THE POROUS ABSORBER

for

SOLAR AIR HEATERS

MASTER

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Efficient Forced Air Solar Collector"

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Introduction

The project of which this discussion is the final report was a conceptual and experimental evaluation of a solar air heater design based on a porous absorber geometry with air passed through concurrently with the "in depth" absorption of solar radiation. The basic absorber consisted of a stack of fine black dyed sheer fabric screens. A superior collector efficiency results, due to the in-depth radiation absorption, the good heat transfer characteristics of the absorber, and the partial recapture of infrared radiation due to the temperature stratification in the screen stack.

The porous, throughflowing absorber is not widely discussed in the literature nor, to the best of the author's knowledge, is it employed in any commercial collector designs. It was independently conceived by the author and, due to the paucity of literature references, was initially considered to be original. Later, I discovered that the design has been known, in certain forms, for some time. It appears to have been initially conceived by Donovan and Bliss (1), who reported on its high efficiency as early as 1955. The concept was later discussed by Whillier (2) and by Chiou, El-Wakil, and Duffie (3).

The lack of commercial use of the porous, throughflowing, temperature stratified absorber in solar air heaters is unfortunate. Collector tests on the simplest version of the design indicate an average energy capture efficiency, at a usable temperature rise, of better than 45% under good conditions. This compares favorably with the 30% to 35% efficiencies typical of commercial flat plate solar air heaters. Perhaps the lack of commercial acceptance is due to reasons of cost and of appropriate materials availability. Experiments showed that very finely woven fabrics were most suitable, and fabric materials probably do not have the requisite lifetime. Low cost, finely woven fiberglass screens do not appear to be widely available, and metal cloth is excessively expensive. Assembly procedure, using separate fabric layers, can also be expensive.

As a result of the author's tests and design concept development, a novel and original type of porous, throughflowing absorber is proposed. Of plastic laminate construction, the design should be of relatively low cost. Unitized absorber construction should facilitate collector final assembly. Most importantly, the novel use of optically transparent, infrared opaque elements in the coolant flow path can lead to a very efficient collector. Tests on a crude experimental version showed energy capture efficiencies of generally better than 50% at air temperature

increases of up to 60 °F on passage through a single collector module. The design appears at this point to have considerable commercial potential.

The following discussion is organized into four main sections. A simple general discussion of the factors affecting solar collector performance is presented. This is followed by an explanation of bench scale tests done to try to determine the heat transfer characteristics of various screen materials. The design, performance, and evaluation of a "crude" collector, with a simple screen stack absorber, is then treated. The more sophisticated absorber concept, and its first crude experimental approximation, is then examined. The discussion concludes with a short summary of future plans for the collector concept.

General Considerations

Many factors affect the performance of a flatplate solar collector design, and both internal and external variables must be taken into account. The internal variables are under design control. External variables are climate and local weather patterns. A given collector will not perform with equal efficiency under all conditions. The ambient temperature, sky temperature, and insolation level are all of great importance. In general, collector efficiency will increase with increasing insolation, as losses are approximately a constant of operation. Lower ambient temperatures, of course, decrease efficiency.

Although detailed analysis of collector performance is complicated, a number of highly accurate models, amenable to computer methods, have been developed. These models do not extend to the evaluation of the type of system under present discussion; they are largely limited to near-uniform absorber temperature and ideally free convective conditions within the system, and are thus applied largely to liquid cooled collectors. Liquid cooled systems generally have very good heat transfer which, coupled with the small rise in coolant temperature on passage through the absorber, allows an approximation based on an homogeneous absorber temperature to be quite accurate. A typical "efficiency factor" based on local temperature field effects in such an absorber is around 0.98 or 0.99, which is unity for all practical purposes.

Air-cooled absorbers do not appear to have been as well analyzed, in the open literature, as have been liquid cooled designs. This may be because flat plate designs have been historically more oriented towards water heating. The preference for liquid cooled systems as objects of analytical model building may well also

reflect the greater ease of their analysis. As opposed to liquid systems, air cooled collectors generally heat the working fluid through a fairly wide temperature range, generally from about 60°F to 120° to 140°F . The performance analysis is further complicated by the standard practice of ducting several air heating collectors in series, which gives them different operating temperatures and therefore different efficiencies.

Analysis of flat plate air heater performance is thus not simple, and the type of efficiency charts available from manufacturers do not necessarily provide a realistic guideline. Manufacturers will, however, provide candid overall efficiency estimates for operation under typical conditions; generally these range around 30% to 35%. Water heating collectors are generally more efficient, around 45%. This is due both to the smaller temperature difference between fluid and plate encountered in the liquid cooled designs and, also very importantly, to the year round operation of water heating solar systems. All collectors operate at greatest efficiency under high insolation.

Losses through the cover glazing account for the great majority of the inefficiency of most solar collector designs. The back and sides of most commercial designs are well enough insulated to make losses from these surfaces of relatively minor importance, at least in the case of the better products on the market. Cover plate losses are caused by a complex interaction of conductive, convective, and radiative mechanisms. Sophisticated analytical methods, based on thermal circuit network analysis, have been devised to separate these contributing factors, but these methods will not be reviewed in this discussion. Rather, some of the loss mechanisms will be qualitatively discussed from the point of view of a comparison between the porous absorber and conventional designs.

The diagram below illustrates an idealized porous absorber. All stages are

considered to be perfect blackbody radiators, and therefore absorbers, in the infrared. Optical configuration can vary; the absorber can either employ optically transparent elements and optically dense elements, or it can employ optically dense elements only, with the degree of transmittance of each element being under design control. The latter is, of course, the only real option. Assuming perfectly transparent elements possible, and infinite heat transfer coefficients, it is clear that the arrangement of an infinite number of optically transparent elements in front of a single optically black element can give perfect absorption with no radiation loss, assuming the inlet and background temperature to both be at ambient. Furthermore, any desired outlet temperature can be achieved, short of temperatures approaching the optical region of the blackbody spectrum.

Real porous absorbers, of course, suffer from various limitations, and an evaluation of these limitations forms the major important aspect of the present work. In particular, the screen stack suffers from an inability to dissociate optical and infrared properties, due to the opaque material of screen manufacture. This disadvantage will be shown to be responsible for a considerable inferiority in screen stack performance to that of an ideal absorber, resulting in a collector less efficient than is possible. The remedy, placement of one or more transparent porous heat transfer elements in front of the optically dense elements, will be shown to greatly improve performance.

The cover glazing of a flat plate collector is very important in returning radiation, and in conserving energy within the system by a straightforward greenhouse effect. A simple heuristic example shows this. Assuming an optical radiation flux W incident through one or two transparent cover glazings, and a background temperature of 0°K , the absorber plate is assumed operating at a temperature where its radiant output is W in watts/meter². The idealized effects of single and double glazings

are illustrated. It is easy to show, in this idealized example, that a fraction $n/n+1$ of the infrared reradiation can be returned by n glazings. This return constitutes the energy available to the cooling medium.

While the heuristic example has little to do with real collectors, several important points are highlighted. Double glazing is preferable for radiative, as well as convective, loss minimization. Most importantly, a glazing material highly opaque in the thermal infrared spectrum is to be greatly preferred. This has been one of the traditional advantages of glass over plastic glazing materials such as Tedlar. The infrared opacity is particularly important at the inner glazing in a double glazed collector, which is where plastic materials have generally been employed. The author does not have data, but it would seem that the .001 inch thick Teflon film proposed as an inner glazing would, on the basis of thinness alone, lead to serious radiation loss. For example, an absorber operating at 90°C (363°K) has a blackbody radiative output of 985 Watt/meter^2 , assuming emissivity of unity, from the Stephan-Boltzmann equation;

$$W = 5.67 \times 10^{-8} \times T^4 \quad \text{Watt/meter}^2 - ^{\circ}\text{K}^4 .$$

Assuming an entirely transparent inner glazing (in the thermal infrared) the outer glazing then, in the simplest example, must reradiate 493 Watt/meter^2 , at a blackbody temperature of 305°K , or 27°C . The actual losses due to both radiation and convection would, in such a case, be severe. It can be assumed, in most cases, that the background temperature averaged over the collector field of view is no greater than ambient, and in many cases considerably less. A winter ambient of -5°C would contribute less than 300 Watt/meter^2 of background radiation, making the actual radiative loss from a collector with an infrared transparent inner glazing quite serious.

Actual collector operation is quite complex, as convective effects augment radiative transfer as loss producing mechanisms. Although plastic inner glazings are probably not anything close to 100% transparent in the thermal infrared, glass has nonetheless almost always been the choice for inner, as well as outer, glazings in most top quality commercial solar collector designs, showing the importance of this loss mechanism.

Perhaps most important in the case of a number of reputable designs of solar air heater is an air flow path which transfers heat from the absorber to the inner cover glazing by forced as well as natural convection. Three conventional flat plate solar air heater designs are illustrated below. Designs one and two differ

only in the inclusion in the latter of an extended heat transfer surface, which could be fins, a slit-and-expanded metal structure, a wire wool, or some other material. In both design one and design two, forced convection will transfer heat from the absorber to the inner glazing, thus short-circuiting, to one extent or another, the full radiative temperature field of double glazing. In design three, natural convection will perform the same function to a lesser extent. Design one is not very good. Poor heat transfer between absorber and air force the absorber temperature to be high, which compounds losses. Design two is quite intelligent, if the heat transfer ability of the porous crossflowing system is sufficient to keep the mean absorber temperature within a reasonable distance from the local bulk mean gas temperature. The disadvantage, however, is that the losses from the hot section of a collector series will reduce overall efficiency.

