width of air gap, L , for various angles of inclination, θ . The plot is based on a mean air temperature, T , of 300K and a mean temperature difference, ΔT , of 50K. Correction may be made for other values of T and ΔT by modifying L and h with functions ϕ_1 , ϕ_2 and ϕ_3 , defined in the Appendix. The general shape of these plots had been pointed out earlier [3]. Note that there exists a local minimum in these plots, but this local minimum is not a global minimum, the least heat transfer occurring when the spacing is infinite, - an obviously unacceptable option. The best spacing would seem therefore to have to include consideration of other factors than purely free convective heat transfer, - for example the problems of shading of plate by the collector sides if too large a depth is used, the cost of deeper collectors, - both material, and package-and-handling costs. It seems to us, however, that a design at the local minimum may have considerable merit, since it gives good insulative properties of the air gap, but allows a compact and non-shading design. The actual decision would have to be made by the manufacturers, based on a total system study.

Figure 14

Dimensional plot of equation (1) show free convective heat transfer coefficient across an inclined air gap, as a function of spacing and angle of inclination. h_{REF} and L_{REF} refer to air at a mean temperature of 300K and a temperature difference of 50K. h for other conditions is obtained by modifying h_{REF} and L_{REF} by Φ_1 , Φ_2 and Φ_3 (See appendix).

The angle range $0 < \theta < 75^\circ$ appears to cover the vast majority of solar panel installations in the U.S. However, it is still considered that the range $75 < \theta < 90^\circ$ may be of importance in some cases and we plan to develop recommended correlation for this situation during the term of the extension to this contract. The difficulty experienced in this case is that we have found the Nusselt number to be very aspect ratio dependent in this case, - that is, the Nusselt number depends not only on the Rayleigh number and the angle of inclination, but also on the ratio of the height of the air gap H , measured along the plate, to the width of the air gap, measured normal to the plate, L , i.e. to H/L . In the range $0 \leq \theta < \sim 70^\circ$, the Nusselt number is insensitive to this ratio, provided it is greater than about 10, which is always the case in conventional collectors. However, in the vertical case there is a substantial dependence. The result of this is that: (i) the development of a correlation equation requires considerably more effort and knowledge and (ii) by its very nature, because it measures heat transfer only over the central region of the air layer, the University of Waterloo Natural Convection Apparatus is incapable of measuring the proper Nusselt number of interest, at least not without significant modification. Consideration is being given to these modifications. A correlation equation for the situation which correlates the data of earlier studies very well, and should be very suitable for solar work, has been obtained. However, it is also hoped to substantiate this correlation from measurements on the Apparatus.

Study 3: The V-Corrugated Inner Cover

The principle of the V-corrugated inner cover is sketched in Figure 15. In a standard inner cover, solar rays reflected off the surface are reflected away from the absorber plate and hence are lost. In the V-corrugated inner cover, reflected rays are reflected back toward the absorber plate and are therefore not lost. Thus the V-corrugated inner cover can be expected to have a much higher transmittance to solar rays than a standard cover. (The V-layer can be fabricated from glass or plastic films; the principle should work in both cases). The main commitment of this contract was to establish whether in fact the above mentioned principle is in fact valid. This was to be done both theoretically and experimentally.

A computer program has been written to theoretically model the radiant interaction of a solar ray with the V-layer. Using the Monte Carlo method, the program traces each of a large number of solar rays, of some arbitrary incidence angle, until it is ultimately (i) absorbed on the absorber plate, (ii) absorbed in the V-layer or (iii) reflected out from the V-layer. The path followed by a particular ray is complex and usually covers a number of interactions with the layer walls. (The number of possible paths is much greater than in the case of a V-corrugated absorber plate). Account must be taken of the: (i) reflection at the walls, for a photon incident on the wall from both outside, and inside; (ii) absorption of the photon by the walls when it is inside the walls. The reflection process can involve very high angles of incidence and it is therefore very important to use the full Fresnel equations in evaluating the reflection interaction.

STANDARD INNER COVER

V - CORRUGATED INNER COVER

Figure 15
Sketch Indicating why Increased Transmittance is Expected for a
V-corrugated Inner Cover.

