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MASTER

THE IMPROVEMENT OF SOLAR AIR COLLECTORS -
STUDY AND EXPERIMENTAL RESEARCH PROJECT

FINAL REPORT

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FOREWORD

This is the final report that describes the work performed by Solaron Corporation under contract with the Department of Energy, Contract No. EY-76-C-04-3713, entitled "The Improvement of Solar Air Collectors".

The authors, on behalf of the Solaron Corporation, would like to express their appreciation to those in the Department of Energy who made this development project possible.

ABSTRACT

The work that ultimately lead to the design of an improved air-heating absorber plate was organized as follows: The first step in the study involved a thorough review of literature and patents relevant to solar air-heating collectors. Based on this review, the ideas that fit within the following design constraints and appeared to potentially be a mass-producible item, were compared analytically.

Design Constraints

1. The design must represent a cost-effective improvement over the baseline design, i.e., the end result of any improvement must be reduced energy cost to the consumer.
2. The pressure loss through the collector (ΔP) must be maintained at approximately the same level as the baseline collector for readily compared results.

$$0.08 < \Delta P < 0.17 \text{ cm. w.g. per meter of collector length}$$

$$(0.01 < \Delta P < 0.02 \text{ in. w.g. per ft. of collector length})$$

3. A stagnant air space between the absorber and inner glazing is desirable. This requirement is due to the reduced transmittance that will likely occur as a result of having an air stream in contact with the glass as well as the improved thermal performance of this arrangement in cold climates.

The analysis considered the heat transfer and pressure loss characteristics of the various designs. As a result of this engineering analysis, the second step in the program required that three independent improvement concepts be tested experimentally and compared with the baseline design, a Solaron Series 2000 collector.

The designs selected were:

1. A staggered arrangement of dimples.
2. An absorber with extended surface area (finned absorber).
3. An absorber with periodic boundary layer interruptions.

These absorbers along with the baseline design were mounted in identical collector bodies and subjected to thermal efficiency tests on Solaron's collector test stand. The testing was done simultaneously on the four panels in a side-by-side collector test. Unfortunately, the measured pressure losses varied from the baseline value. This necessitated an analytical adjustment of the thermal performance to that which was expected at the baseline pressure loss. The adjusted test results from the first test series indicated that the absorber with extended surface had the best potential as a mass-produced, improved design. It performed 1.11 times more efficiently than the baseline design, i.e., the heat removal factor (F_R) was 1.11 times larger.

The third step in the program called for the selection of one of the three concepts to be fabricated and tested in a 2 X 2 collector array.

Various manufacturing techniques for producing an absorber with extended heat transfer surface were investigated. It was found that fins could be attached to the absorber in a cost-effective manner by an electrical resistance weld. Four finned absorbers were fabricated by resistance welding and installed in a 2 X 2 collector array to verify that these production model prototypes would, in fact, increase performance by a factor of 1.11. Flow duct dimensions were adjusted to achieve the baseline pressure drop. The test results showed that at $(t_i - t_{amb})/H = 0$, the improved absorber delivered 1.11 times the energy of the baseline design. At $(t_i - t_{amb})/H = 0.04 \text{ Cm}^2/\text{w}$ ($0.23 \text{ hr. ft.}^2/\text{Btu}$), the improved absorber delivered 1.23 times the energy.

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SECTION I

INTRODUCTION

The rate of heat exchange between the coolant air and the absorber plate with which it is in contact is a major factor affecting the efficiency of flat plate air-heating collectors. An increase in the heat transfer rate between the absorber and air stream results in more energy collected per unit of collector area. It is not sufficient, however, to merely increase the collector efficiency, as the economic aspect of the problem requires that the maximum amount of energy be delivered for the least cost. The economic constraint may require less efficient absorbers to be used if the delivered energy is less costly than that from more efficient absorber plates. The goal of the present development project is to reduce the cost of energy delivered by flat plate air-heating collectors, if possible, through improved heat transfer between the absorber and air flow.

The procedure that has been followed for evaluating the cost-effectiveness of various absorber improvements is outlined below:

1. Review of literature and applicable patents - The purpose of this review is to compile concepts, inventions and the numerous flat plate, air-heating collector designs that have been discussed in the literature over the years.
2. Through engineering analysis the various concepts that appear to have cost reduction potential are compared.
3. Three designs that showed potential as mass-produced absorbers were tested and compared in a side-by-side thermal efficiency test with a Series 2000 Solaron collector used as the performance baseline.
4. Based on the test results (of step (3)) which were adjusted analytically to account for differences in pressure drop, the economics of the improvements were evaluated a "budget", i.e., the additional cost that was justifiable at the break even point, was determined.

5. A matrix of manufacturing processes that could be used to produce the final design was evaluated. Cost estimates for the various processes were made.
6. The most practical and cost effective manufacturing process was chosen and four prototypes were built and tested in a typical array configuration.

The report is divided into seven sections and three Appendices. Section II contains summaries of numerous technical articles that pertain to solar air-heating collectors. Relevant patents are reviewed in this section also. Section III covers the engineering analysis and the theoretical considerations that pertain to air-heating collectors. Section IV contains the test results from the first phase of testing, Section V covers the economic analysis of the problem. The results of the second phase of testing are given in Section VI. Section VII gives the conclusions and recommendations. Appendix A contains the details and backup materials that pertain to Section III. Appendix B lists the data collected for both the first and second series of tests. Nomenclatures and references are given in Appendix C.

SECTION II

LITERATURE SEARCH AND PATENT REVIEW

The object of this review is to compile in one publication a survey of all the designs, patents and concepts that relate to solar air-heating apparatus. The primary journal in which solar research has been reported, is the technical journal of the International Solar Energy Society, entitled Solar Energy. This journal was reviewed from its inception (1957) through the most current issue, No. 5, 1977. Other journals and technical papers were reviewed. These included appropriate papers published in the transactions of the ASME Journal of Engineering for Power, the Journal of Heat Transfer, the ASHRAE Journal, and the ASHRAE Handbook and Product Directory. The published abstracts and proceedings of solar energy conferences were reviewed for relevant work. An extensive bibliography published by the U.S. Department of Commerce, entitled, Solar Energy - A Bibliography, was also reviewed for pertinent work relating to flat-plate air heaters. Additionally, solar patents that were available from 1855 through 1976 were reviewed.

Most of the work that has been reported in the literature, has been done in the past thirty years. A standardized collector test has only recently been developed and used by researchers. Consequently there is some difficulty in making quantitative comparisons and/or judgements of the different collector designs, based on the reported efficiencies. In some instances the important variables of inlet air temperature, insolation and ambient temperature are not discussed or mentioned in the work. With these considerations in mind it should be understood that this survey is simply a review of what the individual authors have chosen to publish. No attempt to editorialize was made by the reviewer.

Many published papers that were not included in this review concerned themselves primarily with various applications of air-heaters and the associated systems. Since this review is concerned primarily with collector design, these papers were not included in the review.

The summaries are grouped in the following categories:

- A. Review of journals, engineering literature.
- B. Review of textbooks, theses, handbooks, unpublished papers, abstracts.
- C. Review of Patents.

A. Review of Journals, Engineering Literature

Author: Bevill, V.D., Brandt, H.

Title: "A Solar Energy Collector for Heating Air"

Publication: Solar Energy, Vol. 12, pp. 19-29, 1968

Type of Work: Experimental

Abstract: This paper presents the results of an experimental study on a flat plate air heater that incorporates fins on the solar-exposed side of the absorber plate. The object of the study is to determine the efficiency of a collector which utilizes fins that reflect the incident solar radiation in a specular manner (Figure 2-1). These results are compared to the efficiency of a similar collector, with fins that reflect diffusely. Efficiencies of more than 80 percent were reported for the collector using specular reflecting fins. When the fins were made to reflect diffusely, the efficiency decreased by about 15 percent. The experimental data were taken at different times of the year. A detailed discussion of all the test conditions was not included in the paper.

Conclusions: An absorber with specular reflecting fins operates approximately 15 percent more efficiently than one with diffusely reflecting fins.

Figure 2-1
Specularly Reflecting Finned Absorber

Authors: Charters, W.W.S., Lim, T.M., Tong, Y.C.

Title: "The Design Evaluation of Flat Plate Solar Air Heater Absorbers by Digital Computer"

Publication: Mechanical and Chemical Engineering Transactions, I.E. Australia, Vol. MC8, No. 1, pp. 55-58, May, 1972

Type of Work: Analytical

Abstract: The location of the absorber plate in the flow duct of single glazed flat plate air-heaters is analyzed using a numerical model. The following designs are considered: Type I - air flow above the absorber, Type II - air flow below the absorber, and Type III - air flow both above and below the absorber (Figure 2-3). The results indicate that the design choice is dependent on application, i.e., collector length.

Conclusions: (1) "The numerical technique employed has shown the worth of a detailed feasibility study for any major solar flat plate collector to optimize as far as possible the basic design.

(2) The stagnant air gap principle employed in the Type II heater is an effective system for stabilizing the glass cover plate temperature, and should be given serious consideration for all long flow ducts.

(3) Careful calculations for a particular installation should help to guide the designer to the choice of a Type I, II or a Type III unit."

Figure 2-2

Air-Heaters Classified by Flow Channel Location

Authors: Charters, W.W.S., Macdonald, R.W.G.

Title: "Heat Transfer Effects in Solar Air Heaters"

Publication: Proceedings of the ISES, Paris Conference, July, 1973

Type of Work: Analytical

Abstract: A review is made of two solar flat-plate air-heater designs. The advantages and disadvantages of stagnant-gap and double-channel flow type collectors are discussed. A comparison is made of flat-plates and "vee" corrugated plates on the basis of heat transfer and material utilization. The results of the study show that the longer the flow duct the greater the need for a stagnant air gap between the absorber and cover plate. One plot for the flat-plate type collector, suggests that for the design considered, the collector length where a stagnant gap becomes advantageous over the double channel type collector is 3 meters. It is pointed out that "vee" corrugations probably enhance

natural convection between the absorber and cover glass, and therefore, may be put to greater advantage in a double channel type collector. The effect of the angle of the "vee" opening on the natural convection heat transfer was not determined in this study.

Conclusions: (1) Since there are numerous design parameters involved in flat-plate collectors it is difficult to generalize on optimum designs.

Author: Charters, W.W.S.

Title: "Some Aspects of Flow Duct Design for Solar Air-Heater Applications"

Publication: Solar Energy, Vol. 13, pp. 283-288, 1971

Type of Work: Analytical

Abstract: In a flat plate solar air-heater provision must be made for passage of the air in direct contact with the absorber surface. This paper discusses the shape (Figure 2-3) of rectangular cross-sectional flow passages from the stand point of cost and performance. A design procedure based on technical and economic considerations is presented.

The important parameters are related by the expression

$$a^{0.8}(1+a)^{0.2} = \frac{Q}{Lk_1} \left(\frac{\mu b}{2\dot{m}}\right)^{0.8}$$

It is possible to select a series of design aspect ratios by choosing values for b and $\dot{m}$, for a fixed Q and L . If the collector is divided into three segments; absorber plate costing x cents/unit area, cover plate costing y cents/unit area, and bottom and side walls costing z cents/unit area, the total channel cost is:

$$C = (bL)x + (bL)y + (2a+b)Lz.$$

If $A = ab$ then $\frac{dC}{db} = 0$ to give:

$$x + y + z = \frac{2Az}{b^2} \quad \text{or} \quad b = \left(\frac{2Az}{z+y+z}\right)^{\frac{1}{2}}$$

Conclusions: There is no simple method for optimizing the flow duct dimensions if only the technical aspects of the problem are considered. A combined technical and economic evaluation must be made.

Figure 2-3
Optimized Flow Channel

Authors: Chiou, J.P., El-Wakil, M.M., Duffie, J.A.

Title: "A Slit-and-Expanded Aluminum Foil Matrix Solar Collector"

Publication: Solar Energy, Vol. 9, No. 2, pp. 73-80, 1965

Type of Work: Experimental, Analytical

Abstract: Porous matrices are tested as the heat-absorbing media in air-cooled solar flat-plate collectors. Data was obtained from earlier studies on the heat transfer and fluid friction characteristics of various porous matrices. An analytical model is discussed that permits performance calculations to be made for various types of matrix collectors. The six different matrices examined in this study are made of slit-and-expanded aluminum foil, the material commonly used in air filter grease traps. Several layers of the different matrices were used so that in all the tests conducted the voidage was uniform, ranging from

96.2 to 98.3 percent. From the tests it was found that a foil 0.05 mm (0.002 in.) thick, 97.4% voidage, β = extinction coefficient = 156/m (47.4/ft.) and a hydraulic radius, r_h = 0.097 cm (0.00317 ft.), will require the least amount of blower power for a given value of h_v , the volumetric heat transfer coefficient. The thickness of the matrix was set by that depth which will absorb 99.7% of the incoming radiation. In this case the thickness was 3.8 cm (1.5 inches). The flow arrangements of the matrix air-heaters are shown in Figure 2-4..