Design three can be quite good, and is a natural candidate for loss suppression by means of either a honeycomb structure between the absorber and the inner glazing, or a selectively absorbent coating on the absorber. The disadvantages with both of these options appear to be cost and, in the case of the honeycomb, a significant deterioration in performance at higher incidence angles. For a number of reasons; economic, energetic, and practical; the use of honeycombs and/or selective surfaces has not gained commercial acceptance in air heaters, although the technology has

been available for many years. Present state-of-the-art in commercial solar air heating systems is concentrated on designs two and three. They appear to be about equally efficient, with manufacturers claiming in the neighborhood of 30% to 35% energy capture for transferral to the air stream.

Application of an ideal porous absorber to a double-glazed solar collector, as schematically illustrated below, could result in a dramatic reduction in both radiative and convective heat transfer losses when compared to conventional designs.

In practice, a "perfect" porous absorber is not possible. It is, however, possible to approach this ideal quite closely. The improved collector design tested, after the relatively inadequate performance of the screen stack was noticed, showed that theoretically optimum performance can be reasonably approached in practice, with potentially simple and low cost hardware. Consideration of ideal porous absorbers is therefore of much more than academic interest.

It is beyond the scope of this work, and also not possible in a general case discussion, to do a detailed analysis of the losses incurred by the solar heater design illustrated. Many design variables and site peculiarities are independent, and would require detailed specific knowledge. Some general points concerning the design can be made. These include:

The top absorber element should ideally be no hotter than the air inlet temperature. A higher element temperature implies a loss of absorber efficiency. In a typical home heating application, this temperature will generally be around 60° F for air ducted from a cold air return. In an agricultural or industrial application, collector inlet would be at the outdoor ambient temperature. Losses in this latter case would be largely due to cover glazing opacity. A single glazing for absorber protection only could be used.

The inner cover glazing of a double glazed collector will operate at a temperature lower than the inlet air, but higher than outdoor ambient. The outer glazing temperature will depend on both ground ambient and on background radiation temperature. If this is a true sky temperature, it could be as much as 20° C lower than outdoor ambient. The outer glazing under such conditions could operate at, in theory, less than outdoor ambient temperature. In practice, the field of view of a collector surface is likely to include a good deal of ground, and in many cases heated structures as well. Background radiation in such cases will be well above the true sky temperature, and can approach the outdoor ambient. In such cases, collector losses will be very low.

Series staging of porous absorber collectors is impractical. A high inlet temperature will cancel the efficiency potential of the design. Similar reasoning shows an attempt to design a porous liquid heater to be of no value. The collector depends for its efficiency on a minimal difference between inlet temperature and outdoor ambient, and is thus of use primarily as a residential space heater, and perhaps in agricultural and industrial drying systems.

Collectors will be more efficient at high insolation levels, due to the relatively constant losses and the increased gross energy capture. All solar

collectors exhibit this dependence of efficiency on insolation. For this reason, quoted instantaneous efficiencies based on various desert tests, in which insolation values are typically in excess of $300 \text{ BTU/hr-foot}^2$, show better performance than is achieved in application. ASHRAE 93-77 should be applied on site.

Internal manifolding can be accomplished in a collector array by the use of large ducting passages such as those illustrated. Air flow can be as illustrated below. The takeoff of air from the opposite end of the parallel array from its introduction helps assure a balanced pressure drop and air flow across each module in the assembly. The design is similar to, but simpler than, the internal manifolding in the Soloron air heating system.

Porous absorbers of near-ideal performance can be practically produced by methods to be discussed below. As with any design in the solar energy field, cost will be a factor of major importance, and can be legitimately included with efficiency as a general design consideration of the first importance. A collector used as a space heater in winter only is particularly susceptible to a cost disadvantage, as its use is largely confined to winter months, and it largely sits idle during summer. The installed cost, in dollars per square foot, should thus be low. A solar space heating system, in a location such as Madison, Wisconsin, will typically provide, in a design of average efficiency, approximately 1 therm per heating season per square foot of collector area. This one therm translates into about \$1.00 if oil is used as the conventional heating fuel, about \$1.40 if electricity is used, and about \$0.50 if low cost natural gas is used. The latter figure, in the author's opinion, will soon be a thing of the past. Deregulation of natural gas prices is likely, within ten years, to bring the cost of gas to a

level competitive with oil. Assuming a fossil fuel heating cost of \$1.00 per delivered therm as a reasonable average figure, a solar system must, to be at all worthwhile, pay back at least its cost in its lifetime, and must thus last at least as many years as its installed cost in dollars per square foot of collector aperture. Most systems can do this. A better economic picture, however, on the order of a 5 year maximum payback time for the homeowner, is needed to encourage solar system use. Increased fossil fuel prices will do this, in an unfortunate manner. Far better would be to improve the solar system output/cost ratio.

The installed costs of the highest quality solar air heaters, in particular the Solaron system and the Suncell system of Research Products, Inc. of Madison, Wisconsin are, in the author's opinion, so high that significant sales will never be realized. Local contractors asked about installed system costs quoted figures of between \$35 and \$45 per square foot for complete systems, and collector panel costs of \$15 to \$20 per square foot. At an annual payback rate of \$1 per square foot, these systems do not make economic sense. Tax credits, displacement of roofing in new construction, and enhanced home resale value will help the economics of these systems, which are very long-lived, but their market appeal will nonetheless be limited to a small, highly affluent minority.

To be fair, contractors estimate that between 30% and 50% of the cost of these systems is the installation cost. Ducting is a considerable expense, but it can be greatly reduced by internal manifolding. Another installation cost, which can become excessive in some cases, is the need for roof bracing to withstand the weight of the collector array. Both the Suncell and Solaron panels weigh over 7 lb per square foot, which is more than many roofs were designed to support.

At the other end of the spectrum of solar heaters are the low cost systems which are designed with do-it-yourself installation in mind. One example of such a system is Kalwall Corporation's Solar-Kal air heater, which employs a fiberglass panel body and a fiberglass/Teflon double glazing. This heater weighs only 2 lb per square foot, but does not include insulation. For hydronic systems, the SolarRoll heater is available. Exclusive of labor, these systems can be installed for less than \$10 per square foot.

The low cost systems are generally shorter lived, require more maintenance, and are less efficient in energy capture than the premium products. The Solaron series 3000 panel, for example, is quite efficient. Advertised peak efficiency (obtained under high airflow and high insolation) is 65%. Operating range efficiency is claimed between 32 and 43 per cent, for an average of 37.5%. This collector, however, employs a selective black chrome absorber and is very expensive.

The porous absorber concept is primarily intended for low cost systems, but it could be used to advantage in the high priced units as well. The high energy capture efficiency of the porous absorber, shown to approach 60% at usable flow rate and temperature increases, can likely more than double the useful heat output from a collector. This increase will be a combination of several effects. First is the increase in clear day, high insolation efficiency over most other designs. Also important is the design's ability to provide useful energy on days of low insolation. Most collectors can do this to some extent but the porous absorber, due to its excellent heat transfer ability, can probably excel. Tests of the improved absorber design on a uniformly hazy day showed useful temperature output, at reduced flow rates, at insolation levels of less than 350 watt/meter². Finally, collector transient behavior can be improved with the porous absorber. The thermal mass of the absorber can be made quite low. Most importantly, the low level of participation of the cover glazings in the thermal network will make for a fast transient response. This can facilitate good performance on partially overcast days, and near sunrise and sunset.

Installation methods, ducting, and heat storage methods for the proposed design can be quite versatile. Tests showed that the collector tilt angle made little difference in efficiency, so either sloped or vertical mounting, on a rooftop, auxiliary structure, or south facing vertical wall, could be used. Either latent or sensible heat storage could be used. Collector output temperatures of up to 140° F were achieved with good efficiency in the tests, and a more refined design could undoubtedly do better. This makes rock bin storage efficient. There is also some indication that the absorber design could be efficiently used in thermosiphoning air heaters.

Bench Scale Tests

Prior to the invention of the "improved" porous absorber design, the author felt that the woven wire or fabric screen stack was the best and most convenient porous absorber design. This design, however, is imperfect; the topmost absorber element will be optically active, and therefore must operate at above inlet ambient temperature. Infrared recapture from the deeper levels will, moreover, be less than complete, as the outer screens will have some infrared transmittance. Heat

transfer between a screen and the throughflowing air must be very good, for best performance. Poor heat transfer will raise all of the screen temperatures in the stack, and compound the infrared losses.

The first experimental phase of the project was an attempt to measure the heat transfer coefficients for various screen geometries, and decide on the basis of these results, and of optical properties, which would be the optimum for the absorber stack application. This work was successful in supplying data for a comparative evaluation, but was unable to achieve with full confidence the desired result of an accurate plot of Reynolds versus Stanton numbers for the low Re flow region. The major problems were to find a means to accurately measure the screen temperature and to design a device which could measure the airflow temperatures and screen temperature and at the same time hold artifact due to radiation and natural convection to a minimum. Methods to do both of these were developed over a period of months which eventually gave reasonably satisfactory performance, but the results are probably not accurate enough to be published in the literature. No attempt to publish, therefore, will be made.