Figure 16
 Plot showing Increased Transmittance Predicted for a V-layer using Monte Carlo Radiative Analysis. Angles β and θ area defined in Figure 17.

Figure 17

Sketch defining angles used in Figures 16, 18 and 19.

Specular reflection is assumed for each reflection. Roughly 10^4 photons are required as a sample in order to obtain reasonable accuracy on the expected transmittance.

Assuming that no scattering occurs, the overall transmittance of the V-layer depends on only two parameters the index of refraction of the V-layer material, n , and the product of the extinction coefficient of the material k , and thickness t of the wall, t , i.e. kt . As an order of magnitude, the index of refraction of glass and most plastics is approximately 1.5, giving a reflectance for normal incidence of approximately 4%, at a single interface.

Clearly as the product kt increases the advantage of the V-layer decreases, since, on average a photon travels a greater distance inside the wall material, for a V-layer than for a flat sheet. Therefore it is of interest to assess the performance of a V-layer assuming $kt = 0$. If in this case the transmittance is not greater than that of the flat sheet, then there seems to be no point in pursuing the topic any further. Figure 16 shows the predicted result, assuming a value for n of 1.5 for both a 45° degree opening angle V-layer, and a flat sheet. The angles β and θ for these graphs are the polar and azimuth angles of incidence, respectively, defined by Figure 17. Clearly substantially greater transmittances can be achieved by V-corrugating the inner, as compared with the flat sheet, and the V-layer principle is established, at least theoretically.

The question now arises as to whether it is possible to find a film for making the V-layer which would permit the gains shown in

these Figures, and at the same time, be practical for use in solar collectors. Such a film must:

- (i) have a low extinction coefficient and be available in thin sheets, so that kt is small;
- (ii) have good surface finish so as not to scatter significantly the solar beam;
- (iii) be stable to UV;
- (iv) have high temperature stability;
- (v) be reasonably capable of holding its shape;
- (vi) be inexpensive

Although we have not done an exhaustive survey of all materials, we find that teflon film shows the best promise. It would seem to satisfy very well the first 5 conditions listed above, and it appears to be marginal in cost effectiveness. Teflon was therefore singled out for the experimental tests on transmittance. Based on data supplied by the manufacturer, its index of refraction is estimated as 1.344 and its solar-incident-mean extinction coefficient to be 3.43 inch^{-1} or $.135 \text{ mm}^{-1}$. It is available down to thicknesses of .0005 inch; however the .001 inch thick film is much easier to work with. Figure 18 and 19 show the predicted transmittance of V-layers prepared from this film at both 45 and 30 degrees opening angles. Also shown are the expected curves for flat sheet of this material. The gain of V-corrugating is still clearly demonstrated, although the gain in this case is not nearly as pronounced, since the index of refraction of teflon is relatively low, giving a low reflectance.

The apparatus to test these ideas experimentally is now constructed and is undergoing break-in testing. Shown in sketch in Figure 20.

Figure 18
 Predicted transmittance of a teflon V-layer, for opening angle of 45°.

Figure 19
 Predicted transmittance of teflon V layer for opening angle of 30°.

and in photos in Figures 21 and 22, it contains two solar radiometers, in virtually identical situations, the one to be used as a reference, the other to have the V-layer placed in front of it. Figure 22 shows it with a flat teflon layer across it. The apparatus would ordinarily use the sun as the source, and four angular adjustments on its gimbelled supports permit the easy setting of both of the angles of incidence, β and θ , with respect to the sun. The transmission measurement would include diffuse sky radiation as well as direct-beam so that best results should be obtained on a clear day.

The apparatus can be used in either of two ways: in the first way the non-reference radiometer reading is taken first with and then without the V-layer present, and the ratio of these readings gives the transmittance; in this case the reference radiometer is used only to establish whether the solar radiation did not change over the test period. In the second way, the radiometer with the largest calibration constant is coupled to a voltage divider adjusted so that the overall calibration constant is identical to that of the other radiometer. Then the non-reference radiometer has the V-layer placed over it and the ratio of the instantaneous readings of the radiometers should give the solar transmittance. At the time of writing, measurements of the transmittance of a V-layer are not available.

In summary, theoretical calculations have shown the V-layer principle to be sound. Experiments to test it should be completed shortly.