Of the three flow arrangements shown in Figure 2-4 the unidirectional flow type has the advantage of having the cooler inlet air in contact with the glass cover. Thus, there will be smaller top losses with this collector arrangement than the other types.

Conclusions: (1) The ratio of energy collected to required pumping power, is larger for a matrix type collector, than it is for the conventional parallel plate design.

Figure 2-4

Matrix Air-Heater Types

Author: Close, D.J.

Title: "Solar Air Heaters for Low and Moderate Temperature Applications"

Publication: Solar Energy, Vol. 7, No. 3, 1963

Type of Work: Analytical

Abstract: The author develops an analytical technique that allows a comparison to be made of three types of air heaters, shown in Figure 2-5. Five factors were examined to determine their effects on the collection efficiency of an air heater element. They are:

(a) absorber plate spectral reflectance characteristics (b) plate-to-air heat transfer area (c) "vee" corrugation of absorber plate (d) natural convection barrier (e) insolation. For the one cover collectors examined, those with the "vee" corrugated absorber plate, nominal selective properties $\alpha = 0.80$ $\epsilon = 0.05$, gave the best performance while those with the black painted flat absorber plates gave the poorest. The effect of a stagnant air gap is shown by comparing collector types II and III. It was found that the best type II collector has a somewhat better performance than the best type III, at air temperatures below 37.8 C (100°F). The type III air heater has a notable advantage at average air temperatures greater than 48.9 C (120°F). The effect that insolation has on performance was shown by plotting efficiency versus air temperature for two different insolation rates. It was found that the higher insolation yields higher efficiencies for all models.

Conclusions: (1) Absorbers using selective surfaces, of the optical or directional types, such as vee corrugated surface, all have enhanced thermal performance over the conventional flat-plate design. The type III heater incorporating this feature has the largest percent improvement in comparison to the types I and II.

Figure 2-5
Air-Heater Designs

Authors: Cole-Appel, B.E., Haberstroh, R.D.

Title: "Performance of Air-Cooled Flat Plate Collectors"

Publication: Proceedings of the joint American Section of ISES and the
Solar Energy Society of Canada, Inc. August 15-20, 1976,
Winnipeg

Type of work: Analytical

Abstract: An analytical scheme using a simple linear model is used in this study to compare the thermal performance of five flat plate solar air-heater designs. The air-heaters differ with respect to the location of the flow channel, which may be above and/or below the absorber plate. In addition two of the air-heater designs have fins added to the absorber plate. Performance is determined for the different collector designs by solving the sets of simultaneous equations obtained from the energy balances written for each of the collector components.

- Conclusions:
- (1) It was found that the efficiency, defined as the useful energy collected divided by incident radiation striking the glass, of a flat plate collector can be increased by approximately 10 percentage points (a factor of 1.2) if staggered, interrupted fins are added to the flow side of the absorber plate. This is in comparison to a plane (non-finned) absorber plate air-heater (Figure 2-6).
 - (2) An optimum fin spacing based on the fin density, which when increased does not improve performance substantially, was found to be approximately 0.4 to 0.8 fins/cm (1 to 2 fins/inch).
 - (3) The best flow channel location is dependent on operating conditions.
 - (4) It was found that the thermal performance could be enhanced slightly if radiation heat transfer between the absorber and back plate is maximized.

Figure 2-6
Interrupted Fin

Authors: Gupta, G.L., Garg, H.P.

Title: "Performance Studies on Solar Air Heaters"

Publication: Solar Energy, Vol. 11, No. 1, 1967

Type of Work: Experimental

Abstract: Four air heater designs are compared experimentally in this work. Two of the collectors (types A and B), employed corrugated absorbers and two used a mesh type absorber. The efficiencies were compared holding the blower power constant in all tests. Unequal frictional losses were taken into account by this technique. The collectors using corrugated absorbers had two sheets of metal, which formed the air flow duct. The corrugations ran perpendicular to each other and were welded together. The crests of the corrugations of one sheet touched the troughs of the other, and the heat transfer passage formed between the plates produced a pulsating flow. The mesh collectors were made using two different wire-mesh materials, one using an iron wire mesh the other using an aluminum wire-mesh. The four collectors tested are shown schematically in Figure 2-7.

Conclusions: (1) Air-heater type A has the best plate efficiency factor, F' .

(2) Air-heater type B, however, has a larger collected energy to pumping power ratio.

(3) The type D heater has an unacceptable thermal efficiency in comparison to the other designs tested.

(4) The type C collector performed nearly the same as the types A and B and in view of its ease of fabrication and lower cost, may be the most practical.

Collector Type A

Collector Type B

Collector Type C

Figure 2-7 Air-Heaters Tested by Gupta

Collector Type D

Figure 2-7
(Continued)

Air-Heaters Tested by Gupta

Authors: Hamid, Y.H., Beckman, W.A.

Title: "Performance of Air-Cooled Radiatively Heated Screen Matrices"

Publication: Journal of Engineering for Power, Transactions of the ASME, April, 1971, pp. 221-224

Type of Work: Experimental

Abstract: The problem of combined radiation and convection in a porous bed was experimentally investigated. The porous bed was made of randomly stacked wire screens. A collimated radiation source was used for bed heating and air flowing through the bed was the cooling medium. Comparisons were made with an analytical model, which included the effects of internal radiation exchange and convection between the bed and air.

Conclusions: Despite the differences that were observed between the analytical and experimental matrix temperatures, the overall efficiencies that were measured compared favorably with the analytical results.

Author: Hollands, K.G.T.

Title: "Directional Selectivity, Emittance, and Absorptance Properties of Vee Corrugated Specular Surfaces"

Publication: Solar Energy, Vol. 7, No. 3, p. 108, 1963

Type of Work: Analytical

Abstract: A directionally selective absorber surface is defined as a surface which is optically grey, yet has the characteristics of a "selective" surface. This effect is due to the hemispherical nature of the long wavelength radiation. It is shown in this paper that a vee-corrugated (Figure 2-8) specular surface is directionally selective and that the selective properties of a spectrally selective surface can be improved by vee corrugating it to make it directionally selective as well. Optimum vee-corrugation angles are discussed.

- Conclusions:
- (1) "An ideal directionally selective surface designed for year around use would have a solar absorptance for direct solar radiation of at least 97 percent, and a hemispherical emittance of 54 percent. If collection is desired only over the one-half year between equinoxes, the surface can be 97 percent absorbing and 28.4 percent emitting."
 - (2) "A vee-corrugated specular surface is directionally selective. This is because near normal radiation suffers more interreflections than radiation at high angles of incidence."
 - (3) "The selective properties of a specular spectrally-selective surface can, in almost all cases, be improved by vee corrugating to make it directionally selective as well."

(4) "A method has been given for determining the optimum angle of opening for the vee corrugations for a specific application."

(5) "The method of vee corrugating is best suited to surfaces for which there is little variation in reflectivity with respect to both wavelength over the solar spectrum and angle of incidence, and for locations where the diffuse component of solar radiation is small."

Figure 2-8

Vee-Corrugated Specularly Reflecting Absorber

Authors: Henderson, S.M., Khe, C.V.
Title: "Performance of a Matrix Solar Collector for Heating Air"
Publication: American Society of Agricultural Engineers Annual Meeting,
June 22-25, 1975
Type of Work: Experimental, Analytical

Abstract: An unglazed solar collector for heating air was constructed of black painted woven wire screens through which air was passed. The number of screens was varied between 6 and 8 and the efficiencies were measured for each of the cases. A mathematical model was developed which had good agreement with the experimental results.

- Conclusions:
- (1) The multiple layer screen absorber showed efficiencies of 48-57%. The screen matrix should be guarded from the wind to eliminate heat loss.
 - (2) Ambient temperature and total incoming radiation affect performance slightly.
 - (3) Two to four screens provide optimum efficiency, depending upon the mesh of the screen. Smaller mesh is slightly more efficient but its cost and fragility might nullify its small advantage. One screen, although 2 to 4% less efficient than the optimum number of screens, might constitute a simple inexpensive collector for many air-heating tasks.
 - (4) Rough open weave cloth of jute or other fiber could be used as the absorber of an effective inexpensive air-heater.

Author: Hewitt, H.C.
Title: "Performance Tests of a Solar Energy Collector Used to Heat Air"
Publication: Proceedings of the 1977 Annual Meeting, American Section of ISES, June 6-10, 1977, Vol. 1, Section 2
Type of Work: Experimental

Abstract: A simple flat plate air-heater using aluminum foil as the absorber was constructed and tested. Efficiencies and collector time response factors were measured.

Conclusions: Efficiencies exceeded 60 percent for air flow rates above $25 \text{ m}^3/\text{s m}^2$.

Authors: Lalude, O., Buchberg, H.
Title: "Design and Application of Honeycomb Porous-Bed Solar Air-Heaters"
Publication: Solar Energy, Vol. 13, pp. 223-242, 1971
Type of Work: Experimental, Analytical

Abstract: An air-heater with an optimally sized honeycomb structure placed between the glazing and a porous absorber plate was tested and its thermal efficiency determined. The optimum honeycomb design was found by defining a criterion function, $Z = \text{annual cost/useful energy collected}$. A cutaway view of the collector construction is shown in Figure 2-9. The second part of this work involved a design study of a solar air-heater, crop dryer system. The most cost-effective collector area, flow rate, and exit air temperature were determined.

Conclusions: (Pertaining to collector design)

(1) With cost-effectiveness as the primary criterion the following sized honeycomb structures should be used:

absorber temperature $< 82\text{ C (180F)}$	$L'/d' = 4 \text{ to } 5$
$82\text{ C (180F)} < \text{absorber temperature} < 93\text{ C (200F)}$	$L'/d' = 6 \text{ to } 8$
absorber temperature $> 104\text{ C (220F)}$	$L'/d' = 10$

(2) The optimum w'/d' ratio is a weak function of absorber temperature. $Z_{\min}$ occurs at $w'/d' = 7$.

(3) Z does not vary from $Z_{\min}$ for, δ , the cell wall overcoat thickness over the following range:

$0.01 < \delta < 0.02 \text{ mm}$
($0.4 < \delta < 0.8 \text{ mil.}$)

Figure 2-9 Honeycomb Porous Bed Air-Heater

Author: "Lof, G.O.G.
Title: "Performance of Solar Energy Collectors of Overlapped-Glass-Plate Type"
Publication: MIT Solar Heating Symposium (1950) Open Space Heating with Solar Energy, MIT, 1954
Type of Work: Experimental

Abstract: All-day thermal efficiencies of the overlapped-glass plate collector (Figure 2-10) were measured in an attempt to simulate operating conditions in a residential-heating application. The efficiencies varied from 30 to 65 percent depending on the operating condition. The exit air temperature ranged from 63 C (145F) at 9:00 AM to 93 C (200F) at noon with a flow rate of $0.008 \text{ m}^3/\text{sec m}^2$ (1.6 cfm/sq. ft.) of collector. Studies of the relation between entering air temperature and heat recovery efficiency showed that as the entering air temperature is increased the efficiency decreased. With inlet air temperatures of 26.7 C (80F), the over-all daily efficiency was found to be approximately 50 percent.

Conclusions: Studies of the performance of solar energy collectors of the overlapped-glass-plate type have yielded the following general results:

"(1) A solar energy collector unit has been developed which should be producible on a factory basis and suitable for installation in a new dwelling without extensive fabrication at the site;

(2) with a collector consisting of a multiple of the individual units, heated air ranging in average temperature from 150° to 200°F at solar heat recovery efficiencies from 60 to 40 percent, respectively, can be delivered to a dwelling on sunny winter days at 40° latitude (single cover-plates, untreated glass);

(3) average daily summer performance of the collector system employing air recirculation yields hot air at 210°F with an efficiency of 20 percent. Excluding early morning and the late afternoon hours, mean temperatures of 240°F are obtained (single cover-plates, untreated glass);

(4) certain improvements in construction, such as the use of surface-treated glass and multiple cover-plates, result in performance considerably exceeding that obtained with the simpler construction;

(5) procurement of air at temperatures exceeding 200°F makes possible the operation of an air-conditioning unit with solar energy. Of apparently the most direct application is a refrigeration system of the heat-operated absorption type or a dehumidification system employing heated air for regeneration of the absorbent;

(6) the solar heating system in its present form has been incorporated into the design of a dwelling and preliminary estimates of operating economies have been made. Appreciable refinement of such estimates will require construction and use of actual dwellings;

(7) certain features of construction and operation of the collector units have been satisfactorily devised, but a number of additional requirements will need further study and eventual testing in a dwelling."