The initial experimental attempts used electricity to heat a metal screen, through which air was passed. Thermometers upstream and downstream of the screen measured the air temperatures, and a thermistor placed on the screen measured its local temperature. It became evident, however, that a serious artifact was being introduced by the thermal conductivity of the screen. The convective heat transfer coefficient for a standard iron window screen, which was the first to be tested, was considerably lower than anticipated. In the 3 inch diameter flow aperture used, conductive transfer between the screen and its supports led to a very significant temperature field in the screen itself, of up to 30 Centigrade degrees difference between center and edge, as measured by attached thermistors. It was expected that convection would overshadow conduction, but this did not prove to be the case.

The author does not recommend attempts to electrically energize a screen for heat transfer test purposes. Even if a much larger test apparatus is used, the high amperage required will necessitate heavy bus bars being attached to the screen. Unless these bars operate at exactly the screen temperature, which is very unlikely, a significant experimental artifact will be introduced.

The attempt to use electricity was soon abandoned, and radiant energizing was resorted to in its stead. This was a fortunate choice. The author would recommend

its employment in any future studies of this type. Artifact due to conductive degradation of screen temperature is largely avoided, due to the very low thermal conductivity of the cotton, nylon, and fiberglass materials used in the tests. Radiant energy can be applied uniformly to any screen, regardless of its electrical conductivity. It makes for ease in the experimental setup; screens can be changed in a matter of minutes. The bus system associated with electrical energizing was, by contrast, cumbersome and difficult to work with.

The apparatus used to conduct the bench scale tests was developed over a period of several months. A number of designs were experimented with. The final design was as illustrated on the previous page. Its mode of operation is straightforward. Light from the heat lamp is somewhat concentrated by the crude cone-type unit, and is passed through a cooled plastic glazing whose purpose is to absorb the infrared radiation produced by the hot glass envelope of the heat lamp. The light is then passed through an aperture, and enters the device proper. The aperture is designed to cast the cone of illumination on the screen alone, and to avoid direct illumination of the chamber sidewalls to the greatest extent possible. Intensity of the illumination can be varied, by varying the distance of the lamp and aperture from the screen and from each other. Light not absorbed by the screen exits through the transparent window at the bottom of the device.

A measured amount of air entered the bottom chamber, passed through the device, and exited past a thermometer via the duct shown. Inlet air temperature was also measured by a thermometer. These thermometers were kept out of the illumination, and the bulbs were coated with reflective aluminum foil to minimize any possible radiative artifact. The air flow was supplied by a small blower which fed a small, precision orifice meter. The pressure drop across this orifice was adjusted to provide a flow velocity at the screen aperture of 2 feet per minute free stream, typical of solar heater application. One of the disadvantages of testing on the small scale employed, with a 3 inch diameter screen aperture, was the small size of the orifice, and consequent precision meter construction, and the low pressure used to provide a flow equivalent to the required air supply. An orifice diameter of 0.125 inch was used, which required countersinking of the plate to achieve the desired sharp edge, and a pressure drop of 0.12 inch H_2O was employed.

The screen temperature was measured by a calibrated thermistor, attached to the screen weave by contact cement. When using thermistors to measure the screen temperatures and, later, the glazing temperatures in both the bench scale and the full sized collector tests, it was found that a lower temperature reading would

invariably occur when the thermistor was protected from direct radiation by a small aluminum foil cap. If left unprotected, the thermistor apparently absorbs enough radiation to be significantly heated above the local screen or glazing temperature. There is some danger of an artifact being created by the foil cap itself, as it disturbs the local airflow and radiant energy flux, but a good case can be made that this artifact will be small. Aluminum is a good reflector in both the optical and the infrared, and also a very good heat conductor. The good reflectivity will keep the aluminum foil ~~spread~~ surrounding the thermistor from being an active heat source as either an absorber or radiator. This passivity, in association with the excellent thermal conductivity, should keep the foil temperature close to that of the surrounding screen or glazing. This will, of course, depend to some extent on the size of the foil caps, which were kept small for this reason. In any event, no other method within the scope and budget of the project exists to measure the screen temperature. An attached thermometer, or thermocouple pile, would be no better off than the thermistor with respect to artifact creation. The ideally best way to determine screen operating temperature, in a sophisticated test, would be by means of a 5° acceptance infrared pyrheLiometer aimed at the screen in such a way as to provide a view factor of unity within the acceptance angle, and to calibrate this system against screening material in a passive, nonconvective situation.

By means of the apertures and the chamber design, it was possible to almost entirely avoid direct illumination of the chamber sidewalls. These walls were, in any event, protected by aluminum foil reflectors, as a cautionary measure against both optical and infrared radiative energizing, the latter being due to the heated screen. Some absorption of radiation by the glazings used to seal the air flow chambers was, however, unavoidable. To compensate for this, the glazings were cooled by a fan. The effectiveness of this was tested by operating the device without a screen installed: the cooling air flow was then adjusted until air passed through the device was not noticeably heated. When the desired condition was achieved, a screen was then inserted and tested. As an added benefit, the fan cooling prevented a transient heating of the chambers during testing, which had previously caused problems.

Tests were carried out on a variety of materials. Three are selected as being representative. These are a standard fiberglass window screen of 16 x 18 mesh, with thread thickness of .0065 inch, a finely woven nylon drapery backing sheer fabric with a thread count of 90/inch and thread thickness of .0015 inch, and a black dyed cotton organdy fabric of thread count also 90/inch, and thread

diameter of .002 inch. The averaged results of a number of tests on these materials gives the results below; the flow velocity was a free stream 2 feet per minute in all cases. The temperature rise of the air stream is the measured value and the temperature difference between screen and air is taken as being between the screen measured temperature and the averaged air temperature, assumed to be halfway between its inlet and outlet temperatures.

Material	Temperature Rise °C	Temperature Difference °C
Fiberglass Screen	10	17
Cotton Organdy	8	10
Sheer Nylon	7	7.5

From this data, it appears that the finest weave with the smallest thread diameter provides the highest heat transfer coefficient, i.e. the highest ratio of temperature rise to temperature difference. Values of this ratio are .59, .80, and .93 for the screen, organdy, and nylon respectively.

Optical properties of the three materials were also investigated, using the pyranometer and beam and diffuse insolation. Transmittance was measured at various beam incidence angles. The pyranometer was first checked for cosine response, and found to be within one per cent accurate. Zero degrees of angle is considered as normal incidence, in keeping with the standard convention.

Optical Transmittance Per Cent

	0°	7.5°	15°	22.5°	30°	37.5°	45°	Diffuse
Fiberglass	62	62	62	61	61	59	57	56
Cotton	58	57	57	56	55	54	50	58
Nylon	74	73	73	72	71	69	66	67

From this table, it can be readily seen that the transmittance of the sheer nylon is superior. In-depth absorbtion in a screen stack would thus be best accomplished by this material. Also important is the ratio of beam to diffuse transmittance, as the latter property governs the level of infrared recapture, because this radiation is diffuse. The nylon fabric is slightly superior to the cotton in this regard also. Given its better heat transfer and optical properties, the

black sheer nylon is superior to black cotton organdy in a screen stack absorber. Organdy was nevertheless used. This was for reasons of availability. While enough of the nylon fabric was available to the author, it was from a private party and had been purchased some time ago. A check of five local fabric shops revealed that this fabric is not widely available, whereas the cotton organdy is. It seemed more appropriate to test the fabric which would be more likely to be used by do-it-yourself collector builders. Tests on the finished collector showed, however, that the extra effort a homeowner would need to locate the nylon fabric would be well worth the effort. This problem may ultimately be resolved, however, by the timely commercial availability of improved design absorbers, which are certain to be better than any simple screen stack.

The "heat transfer coefficient" of a given screening can be defined either in the conventional manner, as based on square footage of thread surface, or it can be defined in terms of a "practical" measure, based on square footage of flow cross sectional area. The coefficients measured, as defined conventionally, are 1.70 BTU/hr-foot²-°F for the fiberglass, 1.46 for the organdy, and 2.28 for the nylon. The "practical" measures are 1.23 for fiberglass, 1.68 for organdy, and 1.96 for nylon. Organdy has the highest area ratio, i.e. the thread surface square footage per square foot of fabric, the figure being 1.15; it is 0.7 for fiberglass and 0.86 for nylon.

A flow Reynolds number, based on conventional definitions, can be calculated using the methods of London et. al. (4,5). The method was reviewed in the mid-term report of the project, and need not be reviewed here. Flow Reynolds numbers based on 2.0 feet per minute free stream air velocity are 1.15 for fiberglass, 0.199 for organdy, and 0.209 for for nylon. The conventional definition of hydraulic diameter is used.

A Stanton number, based on the engineering definition of heat transfer coefficient can be calculated. St is 0.84 for fiberglass, 0.75 for organdy, and 1.00 for nylon. These results, within the considerable limits of experimental error, confirm an extrapolation of London's data plots, and of the correlating equation he devised. Organdy is seen to be inferior in effective heat transfer coefficient, but is a better choice in practice than fiberglass due to its superior area ratio.

Efforts were made, as mentioned above, to remove as many artifacts as possible from the experimental results, but there is some likelihood that inaccuracies

remain. For example, the cooling of the glazings was adjusted in the absence of screening. The introduction of a screen may alter the various heat fluxes in the system, and alter temperatures. A cooled top glazing may also, to some extent, remove heat from the exit airstream under test and give an artificially low measured temperature. Convective transferral of heat via these types of mechanisms should in theory be low, as flow is laminar, but the actual streamline pattern in the device could be complex, and natural convections could play a role. Another potential source of error in this experiment could be the thermistor; it is not possible within project scope to check its reading for accuracy under dynamic conditions.