Figure 20
Sketch of Apparatus to be used for Measuring Transmittance of a V-layer.

Figure 21
Photo of Apparatus to be used for Measuring Transmittance of a
V-layer.

Figure 22
Photo of Apparatus for measuring transmittance of V layer, showing
teflon film in place.

RESEARCH UTILIZATION

Dr. Hollands, Dr. Unny and Dr. Raithby all attended the NSF/RANN Solar Collector Workshop in November. Dr. Hollands made a presentation, outlining the work to be carried out under this proposal. Dr. Hollands also participated in the Collector materials workshop meeting.

Dr. Hollands presented a paper (co-authored by R.L.D. Cane, G.D. Raithby and T.E. Unny) at the International Solar Energy Society meeting in Los Angeles in July-August, 1975. This paper described the honeycomb free convective measurements described in this report. An abstract of the paper is given in Appendix B. A very large number (approximately 30) requests have been received for this paper, indicating a wide interest.

Dr. Hollands also presented a paper at the ASME-AICHE National Heat Transfer Conference in San Francisco, in August, 1975, presenting the "free convective heat transfer in inclined air layers" paper discussed in this report (equation (1)). A very large number of requests have also been received for this paper, and copies have been sent to these requestors. The paper has been submitted for publication in the Journal of Heat Transfer.

Dr. Hollands visited Mr. K.N. Marshall and other research personnel at the Lockheed Missiles and Space Company Inc., in Palo Alto, and exchanged ideas in a large number of research topics, mostly to do with the use of honeycombs in flat plate collectors.

Dr. John Howell and Dr. Richard Bannerot of the University of

Houston visited the Solar Energy Group at the University of Waterloo in May of 1975. Another visitor was Mr. R.V. Dunkle, Chief Research Scientist at the CSIRO Division of Mechanical Engineering, Melbourne Australia.

RELATED PROGRAMS

Baldwin, Dunn, Hilliard and Mackenzie, at University of California, Los Angeles, are carrying out studies on methods of fabricating honeycombs from glass. Edwards, Catton and Buchberg at the same university are examining the effect of the clearance distance between the honeycomb and the glass cover and between the honeycomb and the absorber plate, as well as overall heat transfer across glass honeycombs. Most of their work appears to relate to low angles of tilt from horizontal (up to about 30°). K.N. Marshall at Lockheed Aircraft is developing and assessing solar collectors using plastic honeycombs.

To my knowledge, no other investigator is measuring free convective heat transfer rates across inclined air layers, or examining the V-corrugated inner cover.

CONCLUSIONS

1. Honeycombs can suppress free convective heat transfer in solar collectors, even in the inclined position. The charts presented in the report should permit the honeycomb-collector designer to choose the aspect ratio of honeycomb so as to ensure convection suppression, provided the aspect ratio does not exceed 5. If the charts err, they should err on the conservative side.
2. Equation (1) may be used for predicting the free convective heat transfer across inclined air layers, such as occur in solar collectors, with an expected error of less than 3%, provided the angle from the horizontal is less than or equal to 60° . Consideration of the "optimum" air gap spacing to be used in solar collectors is a systems problem as well as a free convective one, since the purely free convective "optimum" is an infinite spacing.
3. Theoretical calculations show that V-corrugating an inner cover, should, under certain circumstances, lead to increased transmittance. A thin teflon sheet shows the best promise in this regard.

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APPENDICES

APPENDIX A

LIST OF RESEARCH CONTRIBUTORS

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Consultants:

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APPENDIX B

OTHER REPORTS
AND PAPERS GENERATED

Free Convective Heat Transfer Across Inclined
Air Layers as Applied in Flat Plate
Solar Collectors

by

J.F. Orgill

and

K.G.T. Hollands

Thermal Engineering Group
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Introduction

With renewed interest in the use of solar energy as a viable alternate energy source for space heating and cooling, considerable research is underway attempting to reduce heat losses, and consequently increase the efficiency, of flat plate solar collectors. Much of this research is concerned with "advanced" forms of the flat plate collector utilizing selective surfaces to reduce radiation losses and cellular structures referred to as "honeycombs" to reduce free convective losses. Recently, however, Buchberg et al [1] pointed out the need to consider the gap conductance between the hot absorber plate and the cover (for a two or three cover collector a similar situation exists between adjacent covers). Tabor [2] has also commented on the effect of gap distance.