Figure 2-10

Overlapped Glass Plate Collector Modules
Cross Sectional View

Authors: Lof, G.O.G., El-Wakil, M.M., Chiou, J.P.

Title: "Design and Performance of Domestic Heating System
Employing Solar Heated Air - The Colorado Solar Home"

Publication: New Sources of Energy, Proceedings of the United Nations
Conference on New Sources of Energy, Rome, Italy, August
21-31, 1961, Volume 5.

Type of Work: Experimental

Abstract: The overlapped-glass plate collector is described in some detail in this paper.

"Each panel has six black plates 12 x 27 inches, and seven clear glass plates, five of which are 24 x 27 and two are 12 x 27 inches. They are common single strength window glass (about 1/10 inch thick), spaced 1/4 inch apart. The black plates were made by thermal fusion of black glass particles to the surface of clear glass." The collector array is divided into two banks. The first bank is an air preheater array, which uses single glazed collectors. The air leaves the first collector array and flows to a second bank of collectors which is double glazed to minimize top thermal losses. The collector efficiency from January through April of 1960 averaged 38.5 percent. The required measurements necessary to compute system performance were also measured, recorded and presented in this paper.

Conclusions: (1) This collector was fabricated on the job site.
Based on this experience it is suggested that from a quality control viewpoint, it would be more efficient to have preassembled units ready for installation.

(2) The overlapped-glass plate air-heater can be used to provide any practical percent of the total heating requirements of a building.

Author: Lof, G.O.G.
Title: "Solar Air-Heaters"
Publication: Proceeding of the Solar Heating and Cooling for Buildings
Workshop, Washington, D.C., March 21-23, 1973
Type of Work: Experimental

Abstract: A summary of the performance of the overlapped-glass air-heating collector (Figure 2-9) installed on Dr. G.O.G. Lof's home in Denver, Colorado is presented in this paper. Efficiencies for two typical operating days are noted. Of particular interest is the significant increase in collector performance due to the low reflective glass used on some of the collector modules.

Conclusions: (1) Flat-plate air-heaters can be made to perform as well, or better than, liquid flat-plate collectors.

(2) Low reflective glass can improve the performance of flat-plate collectors significantly.

Author: Loth, J.L.

Title: "A Grooved Foamglas* Flat-Plate Solar Collector"

Publication: Proceedings: 1977 Flat-Plate Solar Collector Conference,
February 28 - March 2, 1977, Orlando, Florida

Type of Work: Experimental, Analytical

Abstract: The construction and performance of a V-grooved, air-heater is described. The unique feature of the absorber is that it was fabricated from Foamglas (Figure 2-11). The V grooves were formed with a 30° included angle and then painted black. The temperature difference between the air flow and absorber was reduced substantially compared to the conventional flat-plate design. Test results indicate comparable performance in comparison to the referenced water-cooled collector.

Conclusions: (1) Foamglas appears to be a relatively inexpensive, stable, structurally sound material for use as an absorber surface in flat plate collectors.

(2) Foamglas has a relatively fast time response.

(3) Conductive edge loss should be reduced by this design.

Figure 2-11

Foamglas Collector

*Trade Name

Author: Parker, B.F.

Title: "A Solar Panel for Residential Use"

Publication: Proceedings of the 1977 Annual Meeting, American Section
of ISES, June 6-10, 1977, Vol. 1, Section 2

Type of Work: Experimental

Abstract: A vee-corrugated absorber is incorporated in this integral roof air-heater design. Collector efficiencies were measured and compared to a flat plate design. The vee corrugated absorbers showed an improved efficiency.

Conclusions: The solar panel design discussed should provide a relatively efficient, roof-integrated air collector.

Authors: Rankine, A.D. and Charters, W.W.S.

Title: "Combined Convective and Radiative Heat Losses from Flat-Plate Solar Air-Heaters"

Publication: Solar Energy, Vol. 12, pp. 517-523, 1969

Type of Work: Analytical, Experimental

Abstract: An experimental rig was designed to simulate the top loss mechanisms in a flat-plate collector. The experiment was designed so that the convective and radiative losses could be isolated by evacuating the space between the heated surface and the glass cover. If the collector was perfectly insulated on the back side and the edges, then accurate measurements of the top losses would provide closure to the energy balance, assuming that the energy input was known.

Conclusions: An alternative to computing the energy added to the airstream, which is sometimes a difficult task, is to calculate the thermal losses. By balancing the energy equation, the net energy added to the air stream can be computed.

Author: Rask, D.R., Mueller, L.J., Pejsa, J.H.
Title: "Low-Cost Solar Air Heater"
Publication: Final Report, ERDA Contract No. EY-76-C-02-2929
Type of Work: Experimental

Abstract: The results of the development work on the jet impingement air-heater are discussed in this report. A comparison was made between a conventional parallel plate design (baseline) and the jet impingement collector (Figure 2-12). A cost-performance analysis was performed comparing the installed costs of the baseline system (parallel plate) at \$323/sq. m (\$30/sq. ft.) and the final design of the jet-plate collector at \$407/sq. m (\$37.84/sq.ft.). The system pay back periods were 28.5 and 29.3 years, respectively. However, the energy delivered to the space for each system was 293 Kw hr./sq. m yr. (93,142 Btu/Sq. ft.yr.) and 350 Kw hr./Sq. m yr. (111,331 Btu/Sq.ft.yr.), respectively. Note that the final jet-plate design incorporated improvements other than the perforated plate, i.e., a black chrome absorber coating, and etched glass for reduced transmission losses.

Conclusions: (1) Comparing the baseline collector and jet-plate collector at equal pressure drops, the jet-plate design had a 13% improvement in efficiency over the baseline parallel plate collector. (plotted on $(t_{ave} - t_a)/H$.) Note that a 13% improvement in F' does not necessarily result in a 13% improvement in F_R .

(2) The final design jet-plate collector Y-intercept efficiency was 13 percent better than that of the baseline collector and 32 percent better at a typical space heating $\Delta T/H = 0.04 \text{ C m}^2/\text{W}$. (0.23 hr. ft.² F/Btu).

(3) The following jet-plate design is recommended:

(a) jet spacing/diameter = $X/D = 13$ to 14

$D = 0.51$ to 0.64 cm (0.2 to 0.25 inch)

(b) jet-to-absorber spacing = $Z/D = 2$ to 6

(c) a perforated plate is to be used for the jet-plate

(d) a flow rate equal to $0.012 \text{ m}^3/\text{s m}^2$ ($2.45 \text{ scfm/sq. ft.}$) of aperture

(4) A 10 percent increase in overall collector efficiency results in a 6.9 percent increase in energy delivered to space and domestic hot water heating, based on system simulation.

(a)

(b)

Figure 2-12

(a) Schematic of Jet-Impingement Collector

(b) Schematic of Parallel Plate Collector

Authors: Satcunanathan, S., Deonarine, S.

Title: "A Two-Pass Solar Air Heater"

Publication: Solar Energy, Vol. 15, pp. 41-49, May, 1973

Type of Work: Experimental

Abstract: The thermal performance of a solar collector is strongly dependent on the losses from the absorber plate to the surroundings. The top losses from a two cover air-heater can be reduced, by first passing the air through the space between the covers and subsequently flowing the air in contact with the blackened absorber plate. (Two-pass air-heater, Figure 2-13) It was found that the outer glass temperature was reduced when operated in this manner.

Conclusions: (1) The two-pass air heater was found to give a 10-15 percent increase in efficiency in comparison to the conventional single pass collector, with no increase in cost.

(2) The spacing between cover glazings affects the performance of the two-pass collector. The smaller the spacing the greater the improvement. However, there is an optimum spacing for specified operating conditions.

Figure 2-13

- (a) Section A-A
- (b) Two Pass Collector
- (c) Single Pass Collector

Author: Selcuk, K.

Title: "Thermal and Economic Analysis of the Overlapped-Glass Plate Solar Air-Heater"

Publication: Solar Energy, Vol. 13, pp. 165-191, 1971

Type of Work: Experimental, analytical

Abstract: An experimental and analytical study of the Lof-type overlapped-glass plate solar air-heater is presented in this paper. A computer model was used to predict the thermal performance of the air-heater as well as make an economic analysis of the solar collector system. Experiments were then made to verify the model. Excellent agreement between the model and experiment was found. A sketch of this collector is shown schematically in Figure 2-14.

Conclusions; (1) Collector efficiencies were found to be a function of the mass flow rate. At an air velocity of 1 m/sec (0.305 ft./sec.) the efficiency was near 75 percent, whereas at 1/3 of this velocity the efficiency is only 60 percent. This is of course a function of factors such as inlet air temperature and the amount of insolation.

(2) The economic analysis is valid only for the assumed labor costs, expected life, and associated costs.

Figure 2-14
Overlapped Glass Plate Collector

Author: Shoemaker, M.J.

Title: "Notes on a Solar Collector with Unique Air Permeable Media"

Publication: Solar Energy, Vol. No. 4, 1961, p. 138

Type of Work: Experimental

Abstract: Slit-and-expanded aluminum foil is discussed as an absorber material for use in matrix type air-heaters. The absorptance as a function of the number of layers of material is given. Some thermal performance measurements were also made but no thermal efficiency calculations were presented.

Conclusions: No conclusions were discussed.

Authors: Sobel, A.T., Buelow, F.H.

Title: "Galvanized Steel Roof Construction for Solar Heating"

Publication: Agricultural Engineering, June, 1973, pp. 312-317

Type of Work: Experimental

Abstract: Galvanized steel roofs can be used as an absorber plate for agricultural process air-heating applications. In this study the collectors were constructed by simply adding a metal sheet behind the roof to form a duct through which air was blown. The inlet and outlet temperatures were measured and the efficiency was computed. The duct spacing was varied to determine its effect on collector performance. All test runs were made at a flow rate of $0.05 \text{ m}^3/\text{m}^2 \text{ sec}$ (10 cfm/sq. ft.). A goal of this study was to determine the resulting improvement in thermal performance when the air flow was at right angles to the corrugations rather than parallel to the corrugations.

Conclusions: (1) Either the 7.6 or 10 cm (3 or 4 inch) duct spacing gave acceptable efficiencies. Duct spacings smaller than 7.6 cm (3 inches) proved to have excessively large pressure drops associated with them.

(2) Approximately ten percent improvement in efficiency was gained by flowing the air perpendicular to the corrugations in comparison to the parallel flow arrangement.

(3) The corrugated absorber with flow parallel to the corrugations provided a ten percent improvement over the flat sheet absorber.

Authors: Suri, R.K., Saini, J.S.

Title: "Performance Prediction of Single and Double Exposure Solar Air Heaters"

Publication: Solar Energy, Vol. 12, pp. 525-530, 1969

Type of Work: Analytical

Abstract: The "single-exposure" collector is a conventional single glazed air-heater. The flow channel is below the absorber plate (Figure 2-15). The "double exposure" air-heater incorporates two absorber plates; both exposed to solar radiation. The "double exposure" collector is oriented in such a manner that a reflector is placed causing the underside absorber to be heated from the reflected radiation. The overall effect of the double exposure collector is to increase the energy that is added to the air stream.

Conclusions: (1) The calculations show that the "double exposure" collector to be more efficient than the "single exposure" collector; assuming that the reflectors can be suitably located. A precise estimate on the cost-effectiveness of the improvement was not included in the analysis.

Figure 2-15

(a) Double Exposure Collector

(b) Single Exposure Collector

Authors: Tan, H.M., Charters, W.W.S.

Title: "An Experimental Investigation of Forced Convective Heat Transfer for Fully-Developed Turbulent Flow in a Rectangular Duct with Asymmetric Heating"

Publications: Solar Energy, Vol. 13, pp. 121-125, 1970

Type of Work: Heat transfer, experimental

Abstract: A test was designed to simulate a solar-flat plate collector, which uses air as the heat transfer medium. The purpose of the study was to correlate Nusselt number with Reynolds number over the range $9500 < Re < 22000$. An electrically heated plate was substituted for the absorber plate so that the heat flux could be easily controlled and kept constant during the steady-state tests. The test results were as expected. The bulk air temperature increased linearly along the duct and the wall and air temperatures had equal gradients after 30 diameters. Based on this investigation it is recommended that the following correlation be used for determining the heat transfer in an asymmetrically heated rectangular duct.

$$Nu_{\infty} = 0.018 Re^{0.8} Pr^{0.4}$$

- Conclusions:
- (1) The curves of local Nu number against L/D_H exhibit the trends observed for turbulent forced-convective heat transfer in round pipes.
 - (2) For $9500 < Re < 22000$ the development length was found to be 70 diameters.
 - (3) The results of this study agree with earlier work on heat transfer characteristics in asymmetrically heated ducts.