Errors may have crept into the experiment performed, but the basic design of test device appears to be about the only possible method, to the author's knowledge, of experimentally determining this type of data. The use of large, dynamically similar systems is ruled out by the artifact introduced by natural convection at the very low air flow rates which would be required, so actual size screens must be used. A transient cooling test on actual sized screens is ruled out by the great artifact which radiative transfer would introduce. Accurate determination of heat transfer behavior for various porous absorber elements under solar panel flow conditions may, moreover, be of more than academic interest. Optimization of the improved design of porous absorber may be made much easier by a method of accurately conducting the type of test under discussion. Fortunately, it appears that a more sophisticated version of the design of device which the project developed can provide very accurate data, although at considerable cost.

The proposed improved version of the heat transfer test device is illustrated on the following page. The main improvements over the design used are in the more sophisticated wall and glazing temperature control systems, and in the use of the most advanced reflective coatings for the surfaces of the thermally passive chamber walls. Five degree infrared pyrheliometers are used to measure the temperatures of both the active screen element and the lower cover glazing for the ambient-air flow chamber. The walls of the lower chamber are also monitored, and held at the air inlet temperature by a sophisticated temperature control system. Air entering the screen orifice is to be made as close as possible to isothermal, and laminar in flow.

The temperature sensor reading the heated air temperature is to be located within the laminar airflow path above the screen. It is protected from the infrared radiation and optical reflection of the screen by a protective reflector. The

walls of the upper chamber, including the light beam inlet glazing, will be temperature controlled to the measured temperature of the air stream. Clearly, the logic of the control element given this job must be designed with an intelligent deadband system to prevent a "runaway" feedback effect, but this problem should not be too difficult to solve. The temperature sensing element will, finally, be at the end of a long tube, and will be a small unit placed directly to the rear of an orifice, across which a sufficient pressure drop to provide overall minimum disturbance of the air flow velocity profile will be applied. The sensor, pipe, reflectors, and sidewalls of the top chamber will all be silvered to a very high reflectivity in both the optical and the infrared. The purpose of the orifice is to give the air stream a high velocity at the sensor position, leading to a minimal convective film thermal resistance, in the case of a slight radiative disequilibrium and radiative heat transfer to the sensor.

This system may be overdesigned, but some of its features are certain to be of value. In a situation where best accuracy is required, all of the complexities of radiative heat transfer, including multiple partial reflections and absorptions, must be taken into account. This is why the upper as well as lower chamber walls should be temperature controlled, and why the sensor should be protected from a direct view of the test screen, or other porous element.

First Collector Tests

An operating collector was built under the project, and was tested through the months of June-September of 1980. Tests on this unit will be ongoing through the coming winter months as well. Based on the criterion of widespread availability to individual collector builders, the organdy fabric was, as explained above, chosen over the nylon sheer fabric in spite of the demonstrably superior optical and heat transfer properties of the latter.

The collector design was as illustrated below. Four organdy screens were employed. The inner rear wall of the collector cabinet was coated with reflective aluminum foil, with the idea of minimizing infrared reradiation, but as it turned out this was not especially important. Absorptivity of the screen stack, except for the unknown primary reflection from the threads, was very good. Reflection from the aluminum foil was entirely invisible, even at 0° incidence. Assuming random shading, i.e. no anomalous superposition effects, the transmittance of the effective 8-fold screen stack, at 0° incidence, would in theory be $0.58^8 = .013$.

Glazing of the collector was done with "Flexigard", a plastic product developed by the 3-M Corporation specifically as a replacement for glass in greenhouses, solar panels, and storm windows. It is a two layered material, consisting of an acrylic outer layer and a pthallic polyester inner layer. The acrylic provides weathering ability, and the polyester provides mechanical strength and dimensional stability at elevated temperatures. A thin layer of an ultraviolet absorber is included between the two layers, to protect the polyester from ultraviolet degradation. The complete film is 0.008 inch thick. The manufacturer claims an impact strength as high as safety glass, with the product meeting Impact Test 5.13 of the American Standard Safety Code for Safety Glazing Material. Ten year exposure to Florida insolation without significant optical or mechanical degradation is claimed. The material performed very well in the test series, but of course no long term experience was gathered.

Flexigard was chosen over glass because it is inexpensive and lightweight. The major goal of the project was to test porous absorber designs, but another project goal was to evaluate in practice a number of the more promising products recently made available to do-it-yourself and small commercial solar airheater builders. The weight and cost of tempered low iron glass are excessive, and a reliable alternative would be very welcome. Flexigard was chosen over a number of competing alternative products, such as Kalwall's Sunlite, Chemplast's Llumar, DuPont's Tedlar, and Rohm and Haas' Plexiglass, because of its low cost of 50¢ per square foot in quantities suitable to small scale panel construction, and the advantages of high resistance to thermal expansion and long term degradation which are promised.

The optical transmittance of the cover glazing was tested with the pyranometer. This unit, a model MR-5 from Hollis Observatory, was itself tested for cosine response and found accurate to within 1%. This good accuracy at all incidence angles, and the insensitivity of the unit, which is of the temperature compensated silicon cell variety, to use in a tilted position, were important reasons for its preference over a thermal response unit. In actual collector tests, the pyranometer was mounted in the plane of the collector glazing, and was thus subjected to the same precise insolation that the collector received.

Cover glazing optical transmittance, at various beam incidence angles and for diffuse radiation, was measured as:

Cover Glazing Transmittance Per Cent							
0°	7.5°	15°	22.5°	30°	37.5°	45°	Diffuse
92	92	92	92	91.5	91.5	90.5	86.4

The manufacturer claims an infrared absorbance for Flexigard of better than 98% in the region of 5 to 12 microns wavelength, which comprises the great majority of the blackbody spectrum in the temperature region of interest. About 40% transmittance occurs in the region of 12 to 15 microns, but the overall energy capture and greenhouse effect associated with the product should be good. Flexigard thus overcomes one of the more important disadvantages of a number of plastic glazing materials.

The body of the test collector was made of wood, and the screens were held in place by gluing to wooden support strips, as illustrated in the drawing, which were about 1 inch in width. A fairly reflective tape was used to seal the gaps between these strips and the cabinet inner wall. Optical radiation impinging on these screen support strips, and on the glazing supports as well, must be considered as largely lost. Thus, while the total frontal area of the collector assembly was 24 by 48 inches, or 8 square feet, of this only 20 by 44 inches, or 6.1 square feet, can be considered as active absorber area. This inefficient use of area could not be tolerated in a commercially produced collector, and would not have to be. Larger total module aperture areas, and more refined glazing and screen supports, could make much better use of available area. The unit in question was designed as a test model only, and this made efficient use of area unimportant. Of greater importance was simplicity of construction and, most of all, a design which gave priority to ease and low cost of disassembly and reassembly in various, not necessarily foreseen, alternative configurations.

The collector was mounted as a tripod, with two small feet attached to the cabinet bottom and an adjustable third leg, as shown. The unit was set on a stage at ground level, and could be tilted up to 35° off of vertical and swiveled to point in any desired direction. The pyranometer was mounted above the collector body, and pointed in the same direction. A shadow dial was used to measure the beam incidence angle. Data was generally taken in late afternoon, to try to work at the lower levels of insolation, which test the design more fully. The collector was manually repositioned after each test reading to change the solar incidence. Readings were taken between 0° and 45° incidence, at 7.5° intervals. The collector was adjusted sequentially in 7.5° increments, to keep the change in insolation, and the associated transient effect, to a minimum. Generally, about 20 minutes was allowed between readings, to allow an approach to steady state behavior. This appeared to be adequate in trial early tests, and was adopted. As wide a mixture of incidence angles in relation to insolation as possible was secured.

Some shading of the absorber by the cabinet wall and glazing supports occurred, particularly at the higher incidence angles. This shading was excessive compared to commercial practice, due to the small size of the collector. The percentage of the absorber plate so shaded was crudely measured with a ruler (the shadow was plainly visible) and is included in the test data below. This shading was not taken into account in the efficiency calculation, which is on the basis of 100% energization of the absorber area. Some shading loss is felt to be inevitable in any collector design, and must be considered as a real efficiency loss.

Polyurethane foam sheeting was used to insulate the back and sides of the collector. The back was insulated to a total value of R-7, and was of 8 square feet expanse. The sides, top, and bottom were insulated to a total of R-3, and account for a total of 6 square feet, as the collector cabinet is roughly 6 inches deep. Sidewall insulation is probably less than adequate, and may be improved in future tests. The entire assembly was well caulked, after the glazing was installed, with acrylic latex caulking compound. Particular attention was paid to the joints between the glazing supports and the collector body; it was made sure that no significant leaks were present.