Basic Theory

Tabor [2] when commenting on the effect of gap distance in a solar collector based his work largely on a U.S. Government Research Paper [3] which reports extensive measurements of the heat transfer across an inclined air layer, but all the measurements were based on an inclined angle of 45° above the horizontal.

Buchberg et al [1] when reviewing the gap conduction calculation relies extensively on Clever's proposal [4] that the heat transfer across a tilted air layer can be calculated from the results for the horizontal layer (Benard problem) replacing g by $g \cos \tau$ in the Rayleigh Number, where τ is used to represent the angle the air layer is inclined above the horizontal. Clever, in presenting his experimental verification used a few data points published by deGraaf and Van der Held,

which in the region of interest for $1708 < Ra < 10^4$ and for inclined angles of greater than 30° fall well below the recommended curve.

Recently Hollands et al [5] have presented new experimental measurements on free convective heat transfer rates through inclined air layers of high aspect ratio, heated from below. Correlation of the data produced the recommended relation for predicting the Nusselt number as:

$$Nu = 1 + 1.44 \left[1 - \frac{1708}{Ra \cos \tau} \right]^{\bullet} \left(1 - \frac{(\sin 1.8\tau)^{1.6} \cdot 1708}{Ra \cos \tau} \right) + \left[\frac{Ra \cos \tau}{5830} \right]^{1/3} - 1 \right]^{\bullet} \quad \tau \leq 60^\circ \quad (1)$$

where $[X]^{\bullet} = (|X| + X)/2$, X being any quantity. For $0 \leq \tau \leq 60^\circ$ and $0 < Ra < 10^5$ the equations gives values of Nu with a maximum error of 5%. The equation (1) may be used for τ up to 70° but errors of up to 10% may be expected.

Based on this new correlation the values of h' (gap conductance in $W/m^2-^\circ C$) were calculated as a function of L' (the gap spacing in cm) for various angles based on a temperature of $300^\circ K$. Figure 1 shows the results of these calculations. Correction factors $\phi_1(T)$, $\phi_2(\Delta T)$, $\phi_3(T)$ were then calculated as indicated in the appendix so that the plots of h' vs L' could be used to determine h for any L under any combination of mean air temperature and temperature difference. The values for $\phi_1(T)$, $\phi_2(\Delta T)$, $\phi_3(T)$ are indicated in Figure 2.

Figure 3 compares the results obtained by Buchberg [1] with the new results for the specific case where the mean air temperature is $84^\circ C$, a temperature difference of $28^\circ C$ and inclination angle of 40° .

As can be seen from Figure 3 the dependence of the gap conductance

Figure 1

Dimensional plot of equation (1) show free convective heat transfer coefficient across an inclined air gap, as a function of spacing and angle of inclination. h_{REF} and L_{REF} refer to air at a mean temperature of 300K and a temperature difference of 50K. h for other conditions is obtained by modifying h_{REF} and L_{REF} by ϕ_1 , ϕ_2 and ϕ_3 (See appendix).

FIGURE 2

CORRECTION FACTORS

$$\phi_1(\bar{T}), \phi_2(\Delta T), \phi_3(\bar{T})$$

$\left. \begin{array}{l} \text{Mean temperature between boundaries, } T[^\circ\text{C}] \\ \text{or Temperature difference between boundaries, } \Delta T[^\circ\text{C}] \end{array} \right\} \longrightarrow$
 Note: T to be used to evaluate ϕ_1 and ϕ_3
 ΔT to be used to evaluate ϕ_2

FIGURE 3

(h) on the gap spacing (L) is much weaker than previously predicted.

Collector Design Considerations

Based on the information in Figure 1 & 2 and a knowledge of the cost of increasing the collector housing size, a designer could then select the most optimum gap spacing for a range of operating conditions. The following example should aid in explaining the above.