Authors: Tan, H.M., Charters, W.W.S.

Title: "Effects of Thermal Entrance Region on Turbulent Forced-Convective Heat Transfer for an Asymmetrically Heated Rectangular Duct with Uniform Heat Flux"

Publication: Solar Energy, Vol. 12, pp. 513-516, 1969

Type of Work: Heat transfer, experimental

Abstract: A test rig was designed to simulate a solar flat-plate air heater. The purpose of the research was to develop a correlation for heat transfer coefficients in the entrance region of an asymmetrically heated rectangular flow channel. The data was plotted and curve fit for the mean test results. The mean curve is well represented by the following expression:

$$Nu_{ave} = Nu_{\infty} (1 + S D_H / L) \quad \text{for } 9500 < Re < 22,000$$

$$\text{Where: } S = 14.3 \log \left(\frac{L}{D_H} \right) - 7.9 \quad \text{for } L/D_H \leq 60$$

$$S = 17.5 \quad \text{for } L/D_H > 60$$

Conclusions: (1) The above correlation should provide a good estimate of the heat transfer coefficients in the entrance region of a flat-plate air-heater. Since the entrance region can be as long as 60 diameters, this effect cannot be ignored.

Authors: Ward, J.C., Lof, G.O.G.

Title: "Long-Term (18 years) Performance of a Residential Solar Heating System"

Publication: Solar Energy, Vol. 18, No. 4, 1976

Type of Work: Experimental

Abstract: The Colorado Solar House, designed by George O.G. Lof, has been in continuous operation since 1957 with no maintenance done on the solar system. The performance of this system was measured during the heating season of 1959-60 and the results were published. (See review - Lof - "Design and Performance of Domestic Heating System Employing Solar Heated Air - The Colorado Solar House". Proc. U.N. Conf. New Sources of Energy, 1964 p. 185) The performance of the system was remeasured during the heating season of 1974-75 to determine the amount of degradation over the years; it was then restored. It was found to be 73 percent as effective as it was during the 1959-60 heating season. The collectors are the overlapped-glass plate type of air-heater (see Figure 2-14).

Conclusions: (1) Approximately 1.9%/year degradation in performance was found over the 15 years of system operation.

(2) Of the total:

8% can be attributed to lowered transmittance due to dirt and staining.

2-1/2% can be attributed to broken black glass plates

16.2% can be attributed to leakage, unequal air distribution, and some of the hot panels had one of the glass covers broken over the years.

Author: Whillier, A.

Title: "Black Painted Solar Air-Heaters of Conventional Design"

Publication: Solar Energy, Vol. 8, No. 1, 1964

Type of Work: Analytical

Abstract: The objects of this paper are: (1) To develop a rational heat transfer analysis of solar air heaters of conventional design (see Figure 2-12b) (2) To show that the resistance to heat transfer between the flowing air and the absorber plate is a major limiting factor of solar air-heaters (3) To compare the performance of glass and tedlar covered air heaters (4) To show that for heating air to 10 C (50°F) above ambient the economically optimum construction of a solar heater employs a black painted absorber of mesh screen covered with tedlar.

Conclusions: (1) It is desirable to maintain an effective heat transfer coefficient between the absorber and flowing air of at least $34 \text{ w/m}^2\text{C}$ ($6 \text{ Btu/hr. ft.}^2 \text{ }^\circ\text{F}$) and if possible as high as $45.4 \text{ w/m}^2\text{C}$ ($8 \text{ Btu/hr. ft.}^2 \text{ }^\circ\text{F}$). The author notes, however, that the required pumping power increases as the heat transfer coefficient is increased. (2) A heater without a glass cover performs so poorly that it is usually advantageous to have at least one glass cover on any collector. A solar air heater without a cover is of interest only if the h exceeds $45.4 \text{ w/m}^2\text{C}$ ($8 \text{ Btu/hr. ft.}^2 \text{ }^\circ\text{F}$). (3) When air is being heated from ambient temperature to about 10 C (50°F) above ambient, the quality of the glass as measured by its extinction coefficient is not of great importance. Also, under these same conditions the performance of a solar air heater is independent of the number of cover layers, excluding the case of no covers. (4) At relatively low temperatures of operation, i.e., the inlet temperatures being not more than -1 C (30°F) above ambient and with the exit air being up to above 10 C (50°F) above the inlet temperature, the most economical air heater would comprise a

single layer of tedlar above a gauze screen absorber. (5) When air enters the heater at a temperature much higher than ambient, i.e., when the difference between inlet and ambient is greater than -1°C (30°F) care should be taken in the selection of a cover system. If three or four covers are used it is very important to use low iron content, high quality glass. (6) The most economical cover system would be one made with glass as the outer cover and tedlar covers for the inner one. (7) There is merit in using simple reflectors to increase the rate of solar flux incident on the collector surface of a high temperature solar air heater. A high temperature collector is defined as a heater that operates with inlet temperatures of 93.3°C (200°F) or greater and uses three or more cover glazings.

B. Review of Textbooks, Theses, Handbooks, Unpublished Papers, Abstracts

Textbooks

Authors: Duffie, J.A., Beckman, W.A.

Title: "Solar Energy Thermal Process", 1974, Wiley-Interscience

Chapter 7 is entitled "Other Collector Geometries". The governing performance equation and the plate efficiency factors for various flat plate air-heaters are derived in detail. A thorough bibliography of previous research work on collectors in general is outlined at the end of the chapter. Some of the practical aspects of collector design are discussed. The heat transfer correlations required to compute the number of collector design parameters are given and the original research is cited for those readers requiring greater detail.

Authors: Kreider, J.F., Kreith, F.

Title: Solar Heating and Cooling: Engineering, Practical Design, and Economics

(continued)

In Chapter 4, p. 139, a section entitled "Air-Cooled Solar Collection Systems" covers the advantages/disadvantages of air heaters, thermal storage, and collector designs that commonly employ air as the heat transfer medium.

Editor: Sayigh, A.A.M.

Title: Solar Energy Engineering, 1977, Academic Press, Inc.

M.K. Selcuk is the author of Chapter 8 which is entitled "Solar Air Heaters and Their Applications". Various conceptual air heating designs are discussed and the Hottel-Whillier collector model is derived in detail. A number of means of providing extended surface area are discussed and some of the recent research in this area is cited. A section of the chapter deals with air heating applications. Space heating, cooling, drying and source heat are discussed as possible applications.

Theses

Author: Malik, M.

Title: "Hydrodynamic and Heat Transfer Characteristics of a Solar Heated Air Duct"

Publication: Ph.D. Thesis, Mechanical Engineering Department, Michigan State University, 1967

Type of Work: Experimental, Analytical

Abstract: Hydrodynamic and heat transfer characteristics of two rectangular ducts fabricated from: (a) sheet metal and (b) corrugated steel were compared in this experimental work. Flow was perpendicular to the corrugations for design (b) above. Neither of the surfaces had cover glazings. The test apparatus is shown in Figure 2-16.

Conclusions: (1) In fully developed, turbulent flow the friction coefficient was 40% greater for duct (b) compared to duct (a).

(2) The convection heat transfer coefficient associated with an unsymmetrically heat duct is less than that of a symmetrically heated duct.

(3) The improvement in heat transfer coefficients of (b) over (a) is proportional to the square root of the ratio of friction factors $(f_b/f_a)^{1/2}$. The increase in pressure drop is proportional to (f_b/f_a) .

Figure 2-16
Test Apparatus for Comparing Corrugated
and Smooth Sheet Duct

Handbooks

Author: Edwards, D.K.

Title: Solar Collector Design

Publication: Solar Energy Series, Franklyn Institute Press, 1977

Type of Work: Model development, handbook

Abstract: The design parameters of flat-plate collectors are derived in this solar collector handbook. Several basic air-heater designs are illustrated and the apparent advantages of each are discussed.

Authors: Jordan, R.C., Liu, B.Y.H., Editors

Title: "Chapter VIII - Prediction of Performance of Solar Collectors" by Austin Whillier

Publication: Applications of Solar Energy for Heating and Cooling of Buildings, ASHRAE GRP 170, 1977

Type of Work: Model development, handbook

Abstract: A linear collector model that assumes a constant loss coefficient, U_L , is developed in chapter 8. Design parameters such as F' and F_R are derived and expressions for F' that relate to several air-heater designs are given. The thermal and transmission losses are discussed in detail.

Author: Los Alamos Scientific Laboratory in conjunction with the San Francisco Operations Office

Title: ERDA's Pacific Regional Solar Heating Handbook

Type of Work: Analytical, modeling, handbook

Abstract: Design parameters for both liquid and air-heating collectors are discussed in this design handbook. Parameters that are of particular interest in the design of air-heating collectors are: (1) the hA product associated with the absorber and air flow and (2) the air flow rate through the collector. Detailed simulation studies were done for six cities. They are: (a) Phoenix, Arizona (b) Santa Maria, California (c) Fresno, California (d) Medford, Oregon (e) Seattle, Washington and (f) Bismarck, North Dakota.

Conclusions: (1) The minimum heat transfer effectiveness, hA , that is acceptable is $23 \text{ w/m}^2 \text{ C}$ ($4 \text{ Btu/hr.ft.}^2\text{F}$). The percent solar carried by a system increases linearly with increasing hA up to a value of $hA = 45 \text{ w/m}^2 \text{ C}$ ($8 \text{ Btu/hr.ft.}^2\text{F}$). Note that the simulations were made with a mass flow rate of $0.01 \text{ m}^3/\text{s m}^2$ (2 cfm/sq. ft.) of collector area.

(2) Assuming the absorber to air heat transfer effectiveness is held constant and equal to $23 \text{ w/m}^2 \text{ C}$ ($4 \text{ Btu/hr.ft.}^2\text{F}$), a severe performance penalty will result with an airflow less than $0.005 \text{ m}^3/\text{sec m}^2$ (1 cfm/sq.ft.) of collector. There is no apparent advantage of using air flow rates greater than $0.01 \text{ m}^3/\text{sec m}^2$ (2 cfm/sq. ft.).

(3) As air flow rate is changed, hA varies as $(\text{cfm})^{0.8}$ for a fixed geometry.

Unpublished Papers

Author: Charters, W.W.S.

Title: "Theory and Performance of Solar Air-Heaters"

Publication: Unpublished paper

Type of Work: Analytical

Abstract: Three air-heater designs are discussed; (1) flat-plate (2) overlapped glass plate, and (3) matrix. The heat transfer mechanisms and the available correlations that predict the heat transfer coefficient associated with the flat plate and overlapped glass plate collectors are given. The parameters that effect thermal performance and fundamental definitions such as collector efficiency are discussed.

Conclusion: No conclusions were drawn.

Authors: Kuzay, T.M., Malik, M.A.S., Ridenour, S.M., Boer, K.W.

Title: "Experimentation with Solar Collectors, Solar One: First Results"

Publication: Unpublished, Institute of Energy Conversion, University of Delaware, Newark, Delaware, 1971

Type of Work: Experimental

Abstract: The University of Delaware has a twofold interest in air-heaters. (1) The air is used as a cooling medium for the CdS/Cu₂S photovoltaic cells mounted on the roof of the Solar Laboratory. (2) The heated air is the heat transfer medium from the roof collectors to the storage device. Various collector designs were tested. Some had fins attached to the absorber (Figure 2-17).

The fins were attached by various fastening techniques. Some were riveted with the voids filled with epoxy. Others had the fins soldered to the galvanized plate. Different styles of corrugated aluminum plate were also used as an absorbing surface. The two-cover collectors were all installed on the roof of the house, side-by side, such that they would be tested under identical conditions. The test results reported were for tests performed at 15 minute intervals on one day. The results showed that the non-finned variety of collectors tested had efficiencies in the range of 30-35 percent. Due to poor thermal contact the absorber with the riveted fins showed an efficiency of 13 to 19 percent lower than the one with soldered fins, which showed efficiencies from 46 to 69 percent. It is of interest to note the difference in temperature between the absorber at its maximum temperature and the flowing air at its maximum value (outlet temperature) for the different collector designs.