Testing of the collector in its first configuration was primarily designed to test the screen stack, but other design aspects were simultaneously tested when possible. A severe mechanical test of the glazing material was carried out, by means of placing the air flow orifice meter upstream of the collector, and the suction source, which was a powerful workshop vacuum cleaner, downstream. This subjected the collector interior, and the glazings, to a vacuum of as high as 6 inches H_2O . The purpose was to attempt, in a short time, to mimic the effect of repeated wind and snow loadings on the plastic glazing. The results were that the glazing itself was undamaged and did not appear to stretch, but the need for a very secure mounting system was demonstrated. The 3-M corporation recommends a framing system wherein the Flexigard film is sandwiched between two layers of double adhesive tape, which in turn are attached to rigid framing structures. This tape was not used in the tests; contact cement was tried first without supporting hardware, and found to be inadequate. A serious ballooning of the cover glazings occurred, until they were practically touching. The modified collector incorporates a sandwich configuration in which the film is bonded to a wooden frame attached to the collector, and another frame is then tightly attached with wood screws over the first, making a strong mechanical clamp. Two glazed frames, and two unglazed, are employed. This configuration appears to be adequately strong. Epoxy glue appears to be satisfactory as a bonding agent. Of potentially

greater practical interest to do-it-yourself workers may be a silicone bonding and sealing compound. In a collector not designed to be disassembled, all joints could be secured with this compound, and subsequently put under mechanical compression with screws. This could give an excellent permanent joint.

The Flexigard product is rated at only 130°C for thermal resistance, which is below the stagnation temperature of many collector designs. Capped off stagnation tests were done on the test collector, under conditions of high ambient temperature and high insolation. Stagnation temperature never rose above 80°C . This could, however, be in part due to the relatively poor insulation of parts of the system. It is, however, likely that a large natural convective loop developed in the collector interior, with air circulating through the porous absorber and down the inner cover glazing, in these tests. This being the case, the considerable depth of the cabinet, and the negligible pressure drop associated with the porous absorber, are a natural convective protection against a high stagnation temperature.

The pressure drop through the screen stack was measured during airflow, and was found to be less than 0.01 inch H_2O ; in fact too small to be noticeable even with a manometer of 0.01 inch resolution. The major pressure drop in the system, requiring a powerful blower, was the orifice meter itself. A head loss of around 0.1 inch of H_2O was found to be attributable to the ductwork leading to and from the collector box. The ducts were 3 inch elbows, and could have been larger for better performance. Given adequate ductwork, the very low pressure drop associated with adequate absorber heat transfer performance implies the possible application of the absorber concept to thermosiphoning collector systems. In situations where thermosiphoning cannot be employed, the blower horsepower requirement could be very small, even for large systems.

The energy capture efficiency of the design, and the degree to which in depth absorption and temperature stratification occur are the focus of major project interest in the evaluation of the design. A good deal of testing was done, and much data gathered. That which is to be presented below will be representative but by no means exhaustive. The data consists of the air flow measurement, the air temperature rise as measured by thermometers in the inlet and outlet ducts, the ambient temperature as measured by a shaded thermometer, the insolation and its incident angle, and the temperatures of the two cover glazings and four screens, as measured by calibrated thermistors protected by aluminum foil covers.

Two modes of collector operation were experimented with. Air in the first mode was inlet at near the outdoor ambient, and in the second mode was preheated by 20°C or more prior to collector inlet. It may be argued that the first mode

of operation is unrealistic, for under actual operating conditions the inlet temperature is always higher than the ambient. This point is correct, but it is also true that the radiative losses associated with a given temperature difference are a function of the cube of the absolute temperatures; by differentiating the Stephan-Boltzmann equation we have:

$$dW/dT = 2.27 \times 10^{-7} T^3 .$$

Calculations show that at the typical glazing temperature in the 45° C range typical of summertime tests, the radiative losses for a given temperature difference will be close to 50% greater than for the same temperature difference at an average temperature of 15° C. The differential argument is heuristic rather than precise, but the major point is clear. Neither summertime mode of collector operation can truly correspond to wintertime conditions. The ambient inlet mode may give a slightly positive bias to the results, but by the same token the preheated inlet mode will put collector operation at a serious disadvantage. Wintertime tests will have to wait until winter. There is no penalty in this, as the improved design, which is the main hope for truly efficient operation, was conceived of only a short time ago. In addition to excessive radiative losses, an anomalous effect which enormously reduces collector efficiency is noticed in some of the hot-inlet experiments. This will be discussed below.

The calculations upon which the collector efficiency determination was made will be presented in detail, for the convenience of the reader. Beginning with the flowmeter, an orifice plate of .125 inch thickness was employed. The orifice was of 0.75 inch diameter and the ducts were of 1.125 inch inner diameter, giving a beta value of 0.6 and values of Re_d of between 35,000 and 65,000 at the flow rates employed. Calculations and references showed an acceptably accurate average discharge coefficient of 0.61 could be employed, with variation over the test range of less than 2%. The reports of the ASME Research Committee on Fluid Meters were consulted for the relevant data.

With a discharge coefficient of 0.61, including the velocity of approach factor, the orifice flow velocity can be calculated from the measured pressure drop across the flange taps by the corrected Bernoulli equation;

$$V \frac{\text{feet}}{\text{sec}} = \left(\frac{2 \times g \times \Delta p \frac{\text{lb f}}{\text{ft}^2 \text{a}}}{\rho (T) \frac{\text{lbm}}{\text{ft}^3}} \right)^{\frac{1}{2}} \times 0.61$$

For the ambient inlet collector tests, the air temperature at the flowmeter was generally at about 30°C , at which temperature atmospheric pressure air weighs about 0.0722 lbm/foot^3 . Two working pressure drops were chosen; they were 2.5 and 4.5 inches H_2O , measured by a manometer placed across the flange pressure taps. These pressure drops, equivalent to 13.0 and 23.4 lbf/foot^2 , resulted in orifice velocities of 65.6 foot/sec and 88.2 foot/sec respectively. The 0.75 inch diameter orifice therefore passed $\pi/4 \times 0.75^2 \times 144^{-1} \times 60 \times V = 12.1$ and 16.2 cubic feet per minute at the respective pressure drops. Given an active absorber area of 6.1 foot^2 , the net flow rates are 1.98 and $2.66\text{ SCFM per foot}^2$. These values can be rounded off to 2.0 and 2.7 respectively, without an important accuracy loss.

Assuming an air mass flow of $2.0\text{ cfm per foot}^2$, at 0.0722 lbm/foot^3 inlet density this amounts to a mass flow of 8.66 lb per hour , or $0.3\text{ lb mole per hour}$. C_p for air at the temperatures of interest is $7.0\text{ BTU/lbmol-}^{\circ}\text{R}$. This implies an air heating energy requirement of $2.1\text{ BTU per }^{\circ}\text{R or }^{\circ}\text{F temperature increase}$. Assume an insolation of 1 langley , equal to $1\text{ cal/cm}^2\text{-minute}$ or to $221.2\text{ BTU/hr-foot}^2$. A 100% efficient solar air heater at this flow rate would heat the inlet air by 105°F , or 58.5°C . Actual heater performance can be calculated by comparing the real temperature rise to this theoretically 100% efficient figure. For the case of $2.7\text{ cfm per square foot}$, the figure for 1 langley is 43.4°C .

Actual test data, representative of the body of results collected, is shown below. Glazing and screen temperatures are shown, in Centigrade degrees, with G_1 the outer glazing, and S_1 the top screen. T_a , T_i , and T_o are the ambient, inlet, and outlet temperatures respectively. T_i is often a few degrees higher than T_a , perhaps due to solar heating of the inlet duct. The data is organized by flow rate, with the 2.7 cfm data presented first, and by incidence angle. Shading factors for the absorber are shown for the first data set, but they are not factored into the efficiency calculations.

Temperatures shown are accurate to about 1°C . Flow measurement is probably accurate to about plus or minus 5% . Care was taken to attempt to achieve steady state conditions for each data point. Data taking on excessively cloudy days was avoided, and data points of questionable value due to cloud intermittent interference are not included.

The format of data tables is arranged to read from lower to higher temperatures. All numbers not otherwise labeled are Centigrade temperatures. The pattern which emerges from the data is clear.

T_a	T_i	G_1	G_2	S_1	S_2	S_3	S_4	T_o	$n\%$	I
				<u>Flow</u>	<u>2.7</u>	<u>Incidence</u>	<u>45°</u>	<u>Shading</u>	<u>20%</u>	
33	34	41	45	45	47	50	49	48	34%	.970
37	38	43	48	48	50	53	52	50	32%	.916
				<u>Flow</u>	<u>2.7</u>	<u>Incidence</u>	<u>37.5°</u>	<u>Shading</u>	<u>15%</u>	
31	33	42	46	47	49	53	52	50	41%	.954
32	32	42	46	46	49	52	51	50	40%	1.01
				<u>Flow</u>	<u>2.7</u>	<u>Incidence</u>	<u>30°</u>	<u>Shading</u>	<u>10%</u>	
32	33	44	48	49	53	57	56	56	49%	1.09
				<u>Flow</u>	<u>2.7</u>	<u>Incidence</u>	<u>22.5°</u>	<u>Shading</u>	<u>2%</u>	
30	32	43	48	49	53	57	56	56	52%	1.06
31	33	41	45	46	48	52	52	52	49%	.990
				<u>Flow</u>	<u>2.7</u>	<u>Incidence</u>	<u>15°</u>			
31	31	38	42	43	45	48	49	50	49%	.90
31	32	43	48	49	53	57	57	56	51%	1.08
				<u>Flow</u>	<u>2.7</u>	<u>Incidence</u>	<u>7.5°</u>			
30	31	41	47	47	5-	55	55	54	50%	1.07
				<u>Flow</u>	<u>2.7</u>	<u>Incidence</u>	<u>0°</u>			
30	30	40	46	47	51	55	54	53	51%	1.04
				<u>Flow</u>	<u>2.0</u>	<u>Incidence</u>	<u>45°</u>			
32	34	39	43	49	52	56	56	52	30%	1.04
				<u>Flow</u>	<u>2.0</u>	<u>Incidence</u>	<u>37.5°</u>			
35	35	41	44	50	53	57	55	54	34%	.970