A flat plate solar collector is to be used in Toronto, Ontario for collecting solar energy for space heating. The collector tilt angle has been optimized at 60° above the horizontal. The collector is to have a single glass cover. The inlet fluid temperature will be 51.7°C (125°F). By examining temperature records for Toronto, the lowest recorded temperature was -32°C (-27°F), the mean maximum recorded temperature during the period October to May was 17.2°C (63°F) and the average mean recorded temperature for the months of January and February was -9°C (23°F). Assuming the hot plate temperature to be equal to the inlet water temperature and the glass cover temperature to equal the ambient temperature three curves for h vs L can be plotted as shown in Figure 4. The bounds on the gap conductance are set by the curves for $T_{\text{amb}} = -32^{\circ}\text{C}$ and $T_{\text{amb}} = 17.2^{\circ}\text{C}$. The curve for $T_{\text{amb}} = -9^{\circ}\text{C}$ is the most probable operating condition for the high heating demand months of January and February. For this particular case the designer might choose a gap spacing of 1 cm producing a relative minimum of h for the most probable operating condition during January and February. For temperatures less than -9°C the gap conductance will not be a relative minimum although considering the worst case for $T_{\text{amb}} = -32^{\circ}\text{C}$ the conductance only increases from a relative minimum of ~ 3.3 to 3.75

FIGURE 4

W/m^2 $^{\circ}C$ for a spacing of 1 cm. For higher ambient temperatures the gap conductance will not be a relative minimum although for higher ambient temperatures the efficiency of the collector becomes increasingly less important because the heating demand is lower.

The alternative for the designer would be to choose a gap spacing of 4cm or greater which would give the same gap conductance as 1 cm for an ambient temperature of $-9^{\circ}C$ and a lower value for the conductance for all other ambient temperatures when compared to 1 cm. The consideration the designer must apply, however, is that the larger gap conductance will increase the cost of the collector housing, packaging and handling, as well as provide some shading of the plate.

Conclusions

1. Gap conductance is not as strongly dependent on the gap spacing as previously predicted.
2. When optimizing the gap size, consideration must be given to the gap conductance, operating conditions, and cost of the collector housing as a function of gap spacing.

Appendix

The basis for choosing the abscissa and ordinates for figure 1 is the Nu-Ra relation where

$$Nu = qL/\Delta Tk = hL/k$$

$$\text{and } Ra = g\beta_c \Delta T L^3 / k\mu_p$$

$$Nu = f(Ra, \tau) = hL/k$$

$$\begin{aligned} \text{Nu}' &= \text{Nu}/\text{Ra}^{1/3} = (h/k) / \left(\frac{g \rho^2 \beta c_p \Delta T}{k} \right)^{1/3} = h/kf(T) \cdot \Delta T^{1/3} \\ &= h/f_1(T) \cdot \Delta T^{1/3} \text{ where } f_1(T) = k \cdot f(T) \end{aligned}$$

For a reference temperature (T_r)

$$f_1(T) = f_1(T_r) \cdot \left(\frac{f_1(T)}{f_1(T_r)} \right) = f_1(T_r) \cdot \phi_1(T)$$

$$= \text{constant} \cdot \phi_1(T) = c_1 \cdot \phi_1(T)$$

$$\text{where } \phi_1(T) = f_1(T)/f_1(T_r)$$

Therefore;

$$\text{Nu}' = h/c_1 \cdot \phi_1(T) \cdot \Delta T^{1/3}$$

Similarly for a reference temperature difference between the air layer boundaries ΔT_r

$$\Delta T^{1/3} = c_2 \cdot \phi_2(\Delta T)$$

Combining these results and rearranging yields:

$$\text{Nu}' \cdot c_1 \cdot c_2 = h/\phi_1(T) \cdot \phi_2(\Delta T) = h'$$

where $h' = f(L, \tau)$ only.

A similar exercise is performed with the Rayleigh number:

$$(\text{Ra})^{1/3} = \left(\frac{g \rho^2 \beta c_p}{k \mu} \right)^{1/3} \cdot \Delta T^{1/3} \cdot L$$

$$(\text{Ra})^{1/3} = c_2 \cdot \phi_2(\Delta T) \cdot c_3 \phi_3(T) \cdot L$$

$$\text{Ra}^{1/3}/c_2 \cdot c_3 = \phi_2(\Delta T) \cdot \phi_3(T) \cdot L = L'$$

where L' is no longer a function of T or ΔT .

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