<u>Collector type</u>	<u>Δt between maximum absorber & outlet air temperature</u>
soldered fins	19 C (35°F)
riveted fins (voids filled with epoxy)	28 C (50°F)
corrugated, non-finned	56 C (100°F)

The flow rates associated with the collectors in these experiments varied between 0.005 and 0.03 m³/sec m² (1 and 6 cfm/sq. ft.) of collector.

The pressure drops through the corrugated collectors were less than 0.025 cm (0.01 inches) of water per panel. The pressure drop across the finned collector is approximately 0.025 cm (0.01 inches) of water at a flow of 0.035 m³/sec m² (7 cfm/sq. ft.). A second type of finned absorber is shown in Figure 2-17. The collector using this fin was found to have an efficiency near 75 percent. The Δt between the absorber and air is near 4 C (7°F). The pressure drop at the flow rates tested were near 0.51 cm (0.20 inches) of water for this collector.

Conclusions:

1. The temperature difference between the air flow and corrugated absorber plates were not sufficiently low to provide adequate cooling for the CdS cells.
2. Interrupted and staggered fins are an attractive means of minimizing the plate-to-air temperature difference.

Figure 2-17

University of Delaware Air-Heaters

Author: Hedstrom, J.C.

Title: "Single Solar Collector Results"

Publication: Unpublished Office Memorandum from Los Alamos Scientific Laboratory, date May 18, 1976

Type of Work: Experimental

Abstract: This memo describes the work that LASL has done with three air-heater designs. One was a roof-integrated design the other two were collectors for use on their mobile home application. One of the objects of the LASL collector comparison was to compare the effects of various fastening techniques between the absorber and the extended heat transfer surface. The fastening methods are shown in Figure 2-18.

Conclusions: (1) The absorber configuration used in Mobile Home #2 was not as efficient as that used in Mobile Home #1. The thermal contact was probably better in the first design.

(2) The LASL-1 performance was midway between the Mobile Home absorbers #1 and #2.

(3) The Mobile #1 absorber provided a measured effective heat transfer coefficient equal to $17 \text{ W/m}^2 \text{ C}$ ($3.0 \text{ Btu/hr.ft.}^2\text{F}$).

Figure 2-18

Absorber/Fin Fastening Techniques

- (a) Spot weld (LASL-1)
- (b) Sheet metal screw (Mobile #1)
- (c) Stud gun weld (Mobile #2)

Author: Moan, K.L.

Title: "An Analysis of the Low Loss Evacuated Tubular Collector Using Air as the Heat Transfer Fluid"

Publication: Unpublished paper, presented at the 1977 Annual Meeting of the American Section of ISES, June 6-10, 1977, Orlando, Florida

Type of Work: Experimental, analytical

Abstract: The practicality and cost-effectiveness of evacuated tubular collectors that use air as the heat transfer fluid are evaluated in this paper. Such problems as the ease of insertion and support of tubes, leakage at the interface, flow distribution, cost, and aesthetics are addressed. Pressure drop through two 1.14 m (45 inch) tubes was found to correlate with

$$\Delta P = (\text{cfm}^{1.84})/27 \quad \Delta P \text{ in inches water guage}$$

Calculations based on the above relationship indicate that the fan power required to drive the system should be less than 5% of the total useful energy collected. An all-day efficiency curve is presented.

- Conclusions:
- (1) Evacuated tubular collectors used to heat air provide the high grade of energy required for cooling cycles and many industrial process heat applications.
 - (2) Since the performance of evacuated tubular collectors are not particularly sensitive to plate temperature, tubular collectors used to heat air perform at approximately the same level as liquid-cooled tubular collectors.
 - (3) The air-cooled tubular collector has a relatively long time-constant.
 - (4) Because of the higher associated temperatures, storage size can be reduced.

C. Review of Patents

Patentee: Angilletta, D.J.

Patent No.: 4,002,159, November 7, 1975

Summary of Patent: The absorber plate is a venetian blind-type device that is mounted between two panes of glass. Air flows over and through the venetian blind slats where it is heated. Figure 2-19

Figure 2-19

Venetian Blind Air-Heater

Patentee: Barques, M. Amat

Patent No.: 3,194,228, July 13, 1965

Patent Summary: One of the stated objects of this patent is to ".... provide an absorbent surface comprising a sheet of corrugated cardboard of the usual type whose surface has been blackened with the application of amorphous carbon in the form of a paint or varnish which is deposited by any desired means...." It is specified in the patent that the corrugated cardboard be attached to a thin metal sheet. If a liquid is to be heated it will flow in tubes wound through the corrugated absorber. A gas could be heated by passing it in contact with the absorber surface. A cross-sectional view of the collector is shown in Figure 2-20.

Figure 2-20

Corrugated Paper Absorber

Patentee: Bourne, R.C.

Patent No.: 3,971,359, July 27, 1976

Summary of Patent: A flat-plate solar air-heater that can be installed between light frame structural members is the subject of this patent. A louvered structure placed behind a cover glazing and in front of the absorber plate reflects high angle solar rays in summer and passes low angle rays in the winter. The louvered structure also serves to direct the air stream over the absorber plate for enhanced heat transfer. See Figure 2-21.

Patentee: Close, D.J.

Patent No.: 276,788 - Australian, November 26, 1965

Summary of Patent: The invention covered by this patent is a solar air-heater with an absorber fabricated into a series of vee-corrugations. The included angle is in the range of 40° to 90° . If the metal from which the absorber is made, reflects in a specular manner, then the effective absorptance of the surface will be increased over that of a flat absorber made from the same material. The advantages of the vee-corrugated absorber design are discussed. See Figure 2-22.

Figure 2-21
Louvered Collector

Figure 2-22
Vee Corrugated Absorber/Collector

Patentee: Gardner, J.G.

Patent No.: 3,561,425, February 9, 1971

Summary of Patent: This patent applies to a solar water heater. A transparent cover rests on the embossed absorber plate. The dimples serve two functions: (1) Additional heat transfer area and (2) a means of support for the cover.

Figure 2-23

Embossed Absorber

Patentee: Johnston, J.G.
Patent No.: 3,875,925, April 8, 1975

Summary of Patent: A matrix type air-heater, shown in Figure 2-24, is the subject of this patent.

Figure 2-24
Matrix Absorber

Patentee: Keyes, J.H., Strickland, C.I., Strickland, R.G.
Patent No.: 3,946,720, March 20, 1976

Summary of Patent: An air-heating collector that uses black painted cup-like devices as the absorbing surface is described in the patent. Typically a cup 7 cm (2-3/4 inches) in diameter and 5 cm (2 inches) in length will increase the surface area of the accumulator approximately 4.75 times. Air is circulated over the absorber surface and is thereby heated. Baffles direct the air over the entire absorber surface. Figure 2-25.

Figure 2-25

Cup Air-Heater

Patentee: Lof, G.O.G. - Solaron Corporation, Denver, Colorado

Patent No.: 4,072,142, February 7, 1978

Summary of Patent: It is desirable to add heat to the air flowing through an air heating collector at a rate which is as large as practical. One method of enhancing the heat transfer is by increasing the turbulence in the air stream. This patent describes practical methods of creating turbulence in the collector air channels, i.e., by inserting perforated plates or corrugated screens in the airstream. The additional surface area gained by the plate or screens also improves the heat transfer. See Figure 2-26.

Figure 2-26

Air-Heater With Turbulator

Patentee: Lof, G.O.G. - Solaron Corp., Denver, Colo.

Patent No.: Patent number not issued to date

Summary of Patent: A common method of increasing heat exchange between a gas and a metal is by increasing the surface area of the metal. Relatively simple, inexpensive methods of doing this are required as the heat exchanger areas associated with solar heating systems are relatively large. This patent describes various means of adding surface projections,

dimples or discontinuities in the absorber surface of an air heating collector. An example of a dimpled absorber is shown in Figure 2-27.

Figure 2-27

Dimpled Absorber Assembled into Collector

Patentee: Miller, K.W.

Patent No.: 2,680,437, June 8, 1954

Summary of Patent: The air-heater described in this patent is a series of overlapping plates, which may be ordinary window glass (Figure 2-14). The exposed or bottom area under the inclined plates is covered with aluminum foil cemented to the glass with a black adhesive. When viewed from below, the entire visible area is a highly reflecting polished metal surface.

Patentee: Moan, K.L., Pei, Y.K., Owens-Illinois Inc.
Patent No.: 3,960,136, June 1, 1976

Summary of Patent: An evacuated tube collector, which uses air as the heat transfer medium, is covered in this patent. The manifolding arrangement is also discussed. (Figure 2-28).

Figure 2-28

Evacuated Tube Air-Heater

Patentee: Moreau, E.
Patent No.: 201,439, March 19, 1878

Summary of Patent: A blackened, wire-gauze matrix absorber is described in this patented flat-plate air-heater. Figure 2-29.

Figure 2-29
Matrix Air-Heater

Patentee: Parker, L.W.

Patent No.: 3,919,998, November 18, 1975

Summary of Patent: A solar air-heater consisting of a series of nested containers, each with its own transparent cover is described in this patent. Air flows through the collectors by natural convection. An air-to-water heat exchange is made where steam is generated. A sketch of the patented design is shown in Figure 2-30.

Patentee: Pyle, D.L.

Patent No.: 3,902,474, September 2, 1975

Summary of Patent: A simple box style air-heating collector is described which is divided into a maze of air passageways (Figure 2-31). These passageways are partially filled with a shredded, heat conductive, blackened material. Air is forced through the collector by a blower.

Figure 2-30

Nested Absorber
Air-Heater

Figure 2-31

Box Air-Heater

Patentee: Rom, F.E.

Patent No.: 3,908,631, September 30, 1975

Summary of Patent: Air is heated as it flows over a blackened, plastic film which forms the air duct. The duct is inflated by the pressure developed by the blower as shown in Figure 2-32. The heated air passage is surrounded on top by an insulating space which is made of clear plastic and inflated by air pressure developed by a small secondary blower.

Figure 2-32

Plastic Film Air-Heater

Patentee: Safdari, Y.B.

Patent No.: 4,019,494, August 11, 1976

Summary of Patent: An air-heating collector which incorporates interrupted fins on the absorber underside is the subject of this patent. It is stipulated that the air flow pass in a general direction parallel with the fins. The relationships among the fin spacing, height and length are discussed. (Figure 2-33).

Figure 2-33

Interrupted Fin Absorber Cross Section

Patentee: Schoenfelder, J.L.

Patent No.: 3,863,621, February 4, 1975

Summary of Patent: An air-heating solar collector is described which uses a gang-nail absorber plate. Air is circulated on both sides of the absorber. The collector is shown in Figure 2-34.

Figure 2-34

Gang Nail Air-Heater

Patentee: Shoemaker, M.J.
Patent No.: 3,102,532, September 3, 1963

Summary of Patent: This patent relates to a matrix type solar air-heater with discussions pertaining to the material to be used as the absorber surface. One material suggested for the absorber surface in this patent is slit and expanded metal foil, similar to the type used in the construction of grease filters. Perforated or expanded sheet metal, a plastic such as Tedlar, metallic ribbon, or a metal wool are all suitable materials for the absorber according to the patent. Figure 2-35.

(a)

(b)

Figure 2-35

- (a) Matrix Absorber
- (b) Collector System

Patentee: Thomason, H.E.

Patent No.: 3,145,707, August 25, 1964

Summary of Patent: The "trickle collector", described in this patent, normally uses water as the heat transfer medium. Water "trickles" from the top to the bottom of the collector. As is suggested in the patent a gas such as air could be blown through the collector. This patent describes the different absorbing surfaces that could be used in this type of collector. The patent stated that in the "trickle collector" it is important to have channels or other means of providing for even flow distribution over the collector surface. A modification of a channeled collector is one that is embossed, which will provide means for distributing the fluid and cause a tortuous flow pattern. The sketches (Figure 2-36) will aid in the visualization of the various types of absorbers that are suggested in this patent.

Figure 2-36
Trickle Collector Designs

SECTION III

HEAT TRANSFER CONSIDERATIONS

Design Constraints

An examination of the steady-state energy balance equation on the absorber plate (equation 3-1), indicates that there are three means of increasing heat transfer from the absorber to the air stream.

$$q_a = q_u + q_L \quad (3-1)$$

One method involves reducing the thermal losses from the collector, q_L . The second method requires increased energy absorption at the absorber plate, q_a . The third method involves increasing the rate of heat transfer from the absorber to the air stream, q_u .

Practical, cost-effective methods of increasing q_u by increasing the heat transfer effectiveness (ηhA) associated with the absorber plate are the subject of the present development work. The following are design constraints imposed on the improved absorber surfaces.