T_a	T_i	G_1	G_2	S_1	S_2	S_3	S_4	T_o	$n\%$	I
<u>Flow 2.0 Incidence 30°</u>										
34	34	40	44	52	55	59	58	57	33%	1.20
<u>Flow 2.0 Incidence 22.5°</u>										
34	34	43	48	53	57	61	61	59	37%	1.15
27	29	37	43	49	55	59	59	57	38%	1.26
<u>Flow 2.0 Incidence 15°</u>										
34	36	43	49	54	58	62	62	61	36%	1.18
27	29	38	44	50	55	60	61	59	41%	1.26
26	28	37	42	47	51	56	56	54	40%	1.11
<u>Flow 2.0 Incidence 7.5°</u>										
27	29	38	43	49	55	59	60	57	40%	1.23
26	28	37	42	47	51	56	56	54	40%	1.11
<u>Flow 2.0 Incidence 0°</u>										
26	28	38	42	48	52	57	57	55	40%	1.16
25	27	34	39	44	48	52	53	51	41%	1.04
25	27	32	36	41	45	49	50	48	38%	.940

An examination of this data shows that the screen stack makes a reasonably efficient collector, but not an outstandingly efficient one. The system appears to be capable of an average 40% efficiency at 130° to 140° F outlet temperature, and of about 50% in the 120° to 130° range. The collector, of course, has an ambient inlet in these tests, which makes it easier to achieve good efficiency. On the negative side, however, the levels of insolation under which testing took place were deliberately chosen to be lower than the average used in bench testing solar hardware, and thus probably more realistic in terms of applications. As mentioned previously, moreover, operation was in a range where radiation losses are augmented; not necessarily so with regard to other collector designs so much as with regard to the greater efficacy to be expected from the porous absorber loss suppression mechanisms at lower inlet temperatures.

There is a noticeable increase in collector efficiency as normal incidence is approached, and this effect does not appear to be solely due to increased insolation levels, as these were made low in some of the 0° incidence tests. The data do not, however, show any dramatic increase in stack temperature stratification, and the artifact of cabinet shading at high incidence may be more responsible for the measured efficiency decrease than any supposed alteration in the system thermal network.

The major result of the tests, however, centers on the high relative temperature of the inner glazing. In the higher temperature test series, this glazing temperature would be an average of 15 Centigrade degrees above the outdoor ambient. The primary reason for this appears to be radiative energization due to the lack of complete radiative recapture in the screen stack. This is strongly suggested by the temperature relations between G_2 , S_1 , and S_2 and particularly by their dependence on the background temperatures from the back two screens (the "asymptotic" screens) in the stack. This can be seen by comparing the typical temperature profiles for the two velocities of airflow.

Forced convective heat transfer from S_1 to G_2 is unlikely to be important. The inlet air enters at ambient, and would require a heating of 15° C before being able to transfer any heat to the glazing. Convective transfer most likely occurs in the reverse manner. In the case of ideal ducting, the airflow between S_1 and G_2 would be laminar. Some turbulence does exist in the test unit, as was shown by a smoke test, and it appears to be due to the restrictive ductwork employed. It does not appear, however, to be excessive. In any event, screen materials are exceptionally good dampers of nonuniform air flow, as the simple experiment of blowing through one will show. If one blows hard through an organ, a gentle breeze on the other side results. Place two fabrics in a row, and no breeze gets through. This phenomenon, plus the fact that air picks up heat from the fabric not by a turbulent, but by a laminar throughflowing mechanism, makes it unlikely that forced convection plays a role in the transfer of energy to the cover glazing.

The insulating ability of the outer cover glazing in these tests was probably not good. The test of the adhesive mounting system caused the glazings to collapse together and, although they never to the author's knowledge made contact, they came at places into very close proximity. Due to this, it is also uncertain if the temperature of the outer glazing was uniform. Due to this type of uncertainty, a quantitative estimate of the loss mechanisms cannot be made. It is likely, however, that some losses could have been suppressed if the outer glazing had been more securely mounted. In such a case, the temperatures of the screens and

the output air, and the inner glazing as well, would be increased.

A final point of interest concerns the temperature profile in the screens S_3 and S_4 , and the output air. In the higher airflow tests, these temperatures are generally within 1° of one another. The air temperature is probably heated in a roughly asymptotic way. In the case of the slower air flow and higher outlet temperature the same general regime applies. It may be noticed, however, that the temperature of S_4 is in general slightly higher than T_o , the outlet temperature. This may indicate the incipient onset of a loss mechanism, to be discussed below, which could be responsible for some instances of very poor performance in the case of the heated inlet tests.

Air at higher than ambient temperatures was supplied in the heated inlet tests by takeoff from the hot exit of the vacuum cleaner employed as an air source. This motor used roughly 600 watts of power, so the heating effect is considerable. Data are presented below.

T_a	T_i	G_1	G_2	S_1	S_2	S_3	S_4	T_o	$n\%$	I
<u>Flow 2.62 Incidence 45°</u>										
26	52	43	57	63	66	68	68	62	18%	.984
<u>Flow 2.62 Incidence 37.5°</u>										
24	48	40	52	60	62	66	65	59	20%	.896
30	54	43	58	67	64	72	69	61	11%	1.07
<u>Flow 2.62 Incidence 30°</u>										
26	54	45	60	68	73	75	73	65	16%	1.14
<u>Flow 2.62 Incidence 22.5°</u>										
26	53	45	60	68	73	76	74	66	18%	1.21
25	50	41	52	59	63	66	65	61	21%	.860
<u>Flow 2.62 Incidence 15°</u>										
25	50	41	52	59	63	66	65	61	21%	.860
26	53	46	60	69	75	78	76	68	20%	1.24

T_a	T_i	G_1	G_2	S_1	S_2	S_3	S_4	T_o	$\eta\%$	I
<u>Flow 2.62 Incidence 7.5°</u>										
26	52	46	60	70	75	78	76	71	26%	1.21
26	50	40	55	62	67	69	68	65	24%	1.01

<u>Flow 2.62 Incidence 0°</u>										
27	53	46	60	71	75	78	77	71	24%	1.23
27	49	40	55	62	67	70	69	65	24%	1.08

The generally low efficiency in these tests is to be expected, given the poor insulating qualities of the damaged cover glazing and, especially, the high inlet temperature. Flat plate collectors are not meant to operate in the exit temperature range of 160° F, unless a selective absorbent is used, with any degree of efficiency. The higher efficiencies achieved in these tests of around 25% are probably typical of airheaters operating in this outlet temperature range. Liquid cooled collectors, especially with double glazing and a selective absorber, can be considerably more efficient; figures of up to 45% are not uncommon.

Data reported above were taken at a manometer pressure reading of 4.5 inch H_2O which, due to the lower density of the heated air, corresponds to a higher actual flow velocity but to a lowered mass velocity. The flow is expressed in standard cubic feet per minute, and is somewhat less than in the case of ambient air inlet. Data was also taken at 2.5 inch H_2O , but will not be presented. The data above contain the information necessary to make all important points.

The inner glazing G_2 is generally still hotter in these tests than is the inlet air, but this is not as pronounced as in the case of the ambient inlet tests. This is probably in large part due to the much greater difference between the glazing temperatures and the outdoor ambient, and the consequent much greater ease in transferring heat to the environment from the collector surface.

The most interesting effect, and one which may in large part explain the low overall collector efficiencies, becomes apparent when the temperatures of S_3 and S_4 are considered in relation to the outlet temperature. In all cases the screen temperatures are considerably higher than the outlet temperature, and in the case of the least efficient operation this temperature difference

is greatest. It is the author's suspicion that this state of affairs indicates a dominance of natural over forced convection in the collector local air flow patterns.

Consider the temperature field in an ideal porous absorber collector, as illustrated on page seven. This is not a stable system and under certain circumstances buoyant air convection could cause a redistribution to occur, with heated air being forced from back to front at the top of the screen stack, forming a convective loop such as illustrated below.

This phenomenon is supported by any convective cooling which may occur as the current impinges on the inner cover glazing. The net effect is to overheat the top sections of the outer part of the screen stack, disrupting the forced convective

temperature field at the top of the screen stack. This situation is most likely to occur when temperature rise across the screen stack is high, forced flow velocity is low and, perhaps most importantly, when the inner glazing temperature is not exceptionally high compared to the inlet temperature. This latter situation especially characterizes the heated inlet test regime. In practice, the natural convective tendencies, illustrated in simplified form above, will be superimposed upon the forced convective flow, leading to the erratic performance and temperature patterns noticed in the heated inlet test data. The data for the lower flow velocity shows this to an even more exaggerated extent.

The placement of the thermistors in the test collector was slightly above the screen centers, and their positioning could have been fortuitous in recording a relatively consistent screen temperature field. Only one sensor per screen was employed, as the situation which appears to have developed was not anticipated. A more sophisticated test would employ at least three independent sensors per screen and glazing, giving three independent temperature field determination at the top, middle, and bottom of the screen stack. On the other hand, behavior in the "transition region," i.e. where flow velocities and temperature profiles are such that neither forced nor natural convection dominate, the local air behavior could well be so erratic that no significant generalizations could be made. It appears to be a valid conclusion, however, that a natural convective regime operates unimpeded during capped-off stagnation, and prevents the stagnation temperature from becoming excessive. Fully forced convection appears to dominate when the imposed flow velocity is fairly high, and if the inner glazing acts as a significant heat source with respect to the inlet air.