1. The pressure drop is limited to approximately 0.17 cm. w.g./m (0.02 i.w.g. /lineal ft.) of collector.*
2. It is desirable to retain two basic features of the present Solaron collector design, i.e., the internal manifold and the stagnant air gap between the absorber and glazing.
3. The design concepts must be potentially cost-effective and adaptable to mass-production.

* Operating costs are held constant between designs by maintaining the baseline pressure drop and equal mass flowrates.

Air-Heater Model

The familiar Hottel-Whiller collector model which describes collector performance as a function of the energy absorbed, thermal losses, mass flow rate, and the rate of heat transferred from the absorber to air-stream was used to predict the effect of increased absorber to air heat transfer.

The general performance equation for a flat plate air-heater of the type shown in Figure 3-1 is:

$$q_u = F_R \{ (\tau\alpha) H - U_L (t_i - t_{amb}) \} \quad (3-2)$$

where F_R = the heat removal factor and is equal to

$$F_R = \frac{GC_p}{U_L} \{ 1 - \exp -(F' U_L / GC_p) \} \quad (3-3)$$

and $F' = \frac{1}{1 + U_L / h_o} \quad (3-4)$

where $h_o = h_a + \left(\frac{1}{\frac{1}{h_b} + \frac{1}{h_r}} \right) \quad (3-5)$

Note that h_a may be modified to take into account the extended heat transfer surface area if the assumption is made that the radiation heat transfer coefficient, h_r , is not changed significantly. In the designs where good thermal contact between the absorber and back plate is provided, and the fin is thermally efficient, there will be a relatively small temperature gradient in the fin. These designs will have negligible heat transfer by radiation between neighboring fins, and the portion of h_o in parantheses (equation 3-5) can be neglected with little error. The overall convection coefficient, h_o , will be the product, hA , modified by the overall surface effectiveness, η_o , which takes the fin efficiency into account.

EXPLODED VIEW OF BASELINE SOLARON COLLECTOR

Figure 3-1

Since equation 3-4 describes the heat transfer mechanisms of an air heating collector, it can be used to evaluate the effect of any given value of h_o (the overall heat transfer coefficient between the absorber and air flow) for any given value of U_L . Figure 3-2 has F' plotted as a function of h_o , with U_L as a parameter.

Figure 3-2

Plate Efficiency Factor vs. Overall Heat Transfer Coefficient

It is evident from these curves that a low loss collector, $U_L = 2.3$ w/m^2C ($U_L = 0.40$ Btu/hr. ft.²F) is improved to a lesser extent with increasing h_o than a conventional flat plate collector $U_L = 5.7$ to 7.4 w/m^2C (1.0 to 1.3 Btu/hr. ft.²F).

Since the Soloron Series 2000 collector has a U_L of approximately 7.4 w/m^2C (1.3 Btu/hr. ft.²F), a value of h_o between 22 and 34 w/m^2C (4 and 6 Btu/hr. ft.²F) results in F' values between 0.74 and 0.82. As h_o approaches infinity, F' approaches 1. The economic optimum may be somewhat less than these values, but the curves do show that there is very little gained in increasing heat transfer beyond $h_o = 34$ w/m^2C (6 Btu/hr. ft.²F). Since the model does not predict performance to the required degree of accuracy necessary to determine the economic optimum, this must be left to experimental work.

Heat Transfer Considerations

Heat transfer in rectangular ducts

The convective heat transfer coefficient, h , associated with fully developed flow in an asymmetrically heated rectangular duct is correlated by the following relationship found experimentally, by Tan and Charters (19).

$$Nu = 0.0182 Pr^{0.4} Re^{0.8} \quad (3-6)$$

or for air $Nu = 0.0158 Re^{0.8} \quad (3-7)$

Note that the constant preceding the Reynolds number is somewhat lower than that associated with flow in round tubes with symmetric heating.

Flow in the developmental region of the duct will usually have increased heat transfer due to the greater turbulence in that region. The convective heat transfer coefficient associated with the inlet region of a rectangular duct was also determined by Tan and Charters (18) and is correlated by:

$$Nu = Nu_{\infty} (1 + S D_H/L) \quad 9500 < Re < 22000 \quad (3-8)$$

where $S = 14.3 \log (L/D_H) - 7.9 \quad L/D_H \leq 60 \quad (3-9)$

$$S = 17.5 \quad L/D_H > 60 \quad (3-10)$$

It is evident from equation 3-6 that the heat transfer coefficient can be increased by simply increasing the Reynolds number or by decreasing the hydraulic diameter, i.e., the spacing between the absorber and back plate. Unfortunately, there is a corresponding increase in the required blower power. This fact is illustrated when the Darcy - Weisbach equation is combined with the Blasius formula for friction factor (equation 3-11).

$$f = 0.079/Re^{0.25} \quad (3-11)$$

Resulting in the following expression for pressure drop through a duct.

$$\Delta P = \frac{2fLV^2\rho}{g_c D_H} = \frac{0.158L \text{ Re}^{1.75} \mu^2}{g_c \rho D_H^3} \quad (3-12)$$

If the friction factor, f , is something other than that given by 3-11, it should be used in 3-12 to compute the pressure drop. The theoretical pumping power is simply the product of volumetric flow rate and pressure drop or,

$$\text{pumping power} = Q \Delta P \quad (3-13)$$

Radiation heat transfer

In a well designed collector the radiation heat transfer between the absorber and back plate is energy that is eventually added to the air stream. The effect that radiation heat transfer has on the collector efficiency factor, F' , can be analyzed from equation 3-5. When the absorber/back plate is idealized as two infinite parallel plates, the radiation heat transfer coefficient can be calculated by the following relationship:

$$h_r = \frac{\sigma(T_1^2 + T_2^2)(T_1 + T_2)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \quad (3-14)$$

Heat transfer from extended surfaces

Kays and London (11) gives both analytic and experimental curves for a large number of symmetrically heated finned heat transfer surfaces. (It was noted in the research work by Charters on rectangular ducts that the actual heat transfer coefficients may be slightly smaller due to the asymmetric nature of the heated surfaces.) The friction factor associated with the designs are also given as a function of Reynolds number and can be used with equation 3-12 for predicting the pressure loss through the collector.

Due to the temperature gradient that is a characteristic of extended surfaces, it is necessary to consider the heat transfer efficiency of a finned surface. Standard heat transfer textbooks such as (12) define the fin efficiency as the ratio of actual heat transferred, to the heat transferred if the entire fin were at its base temperature. If the fin (Figure 3-3) has a uniform thickness, d , a uniform thermal conductivity, k , is cooled by a fluid of temperature t_f , and the heat transfer coefficient between the surface of the fin and the fluid is h , then the fin efficiency for a fin of length l is given by:

$$\eta_{fin} = \frac{\tanh ml}{ml} \quad (3-15)$$

$$\text{where } m = \left(\frac{2h}{kd}\right)^{1/2} \text{ for thin sheet fins} \quad (3-16)$$

When the fin efficiency of the extended surface is combined with the fin efficiency of the absorber surface then the overall surface effectiveness, η_o , is given by:

$$\eta_o = \frac{A_{fin}}{A} (\eta_{fin} - \eta_{abs}) + \eta_{abs} \quad (3-17)$$

Figure 3-4 has the fin efficiency plotted as a function of fin length. Rather than separate the effects of thermal conductivity and fin thickness, the parameter kd was used to cover a range of typical fin materials and thicknesses which would be used in flat plate air heating applications.

Heat transfer from spiked or pocketed surfaces

Pescod (16) has done experimental work on two styles of "spiked" or "pocketed" heat exchange surfaces. Using his data an estimate of the heat transfer coefficient between a dimpled or spiked absorber and air flow can be made. The heat exchangers were tested at Reynolds numbers between 200 and 2000 where the Reynolds numbers are based on hydraulic diameter characterized by the plate spacing. Since it was difficult to measure the increase in heat transfer area, Pescod chose to credit the increase in heat transfer to an increase in h only.

Figure 3-3
Absorber/Fin Schematic

Figure 3-4
Fin Efficiency vs. Length

The following correlations are given for calculating the heat transfer coefficient between the heat exchange surface and air flow, where V_a is the approach velocity,

$$\text{pocketed surface} \quad h = 100 V_a^{0.9} \quad (3-18)$$

$$\text{spiked surface} \quad h = 54 V_a^{0.7} \quad (3-19)$$

where h is in $\text{W/m}^2\text{C}$ and V_a is in m/sec .

The pressure gradient in the flow direction can be found from:

$$\text{pocketed surface} \quad \Delta P/L = 22 V_a^{1.75} \quad (3-20)$$

$$\text{spiked surface} \quad \Delta P/L = 4 V_a^{1.75} \quad (\text{Typical Correlation})$$

(3-21)

where L is in meters.

Caution must be exercised when extrapolating the data presented by the above correlations to Reynolds numbers other than those at which the tests were made.

Heat transfer from matrix surfaces

The heat transfer and flow characteristics of an air-heating collector that uses a matrix or wire-gauze absorber is quite different from one that uses a flat sheet design. Figure 2-4 shows the various flow schemes that have been used in the past. Since radiation is absorbed in "depth" the air must flow through the absorber rather than over the surface. Consequently, the heat transfer coefficient is generally based on volume rather than area. Volumetric heat transfer coefficients, h_v , and fluid friction characteristics have been found experimentally by Chiou et al (3), Beckman (2), Hamid (6), and Kays (11) all give data on various porous heat transfer media.

The work done by Chiou gives the following correlation for convective heat transfer and pressure loss for the matrix screen that had the largest ratio of heat transfer coefficient to pressure loss.

$$Nu_v = 0.0878 Re^{0.97} \quad (3-22)$$

$$f = 12.7 Re^{-1.09} + 1.06 \quad (3-23)$$

Note that the parameters for matrix surfaces have special definitions and are given in Appendix A.

The physical dimensions of the matrix screen to which these correlations apply are as follows:

overall screen thickness	3.8 cm (1.5 inches)
foil thickness	0.05 mm (0.002 inches)
voidage	97.4 percent
extinction coefficient	155/m (47.4/ft.)
hydraulic radius	0.966 mm (0.00317 feet)

Beckman's work with matrix surfaces (2) presents the computational results of a numerical model which predicts the collection efficiency. The fluid temperature rise can then be calculated. Hamid et al (6 , 7) experimentally verified earlier analytical work and presented experimental data on bed efficiencies. The reader is referred to the original papers for details on the screens and the results of the tests.

Design Parameters

Table 3-1 summarizes the design parameters for various absorber configurations that are readily calculatable. The calculations are made for the following general collector design:

double glazing = $(\tau\alpha) = 0.78$
flat black absorber

absorber to ambient
loss coefficient $U_L = 6.8 \text{ w/m}^2\text{C} (1.20 \text{ Btu/hr. ft.}^2\text{F})$

Design flow rate = $0.01 \text{ sm}^3/\text{sec m}^2 (2 \text{ scfm/sq. ft.})$

$\Delta P_{\text{coll}} = 0.08 \text{ cm w.g./m} (0.01 \text{ i.w.g./ft.})$

Table 3-2 summarizes the test results for various collector designs that have been measured and reported in the literature. The collectors that are included in tables 3-1 and 3-2 are restricted to low or moderate temperature air-heaters. The efficiency curves based on the calculated and measured parameters are given for comparison purposes and are presented in Figure 3-5. The details of the absorber designs for which the parameters were calculated are given at the end of Appendix A in Figures A-1 through A-9.

Description of Absorber Concepts to be Tested - First Phase of Tests

The first series of thermal efficiency tests had two objectives.

1. The side-by-side thermal efficiency tests were made so that independent improvement concepts incorporated in a particular design could be compared directly.
2. The practical upper limit on efficiency for a conventional flat-plate air-heater (meeting the above design criteria) was determined. Keeping in mind the overall project objective; the design of an improved, cost-effective, mass producible air-heating absorber, the designs outlined below were chosen for testing.

It is desirable to isolate the heat transfer parameters h and A in the initial stages of testing. The results of this comparison will provide the data essential to the design of an improved cost-effective absorber.

Conceptual Absorber Improvement	Free flow area m^2 (ft. ²)	Hydraulic Diameter m (ft.)	Reynolds Number at $0.01 \text{ sm}^3/m^2 \cdot s$ (2 scfm/ft. ²)	Convective heat transfer coefficient $W/m^2 \cdot C$ (Btu/hr.sq.ft.F)	Heat transfer effectiveness hA $W/m^2 \cdot C$ (Btu/hr.sq.ft.F)	Efficiency factor F'	Heat Removal factor F_R	Friction factor	$hA/$ $100 \times f$ $W/m^2 \cdot C$ (Btu/hr.sq.ft.F)
(1) Baseline	0.0131 (0.141)	0.0318 (0.0935)	5300	11-17 (2.0-3.0)	11-17 (2.0-3.0)	0.73*	0.60	0.0093	15 (2.7)
(2) Trip wire	0.0131 (0.141)	0.0318 (0.0935)	5300	17-23 (3.0-4.0)	17-23 (3.0-4.0)	0.78*	0.64	-	-
(3) Spiked	0.0175 (0.188)	0.0331 (0.125)	5100	45-51 (8.0-9.0)	45-51 (8.0-9.0)	0.89*	0.70	0.020	24 (4.3)
(4) Pocketed	0.0474 (0.510)	0.104 (0.340)	5100	51-57 (9.0-10.0)	51-57 (9.0-10.0)	0.90*	0.71	0.40	14 (2.4)
(5) Interrupted fin	0.0185 (0.199)	0.0203 (0.0665)	2500	8.5-17 (1.5-3.0)	8.5-17 (4.5-9.0)	0.86	0.69	0.012	32 (5.6)
(6) Vee corrugated	0.016 (0.172)	0.0233 (0.0764)	3400	11-17 (2.0-3.0)	22-34 (4.0-6.0)	0.81*	0.65	0.010	28 (5.0)
(7) Corrugated (transverse flow)	0.0153 (0.165)	0.0335 (0.110)	5200	11-17 (2.0-3.0)	11-17 (2.0-3.0)	0.74*	0.60	0.013	11 (1.9)
(8) Continuous fin	0.0141 (0.150)	0.0156 (0.0510)	2400	8.5-11 (1.5-2.0)	8.5-11 (4.5-6.0)	0.30	0.65	0.007	43 (7.5)
(9) Matrix [†]	1.74 (18.7)	0.00387 (0.0125)	4.9	20-30 (3.0-5.0)	unknown	0.30	0.70	3.3	-

[†] definitions of parameters pertaining to matrix collectors are according to Chlou (3) and are defined in Appendix A.

* The effects of radiation heat transfer from the absorber to the back plate are taken into account.

TABLE 3-1
Calculated Parameters for Two Cover Air-Heating Collectors at Constant Pressure Drop

Absorber Design	Number of glazings	Y Intercept of Efficiency Curve	Slope of Efficiency Curve $\frac{w}{m^2 C}$ (Btu/hr.ft. ² F)	Mass Flowrate $\frac{kg}{hr.m^2}$ (lbm/hr.ft. ²) (scfm/ft. ²)	Pressure Loss cm. w.g. (in. w.g.)	Reference
(1) double corrugated (iron)	1	0.73	7.84 (1.38)	56.9 (11.6) (2.58)	(Pressure Drop was constant for the four air-heaters tested but was not specified)	Gupta (5)
(2) Double corrugated (aluminum)	1	0.75	7.66	72.0 (14.7) (3.27)		Gupta (5)
(3) iron mesh above corrugated sheet	1	0.72	8.86	69.5 (14.2) (3.16)		Gupta (5)
(4) Aluminum mesh above corrugated sheet	1	0.52	7.26 (1.28)	43.9 (8.98) (2.00)		Gupta (5)
(5) vee-corrugated [†] foam glass	1	0.73	6.05 (1.07)	120 (25.0) (5.6)		Loth (14)
(6) Jet Plate*	1	0.71	6.87 (1.21)	10.7 (21.9)	0.48 (0.19)	Rask (17)
(7) Parallel Plate*	1	0.65	6.08 (1.07)	89.3 (18.3) (4.07)	0.58 (0.23)	Rask (17)

*Efficiency based on absorber area.

[†]Efficiency based on aperture area

TABLE 3-2

Measured Parameters for Various Low or Moderate Temperature
Air-heating Collectors
That Have Data Reported In the Literature

(a) Efficiency Curves based on
Calculated Parametric Values
found in Table 3-1

$$Y \text{ intercept} = F_R (\tau \alpha)$$

$$X \text{ intercept} = (\tau \alpha) / U_L$$

(2 Collectors in series
 $Q = 0.01 \text{ s m}^3/\text{m}^2 \text{ sec}$)

(b) Efficiency Curves based on
Empirical Parametric Values
found in Table 3-2

$$\Delta T/H = (t_i - t_{amb})/H$$

$$\text{m}^2 \text{ c/W}$$

Figure 3-5. (a) (b)

Design No. 1 - Flat absorber with extended surface

A common method of increasing gas flow heat exchange is to add heat transfer area in the form of extended surfaces. The pressure drop can be controlled quite simply in this design by sizing the free flow area properly. This subsequently decreases the convective heat transfer coefficient by a small amount but not enough to make the method ineffective. An extended surface is also relatively easy to model mathematically and therefore provides a good check for model development. (Figure 3-6)

Design No. 2 - Flat absorber with turbulators

Increasing the relative roughness of a flow duct surface will generally cause an increase in the heat transfer coefficient due to increased turbulence in the boundary layer. The effectiveness of this approach is dependent on the turbulator design and the Reynolds number at which the collectors operate. Unfortunately, much of the research done regarding turbulator design has been done with flow at Reynolds numbers greater than 12000. As evident in Table 3-1, the Reynolds number associated with air-heating collectors are generally much lower than 12000, i.e., $Re = 2000$ to 5000 , which is in the transition region between laminar and turbulent flow. (Figure 3-7)

Design No. 3 - Staggered dimple design

The third and final absorber design to be tested is the dimpled absorber. Pescod's (16) work showed that an arrangement of "spikes" on a diamond pattern provided a favorable $h/\Delta P$ ratio in comparison to smooth surfaces. The "dimpled" design that was tested in the present work was a compromise between Pescod's spiked and pocketed heat exchanger surface. Due to manufacturing limitations it was not possible to duplicate the desirable spiked design exactly. The effectiveness of an increased heat transfer area and an increased convective heat transfer coefficient (Figure 3-8) can be determined by testing the dimpled design.

SECTION A-A

SOLARON CORP.
DESIGN No. 1
3/14/77

FIGURE 3-6
FINNED EXPERIMENTAL ABSORBER

Figure 3-7 (a)
Trip Wire Experimental Absorber

SOLARON CORP.
DESIGN NO 2
ABSORBER
FEB. 15, 1977

SOLARON CORP.
DESIGN NO 2
BACK PLATE
FEB. 15, 1977

Figure 3-7 (b)

Trip Wire Experimental Absorber Back Duct Plate

BY: AK DATE: 1-1-77 SUBJECT: Proposed 1. Dwg. SHEET NO. 1 OF 1
 CHKD. BY: AK DATE: 2-1-77 Salazar, C. A. P. JOB NO. PR-151
44 SAMPLE ARTICLE B-7864 Roll. Extra
 SK 3771

Figure 3-8
 Dimpled Experimental Absorber
 Design No. 3

Blank

3-20

SECTION IV

Instrumentation and Test Results of First Series of Tests

Introduction

The comparison of different collector designs is best accomplished by simultaneous, side-by-side testing. Since collector performance is a function of ambient conditions; insolation, wind speed and ambient temperature, and these parameters are not controllable in an outdoor test, it is highly desirable to compare performance under identical operating conditions.

Collector Test Stand

The collector test stand is a simple A-frame type structure with a 53° south sloping roof line. The south facing roof is capable of holding 10 collector modules (5 wide X 2 high). (See photograph Figure 4-1).

As can be seen from Figure 4-2, the testing of the various collector designs was carried out in a parallel, closed loop, configuration. This configuration allows for simultaneous, side-by-side testing in an outdoor environment.

Located at both the inlet and outlet of each collector is an averaging arrangement of thermocouples. This arrangement permits the constant monitoring of both the inlet and outlet temperatures during testing.

Flow through the collectors is controlled by a series of dampers placed downstream from the outlet thermocouple of each collector. By opening and closing these dampers, the proper ΔP across the appropriate nozzle is obtained. This fixes the mass flow rate through the collector. Additional flow control is obtained by dampers located on the inlet and outlet sides of the blower.

Figure 4-1
Solaron Collector Test Facility

Air leaving the collectors passes to a common air duct where it is collected and reconditioned to the proper temperature. The temperature of the fluid is controlled by a series of air reconditioning devices (Figure 4-3). For low inlet temperatures, a 4-row water coil is used with a three-way mixing valve controlled by a Honeywell M944A Modutrol motor. Higher inlet temperatures require the use of a duct heater. This 4.8 KW heater is controlled by a Barber Colman SCR Model #CA41 which, is controlled by a Barber Colman analog controller model #524G. This device controls the inlet air temperature to within 1°C (1.8°F).

The blower/motor used to circulate air through the collectors is a direct-drive, $3/4$ KW (1 HP), Dayton model #4C108. This fan is rated by the manufacturers to deliver $17 \text{ m}^3/\text{min}$ (610 cfm) at 10 cm (4 inches) of static pressure. (A high pressure fan is required in a test loop such as this to overcome the relatively large ΔP across the flow nozzle.)

Figure 4-2

Test Facility Flow Schematic

○ AVERAGE AIR TEMPERATURE
MEASUREMENT POINTS

Figure 4-3

Air Reconditioning Apparatus

Instrumentation - Test Stand

The instrumentation used in the tests follows the guidelines set forth in ASHRAE 93-77 (1). The measurement techniques are discussed in more detail in the paragraphs that follow. Three deviations from 93-77 were made. They are: (1) The measurement of the direct beam radiation has been deleted. (2) The measurement of the wet bulb temperature in the nozzle throat has been deleted. (3) The accuracy of the temperature difference between inlet and outlet air is limited by the calibration accuracy of the recorder, i.e., ± 0.4 C (± 0.7 F).

A. Temperature Measurement.

All temperature measurements are made with copper-constantan, type T, special limit, thermocouple wire. Temperature measurements are made

following the guidelines of ASHRAE 41.1 - 74, "Standard Measurements Guide, Section on Temperature Measurement". The limits of error are specified by the wire manufacturer to be as follows:

Temperature Range	Limits of Error
-9 to 93 C (15 to 200F)	± 0.4 C ($\pm 3/4$ F)
93 to 370 C (200 to 700F)	$\pm 3/8\%$ (of temperature being measured)

The temperature difference between the collector inlet and outlet air is measured by the averaging arrangement of thermocouples (Figure 4-4). The inlet and outlet ducts are each 20.3 cm (8 inches) in diameter in which there are eight thermocouples placed in equal cross sectional areas of the duct. The thermocouples are connected in parallel to a common terminal point from which thermocouple wires lead to the recording device. The measurement points are approximately 30.5 cm (1 foot) from the inlet and outlets of the collector at all of the test stations, and the ducts are insulated with 3.8 cm (1-1/2 inch) foil backed fiberglass insulation to minimize heat loss or gain from the ducts.

Ambient air temperature is measured approximately 1.5 m (5 feet) from the test structure at approximately mid-roof height. The thermocouple has been installed in a shaded, ventilated, environment which has been painted white to reflect the incident solar radiation.

B. Flow Measurement

The flow is measured independently at each of the five test stations. The nozzle boxes were constructed according to ASHRAE standard 37-69, "Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment," and a sketch of the box is shown in Figure 4-5.

Figure 4-4

AVERAGE INLET & OUTLET AIR TEMPERATURES

Figure 4-5

Nozzle Box

The nozzles are 5 cm (2 inches) in diameter (D), which meet the size specifications of ASHRAE 37-69. The pressure drop across the nozzle is measured with a 10.2 cm (4 inch) manometer subdivided into minor scale divisions of 0.05 cm (0.02 in.) of water. The temperature at the nozzle is measured with a thermocouple located directly below the inlet diffusion baffle (Figure 4-5).

The air flow rate through a nozzle can be found from the following simplifications of the Bernoulli equation, and have been extracted from ASHRAE 37-69.

$$Q = 1096 C A_n (P_v')^{0.5} \quad (4-1)$$

C = nozzle discharge coefficient

A_n in sq. ft.

P in in. w.g.

$$\text{where } v' = 29.92v/P_n \quad (4-2)$$

v' in ft.³/lbm air-water mixture

v in ft.³/lbm dry air

P_n in in. Hg

The volumetric flow of standard air (Q_s) can be found from

$$Q_s = Q/(0.075v') \quad (4-3)$$

with Q_s and Q in cfm

The discharge coefficient from the nozzle based on air temperature and Reynolds number can be found from figures 4-6 and 4-7.

$$\text{where } Re = fV_n D \text{ with } V_n \text{ measured in fpm} \quad (4-4)$$

D = nozzle throat diameter (in.)