Transition region operation appears to be manifested in the data collected by the anomalous relation of output temperature to rear screen temperature. This temperature difference repeatedly occurs to varying extents in different data sets, and appears to correlate roughly with a collector efficiency lower than expected. It predominates in the very low efficiency high temperature inlet tests, and may also play some role in the low airflow ambient inlet tests, reducing the collector efficiency below achievable levels.

The competition of natural with forced convection in the porous throughflowing absorber, and the resulting performance deterioration, was not anticipated by the author. It apparently owes its existence to the very low flow friction resistance of the screens to low velocity air flow. Tests, mentioned above, were carried out to attempt to measure the head loss across the screen stack at various flow rates

of up to 4 cfm per foot². No detectable pressure drop occurred, with a manometer of 1 in 12 slope, and a sensitivity able to resolve 0.01 inch H₂O. While very good at damping jetlike flows, screens of even very fine weave do not offer much resistance to low velocity bulk flows.

The performance degradation due to the transition effect could seriously compromise the practicality of the screen stack absorber, especially in collector units of larger vertical dimension than the one employed in the tests. Several remedies exist. Use of small modules, for such purposes as window-mounted solar heaters, will probably avoid the problem. Larger screen stack units could divide the hot air plenum behind the stack into two compartments, each with access to half of the takeoff duct, by means of a vertical partition parallel to the rear wall and extending from the collector top to the rear midpoint of the screen stack. These points, however, are probably of primary interest only to do-it-yourselfers. The commercial scale solution to the problem is to abandon the screen stack in favor of a superior design of porous absorber, which can both solve the reradiation problem and also allow a high enough forced convective pressure drop to be engineered into the design to avoid buoyant convective backflow difficulties. This absorber design is the subject of the following section of the discussion.

The Improved Porous Absorber Concept

From the data discussed above, it is evident that the screen stack is an imperfect porous absorber to a practically significant degree, and that a considerable amount of useful energy is being lost as infrared reradiation even under the best of circumstances. It is possible, however, in practice as well as theory, to recover much of this radiation by placing one or more optically transparent, but infrared opaque heat transfer elements in the coolant flow path as covers to the optically opaque elements. An early design intended to partially accomplish this, originally studied by Lof (6), consists of overlapping half transparent, half blacked glass plates, arranged so that the clear section overlaps the blacked subsequent section, in a stepwise configuration allowing thin rectangular air passages between plates. This design has never been used in a practical solar heater, to the author's knowledge. Such a design would be very costly, and the reradiation loss would still be considerable due to the relatively low forced convective heat transfer coefficients allowed by the geometry.

It is proposed, as a commercial design, to employ a laminate of spaced plastic perforated films as a porous solar airheater absorber. The topmost one or two of

these films are to be as transparent in the optical spectrum as possible, and the lower films can be of varying number and optical transmittance.

A possible practical embodiment of the design, drawn without attention to scale, is illustrated below.

This illustration is presented only as a qualitative conceptual aid, and represents no hard and fast design rules. All dimensions are nominal.

Actual optimization of the absorber design concept would require considerable effort. The number of stages, their optical densities, and the character of the perforations are independent design variables. The design temperature and typical operating conditions must be taken into account. The heating of air from roughly 60°F to 130°F could turn out to be a job which could be fairly easily accomplished by a low cost, efficient absorber of this type.

An infinite number of transparent stages are required in theory to achieve 100% radiative recapture and ambient first stage operation, but a small number of stages account for a great majority of the recapture. Economical construction, with four as a nominally best number of stages, but perhaps with three being practical also, is thus possible. Heat transfer considerations are important, as in the screen stack case.

The heat transfer effectiveness of the absorber films will depend on the number and type of perforations employed. Several options are possible. Two such are the use of jet creation/surface impingement as a heat transfer technique, and the throughflowing laminar convective option. The latter offers very good heat transfer/flow friction characteristics, especially as entrance effects will dramatically increase the average value of Nu for the passages, which will almost invariably have a length to diameter ratio of less than unity.

It is beyond the scope of this discussion to attempt to optimize the design, or invent the details of a manufacturing process, for an absorber of the proposed type. As far as optical design is concerned, it can be simply pointed out that perforations of the type proposed need not materially decrease the transmittance, or increase the reflectance, of a transparent film. Arguments based on ray tracing in the case of a perfectly transparent material, with perfectly square edged perforations, show the loss to reflection to be no greater with the perforations than without; in other words any ray of incidence sufficient to avoid total reflection will pass through the film, whatever its encounter with a perforation may be. In practice, the use of perforations of diameter significantly greater than the film thickness, and of relatively wide spacing between perforations, may make the question of perforation effect on transmittance one of negligible importance.

Manufacture of absorbers of the proposed type will probably be done most economically in mass production, but this does not necessarily imply a centralized industry. There are many plastics-related firms which employ small numbers of people and are yet able to perform highly sophisticated operations at high volume. Even the domination of absorber production by one or two very large corporations would not necessarily imply a centralized solar industry, as such corporations would be more than likely to engage in the production of absorbers only, and the fabrication of collector modules and, inevitably, system installation, would be a creator of local and regional job opportunities.

Cost of the absorber in mass production cannot be realistically estimated by the author, but there is no reason to believe that it could not be made low. Installation of the perforations may be an expense, but it need not be a great one. It can be efficiently done mechanically by automatic punching, if the perforations are large enough. If an immense number of very small perforations turns out being superior, a method of producing them with electrical discharges could probably be devised.

A very crude approximation to a design of this type was constructed by modifying the test collector. An infrared absorber and throughflowing air heater was assembled from two sheets of perforated Flexigard, and installed in the air flow path in front of the four black organdy screens. The perforations were made with a "pin cushion" of size 3d finishing nails in a wood backing. The nails were of 0.065 inch diameter, and the plastic glazing did not appear to exhibit a significant self-healing effect upon puncture, it being a relatively

hard plastic material. Some crazing of the material around the punctures did occur, but pyranometer tests showed a negligible difference in transmittance between perforated and unperforated glazing. An average of 8 perforations per square inch were installed. This amounts to a free flowing area percentage of slightly under 3%.

The new arrangement was reassembled as shown below.

Thermistors were placed on the new porous glazings, G_{p1} and G_{p2} . Unfortunately, the thermistor on G_{p2} failed after assembly, but the one on G_{p1} remained functional. The orifice meter in these tests was placed downstream of the collector, and operated at an air temperature roughly equivalent to that in the preceeding hot inlet collector tests. The same pressures, and flow rates, as in the hot inlet tests, were used.

The glazing was reassembled in a sturdy manner, and with a greater gap between inner and outer sheets than was previously employed. Authorities differ on the correct gap between glazings in a double glazed collector. Most recommend about .75 inch, but some claim as much as 2 inches is better. A one inch gap was used.

Data from the tests on the rebuilt system are presented. The data for the temperature of the porous glazing G_{pl} is omitted. The reason for this will be discussed below.

T_a	T_i	G_1	G_2	S_1	S_2	S_3	S_4	T_o	$n\%$	I
				<u>Flow 2.62</u>		<u>Incidence 45°</u>				
27	28	36	39	45	52	53	51	51	52%	.988
27	28	36	41	47	53	55	54	50	52%	.965
				<u>2.62</u>		<u>37.5°</u>				
25	26	35	38	45	51	53	52	51	54%	1.04
27	28	36	40	49	57	59	57	54	54%	1.08
				<u>2.62</u>		<u>30°</u>				
27	28	35	41	50	58	61	59	56	55%	1.15
24	26	34	38	46	54	56	55	53	53%	1.15
26	27	36	41	46	54	56	55	53	55%	1.07
				<u>2.62</u>		<u>22.5°</u>				
27	28	37	41	46	54	57	56	54	55%	1.08
23	25	34	38	47	54	59	57	54	55%	1.19
26	28	37	41	52	60	64	63	58	55%	1.23
				<u>2.62</u>		<u>15°</u>				
25	27	37	41	51	59	64	62	57	52%	1.30
24	25	35	38	46	54	59	57	53	53%	1.18
25	26	31	35	41	46	51	50	48	53%	.952
				<u>2.62</u>		<u>7.5°</u>				
24	25	34	36	44	51	55	54	52	51%	1.18
25	27	37	41	52	60	64	63	60	56%	1.32
				<u>2.62</u>		<u>0°</u>				
24	25	32	36	43	49	53	53	51	51%	1.18
26	28	40	42	54	61	65	64	61	57%	1.31

T_a	T_i	G_1	G_2	S_1	S_2	S_3	S_4	T_o	$n\%$	I
				<u>Flow 1.94</u>		<u>Incidence 45°</u>				
26	27	31	35	42	44	54	53	48	49%	.718
				<u>1.94</u>		<u>37.5</u>				
26	27	32	37	46	51	59	58	52	48%	.872
				<u>1.94</u>		<u>30°</u>				
26	27	34	38	49	53	59	57	54	47%	.958
				<u>1.94</u>		<u>22.5°</u>				
26	27	35	39	51	58	62	61	58	49%	1.05
				<u>1.94</u>		<u>15°</u>				
27	28	36	41	54	61	66	64	60	48%	1.114
				<u>1.94</u>		<u>7.5°</u>				
27	28	36	41	57	65	68	66	61	48%	1.148
				<u>1.94</u>		<u>0°</u>				
27	28	37	43	61	69	72	69	63	48%	1.22

An examination of this data shows that the insert improved the collector efficiency considerably, especially by making it more consistent over the range of incidence and insolation. A look at the higher levels of insolation generally, will reveal highly distorted temperature profiles for both flow rates with various screen temperatures being anywhere from 5° to 9° above the outlet temperature. This implies that a serious natural convective disruption of the desired collector temperature field is occurring. This conclusion is further borne out by the data from the thermistor placed at G_{p1} . From the diagram on page 38, it is apparent that all of the other functional thermistors are placed at slightly above the center of the collector. This thermistor was placed roughly one quarter of the way from the bottom. This was for convenience; I expected the perforated glazings to suppress natural convection. As it turned out, this thermistor consistently read about 3° lower than the corresponding G_2 temperature in every test. This makes it almost certain that convection was not suppressed in the least by the perforated glazings.

Perhaps the above statement was slightly exaggerated, because the average temperatures of the glazings, in relation to ambient, are somewhat reduced in the presence of the insert. It is becoming clear, however, that a temperature field extending from top to bottom at the front of the collector exists during operation. This will subsequently be measured. Unfortunately, a lack of extra working thermistors, and a considerable waiting period between their ordering and their receipt, has prevented me from including this measurement in the report results.

It is of interest to note that the efficiency of the low flow operation was improved by the addition of the insert considerably more than was the high flow operation. The data shows why: in the case of the high flow region the insert conserved just enough energy to put operation into the transition region by increasing the interior collector temperatures. The low flow operation was already in the transition region, so the radiative recapture and impediment to convection presented by the insert had a more noticeable effect on performance.

Without a detailed knowledge of the temperatures at various points of the cover glazings, a more detailed analysis of loss mechanisms is of little value. Calculations show, in a crude way, that the back and sides of the collector cabinet, which are not truly adequately insulated, account for about 40% of the total losses, and that the cover must account for the rest, or about 50 BTU/foot²-hr on the average. This assumes an insolation of 220 BTU/foot²-hr, of which 180 reach the absorber. At roughly 50% overall(not absorber) efficiency, the absorber will deliver 110 incident BTU and reject 70 per 180 delivered. At the insulation levels and average temperatures estimated from the data, the total heat rejection of the collector back and sidewalls will be about 30 BTU per hour per square foot of absorber area, at a crude estimate. The small size of the collector, and its relatively poor insulation, is again a disadvantage. As a test unit, however, this poor heat retention is of little importance. It is the internal heat and mass transfer associated with the absorber which are the object of study.

Rejection of 50 BTU/foot²-hr is not difficult at an average 15 to 20 °F difference between the outer glazing and ambient. More difficult is the transfer of this energy between the glazings at only an average 5° C difference. Another 40 BTU/hr-foot² must be accounted for due to the incident radiation which does not penetrate to the absorber; unfortunately no data is available on the percentage division between reflection and absorption which go to make up the extinction coefficient for the Flexigard product. Assuming only 50 BTU/hr-foot² heat flux, a uniform 5° temperature difference will account for a radiative flux of roughly 40 watt/meter² at the average temperatures in question, or about 13 BTU/hr-foot².

This implies a convective U value of 2.7, or an R value for the double glazing of 0.37, which is much too low. The solution must lie in some form of temperature gradient at the glazings, which distorts the significance of the thermistor data. The author regrets not having become aware of this problem and its ramification in time to prepare a more sophisticated test series to be included in this report.

A few heated inlet tests were performed on the remodeled collector. In these tests, the natural convective cell appears to have been partially, but not fully, suppressed by some mechanism, with consequent lessened temperature differential across the screens. Data for these tests includes the contribution of the porous glazing sensor. Due to the partial suppression of convection, this data reads more like it "should."

T_a	T_i	G_1	G_2	G_{pl}	S_1	S_2	S_3	S_4	T_o	$n\%$	I
					<u>Flow 1.94</u>		<u>Incidence 30°</u>				
20	40	36	43	50	55	59	63	67	63	37%	1.026
20	36	31	34	41	45	47	50	51	50	38%	.606
					<u>1.94</u>		<u>15°</u>				
19	35	28	32	37	42	45	49	50	49	42%	.550
20	40	34	42	49	56	61	66	69	64	38%	1.056
					<u>1.94</u>		<u>0°</u>				
20	36	35	39	45	56	61	66	64	58	34%	1.08
19	35	31	34	41	47	51	54	63	50	39%	.640

At a mild level of inlet heating, the collector efficiency is reasonable. Of more interest is the temperature profile in the screen stack. Natural convective cell formation still appears to be happening, but its behavior is erratic. It must be concluded that the response of the screen stack absorber, even with the porous transparent modification, is somewhat unpredictable. Efficiency is being lost by a mechanism unsuspected at the beginning of the tests. Once the exceptionally low pressure drop associated with screen stack operation is understood, however, the formation of competing natural convective flow paths in the collector becomes easy to understand.

Assume an air distribution in the collector consistent with perfect forced

convection, which would practically result in two triangular columns of air, each 4 feet tall in the test collector, and at about 30° C temperature difference. This is roughly a 10% density difference at the temperatures in question or, at an average .073 lb per cubic foot, a difference of 0.0073 lb per cubic foot. To find the maximum buoyant force this figure is multiplied by 4, giving 0.03 lbf per foot², or .006 inch H_2O . This figure is in all likelihood a good deal greater than the forced convective pressure drop across the screen stack, and makes the competition from natural convection inevitable.

The major conclusion of the study, therefore, is that, while the porous absorber is the optimum solar airheater absorber design, the woven screen stack is not a very good embodiment of the concept, and sacrifices a great deal of the potential efficiency of more refined absorber concepts. The problem of radiative recapture can apparently be relatively simply solved with crudely perforated transparent plastic elements, at least to some extent, but the slightly higher temperatures resulting from this efficiency improvement exacerbate the problem of natural convective degradation of the collector temperature field. The woven screen stack therefore does not represent an optimum design. Corrective measures, such as special baffles, could be employed, but it is doubtless easier in practice to abandon the screen stack and resort to any of several more sophisticated designs of porous throughflowing absorber.

This does not imply that the screen stack is entirely inadequate; the efficiencies delivered by the test collector, particularly in modified form, are quite creditable. Better insulation would improve these figures, making an operating cycle efficiency of well over 50% not inconceivable. The question at hand is, however, if this option is worthwhile in light of the existence of a better one.

The plastic layered type of porous absorber design appears to the author, from both theoretical and practical considerations, to be worth a good deal of further investigation. Its design admits of great flexibility. Very importantly, it can overcome both difficulties of the screen stack simultaneously. Radiation recapture can be accomplished by one or more perforated clear layers of high infrared opacity. Natural convection suppression can be simply accomplished by making the design pressure drop in forced convection much higher than the natural buoyant pressure differential; a forced convective pressure drop of 0.05 inch H_2O through the absorber would suffice for this. The horsepower requirement to overcome this resistance is negligible. Ductwork will represent a far greater

penalty in practice.

The work done under the auspices of this project has, in summary, first developed a method for testing the heat transfer characteristics of porous materials in forced convection at very low bulk air flow rates. With refinement, this method promises reasonable accuracy, and may have uses in pure and applied research, especially in the solar energy field. The method was applied to woven fabric screens, in order to evaluate their potential series array as a solar collector airheating absorber element. Screens were then tested in a stack, and collector efficiency was found to be reasonable, but not outstanding. Placement of a pair of perforated, transparent glazings in the flow path prior to the optically dense screens improved performance considerably by aiding in radiative recapture, and possibly in the suppression of natural convection as well. The experience gained in the course of the project finally led to the invention of a design of highly efficient and flexibly applied porous absorber, which can be produced at low cost in large volumes. This invention, in the author's opinion, is the most valuable outcome of the project.

Future Plans for the Porous Absorber

My plans for the future development of the porous absorber concept, particularly of the recently designed multi-layer plastic concept, are to attempt, first of all, to either find the type of perforated plastic material needed on the market, or to try to devise an efficient way to fabricate it in small quantities. I will then assemble an absorber of the proposed type in crude form, and test it in the collector box available. Efficiency will be determined as per above, and thermistors will be purchased and calibrated to perform the same functions as before.

I hope to get some impressive data out of these tests, such as a proven 65% or better operating efficiency under actual winter operating conditions. These results will help to commercialize the design. At the same time I carry out these tests, I plan to search the literature to try to determine originality and grounds for patentability of the design.

If the design appears to be patentable, I will file application. If results warrant, I will submit a discussion of the design concept for publication in the open technical literature, whether or not the design is patentable. If a patent application is accepted, and a professional patent search shows the possibility of infringement to be very small, I will then attempt to commercialize the design. My intentions with regard to commercialization at this time are uncertain.

It is likely that the development of the product would call for considerable sophistication, and this could imply the need for the resources of a large corporation, such as 3-M corporation. The other alternative is to adopt a policy of liberal, non-exclusive patent licensing through an intermediary organization. At any rate, the new design must prove itself in practice before this question becomes of any real importance.

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