Figure 4-6
Temperature Factor

Figure 4-7
Nozzle Discharge Coefficient

C. Insolation

Insolation is measured with an Eppley model PSP pyranometer. The instrument has the following characteristics as provided by the manufacturer.

sensitivity	10.58 microvolts/watt m ⁻²
impedance	650 ohms (approx.)
temperature dependence	± 1 percent for ambient temperatures between - 20C to 40C
linearity	± 0.5% 0 - 2800 W/m ²
response time	1 second
cosine	± 1% from normalization 0-70° zenith angle) ± 3% 70-80° zenith
orientation	no effect
mechanical vibration	up to 20g - no damage
calibration	approx. 1 cal/cm ² min.

The pyranometer is mounted in a plane parallel to the test collectors.

D. Recorder

The data recording device is a Doric model 220 recorder. It has the following characteristics.

resolution	0.1 (F or C)
calibration accuracy	$\pm 0.7F$ or $\pm 0.4C$
stability for 6 months	$\pm 0.5F$ or $\pm 0.3C$
temperature stability	$\pm 0.0025\% \times \text{RDG} \pm 1.7 \mu\text{V}/^{\circ}\text{C}$
repeatability at a point	± 1 count (max.) during a 20 minute period

E. Wind Speed

Wind speed is measured with a Stewart Electronic Wind Speedometer. The Stewart speedometer is a continuous running integrator giving an indication of the immediate past average wind speed (integrated over the past 5 minutes) measured by a 1/60th mile contacting anemometer. The scale range is 0-100 mph using a 0-1 milliamperere DC meter of 2% accuracy.

Error Analysis

Since computation errors have a compounding effect on the final results, it is desirable to determine the expected limits of accuracy on a plot of efficiency (η) versus $(t_i - t_{amb})/H$. The following equations for computing the resulting probable errors (R) were taken from the 1977 ASHRAE Handbook of Fundamentals.

$$(A \pm a) + (B \pm b) \quad R = \pm (a^2 + b^2)^{0.5} \quad (4-6)$$

$$(A \pm a)(B \pm b) \quad R = \pm \{(Ab)^2 + (Ba)^2\}^{0.5} \quad (4-7)$$

$$(B \pm b)/(A \pm a) \quad R = \pm \left\{ \frac{(Ba/A)^2 + b^2}{A^2} \right\}^{0.5} \quad (4-8)$$

where A, B are the measured quantities and a, b are the tolerances on the individual measurements.

For the following typical test conditions (and specified limits of error) the resulting errors using the above equations are as follows:

Case 1:

$$\begin{aligned} t_i &= 29.4 \text{ C} \pm 0.4 \text{ C} \\ &\quad (85 \text{ F} \pm 3/4 \text{ F}) \\ t_{amb} &= 26.7 \pm 0.4 \text{ C} \\ &\quad (80 \text{ F} \pm 3/4 \text{ F}) \\ t_o &= 47.8 \text{ C} \pm 0.4 \text{ C} \\ &\quad (118 \text{ F} \pm 3/4 \text{ F}) \\ H &= 867 \text{ W/m}^2 \pm 1\% \\ &\quad (275 \text{ Btu/hr. ft.}^2 \pm 1\%) \\ P &= 2.69 \text{ cm w.g.} \pm 0.03 \\ &\quad (1.06 \text{ i.w.g.} \pm 0.01) \end{aligned}$$

$$T_{noz} = 321 \text{ K} \pm 0.4 \text{ K} \\ (578 \text{ R} \pm 3/4 \text{ R})$$

$$G = 87.9 \text{ Kg/hr. m}^2 \\ (18 \text{ lbm/hr. ft.}^2)$$

$$\eta = Q_u/HA_g = 0.518 \pm 0.018 ; (t_i - t_{amb})/H = 0.003 \pm 0.007 \text{ m}^2\text{C/W} \\ (0.018 \pm 0.0038 \text{ hr.ft.}^2\text{F/Btu})$$

Case 2:

$$t_i = 93.3 \text{ C} \pm 0.4 \text{ C} \\ (200 \text{ F} \pm 3/4 \text{ F})$$

$$t_o = 96.9 \text{ C} \pm 0.4 \text{ C} \\ (206.4 \text{ F} \pm 3/4 \text{ F})$$

$$t_{amb} = 29.4 \text{ C} \pm 0.4 \text{ C} \\ (85 \pm 3/4 \text{ F})$$

$$H = 867 \text{ W/m}^2 \pm 1\% \\ (275 \text{ Btu/hr. ft.}^2 \pm 1\%)$$

$$P = 3.22 \text{ cm. w.g.} \pm 0.03 \\ (1.22 \text{ i.w.g.} \pm 0.01)$$

$$T_{noz} = 370 \text{ K} \pm 0.4 \text{ K} \\ (654\text{R} \pm 3/4 \text{ R})$$

$$G = 87.9 \text{ Kg/hr. m}^2 \\ (18 \text{ lbm/hr. ft.}^2)$$

$$\eta = 0.10 \pm 0.017 ; (t_i - t_{amb})/H = 0.07 \pm 0.007 \text{ m}^2\text{C/W} \\ (0.418 \pm 0.0038 \text{ hr.ft.}^2\text{F/Btu})$$

The examples are for the two extreme cases of collector efficiency, i.e., the efficiencies associated with a low air inlet temperature and a high air inlet temperature.

It should be noted that the ASHRAE requirement for measuring the temperature rise of the fluid to 0.1 C (0.2F) may not be as applicable to the testing of air collectors as it is to the testing of water heating collectors. The reason for this is that air collectors typically have a temperature rise of at least four times that of an equivalent water heater operating under identical conditions. For an equivalent percent error, the air collector test could withstand a tolerance of ± 0.40 C (± 0.80 F) on the individual temperature measurements.

Testing Objective

Testing was done in two phases as prescribed by the contract. The first phase was designed to determine which absorber concept had the greatest benefit-to-cost ratio, in comparison to the baseline collector. Single collector modules were tested in the first testing phase. For comparison purposes, tests done on single panels are adequate as long as the forced convective heat transfer coefficients between the air and absorbers are maintained at the values which would be realized under field conditions. In other words, the Reynolds number associated with a particular design must be the same as that which a field installed collector would experience. The second and final phase of testing was on a 2 X 2 array (4 collector modules) of the collector type (or combination of types) shown to be the most promising of those tested in the first phase. The results of the second series of tests are discussed in Section VI.

The test absorbers were designed to fit into the collector bodies of the current Solaron collector (Figure 4-8). The inlet and outlet conditions, and edge and back losses were therefore nearly identical allowing the

comparison to be based only on the difference in heat transfer effectiveness between the different absorber designs. The test absorbers were coated with 3M-Black Velvet paint applied according to the paint manufacturer's specifications. Black Velvet was chosen for its ease of application.

EXPLODED VIEW OF
BASELINE SOLARON
COLLECTOR

Figure 4-8

Surface measurements were made at the points shown in Figure 4-9. The thermocouples were located in shallow surface penetrations and then attached with a metal-filled cement. This cement was then painted with Black Velvet paint to match the rest of the absorbing surface.

Additional air temperatures were measured in the air passageways (Figure 4-9). These measurement points were located directly opposite the surface thermocouples so that a surface to air ΔT could be determined. These temperature measurements were useful in the determination of the flow development length. Typical results are shown plotted in Figures 4-15, 4-16 and 4-17.

NOTE:

MEASUREMENTS ARE FROM
REFERENCE LINES (INCHES)

TEMPERATURE MEASUREMENT POINTS

Figure 4-9

Test Description

The first phase of testing consisted of the following two tests: (1) A determination of the collector time constant and (2) A determination of the collector thermal performance curve. The test methods followed were those outlined in ASHRAE 93-77, "Methods of Testing to Determine the Thermal Performance of Solar Collectors" and its subsequently referenced standards. The basic test procedures are outlined below.

(a) Test No. 1 - Collector time constant

In order to minimize the transient effects on the collector test results, it is necessary to determine the collector time response. The collector time constant (T) is a measure of this phenomenon. The test procedure for finding T is outlined below:

1. Air is circulated through the collector at an inlet temperature (t_i) equal to the ambient temperature (t_{amb}). The air is continuously circulated until the outlet air temperature (t_o) is constant and remains so for at least 15 minutes.
2. Under a steady solar flux of at least 790 w/m^2 (250 Btu/h/ft.^2), the collectors are abruptly shielded from the incident solar radiation.
3. With t_i held constant, the exit air temperature (t_o) is monitored as a function of time (t_{oT}) until

$$\frac{t_{oT} - t_i}{t_{o \text{ initial}} - t_i} = 0.368 \quad (4-9)$$

4. The time required for the above equality to hold true is designated as the collector time constant (Figure 4-10).

Figure 4-10

(b) Test No. 2 - Collector Thermal Performance

The steady-state thermal performance of the collector can be described by the following equation.

$$Q_u = A_a F_R (\tau\alpha)_e H - F_R U_L A_g (t_i - t_{amb}) = \dot{m} c_p (t_o - t_i) \quad (4-10)$$

The collector efficiency is defined as $\eta = Q_u / H A_g$

$$\text{or } \eta = \frac{\dot{m} c_p (t_o - t_i)}{H A_g} = \frac{A_a}{A_g} F_R (\tau\alpha)_e - F_R U_L \frac{(t_i - t_{amb})}{H} \quad (4-11)$$

Equation (4-11) shows that if the collector efficiency (η) is plotted as a function of $(t_i - t_{amb})/H$, the y-intercept will be equal to $\frac{A_a}{A_g} F_R (\tau\alpha)_e$, the x-intercept equal to $\frac{A_a}{A_g} (\tau\alpha)_e / U_L$ and the slope equal to $F_R U_L$.

Procedure

Pick a point on the X-axis where a test point is desired and set it equal to $\frac{T_i - T_{amb}}{H}$. Solve for the required inlet temperature since $T_{ambient}$

and H are known. This procedure avoids heating the thermal mass of the system above the temperature required for that particular steady-state condition and results in saving considerable testing time.

With the system on and inlet temperature determined, the flow rate must be adjusted. Since the pressure drop across the nozzle for the appropriate flow rate is a function of collector outlet temperature, the correct flow rates cannot be set until a steady-state outlet temperature is reached. Once a steady-state outlet temperature is attained the pressure drop across the nozzle can be adjusted slightly for the correct air flow.

The data points that establish the performance curves are efficiency values determined by averaging the air temperature rise $(t_o - t_i)$ and the solar input (H) over a time period equal to or greater than the time constant for the particular collector. Since testing is done only on clear days the average is nearly equal to the instantaneous values. Data is taken both before and after solar noon.

Discussion of Results

A. Thermal-efficiency curves

The collector efficiencies, plotted in Figures 4-11 through 4-14, show that for a specified mass flow rate and pressure loss through a collector, the means to increased absorber-to-air heat transfer, is through additional

heat transfer surface area. The addition of a boundary layer trip-wire, or interrupted fin, (in the case of a finned absorber) may further enhance the heat transfer rate. This is evident from the 3% improvement in the y-axis of the trip-wire efficiency plot over that of the baseline collector. Note that the increased heat transfer resulting from an extended surface can be made without increasing the pressure loss through the collector, while a collector incorporating a boundary layer trip-wire had a small increase in pressure loss. The dimpled absorber on the other hand, although the most efficient absorber tested, suffered approximately seven times the pressure loss in comparison to the baseline collector. The large pressure loss is probably due primarily to the turbulence caused by the staggered dimple design. This additional pressure loss means higher system operating costs.

The test results at the lower flow rate, $0.01 \text{ sm}^3/\text{sec. m}^2$ (2 scfm/ft.^2), follow trends which one would expect based on theoretical considerations. The performance is lower for the low flow rate. However, the efficiency drops off at a slower rate for the finned absorber than it does for those absorbers with no extended surface. This is due to the dominant effects of the extended surface. The relative pressure drop through any of the collectors tested at the two flow rates, varies approximately as the ratio of volumetric flow rate to the 1.75 power, which is the expected result.

The scatter in the data can be attributed to several factors. First, it is probable that the collectors never came to a complete steady-state condition. This is due to the relatively long time constants associated with these air collectors and slight (1%) changes in insolation over a testing period (one time constant). The second factor is the changing environmental conditions, i.e., wind speed and direction and the ambient temperature.

Figure 4-11

Figure 4-12

Figure 4-13

Figure 4-14

B. Extrapolating Test Results of Single Collector Modules to Multiple Modules.

The Solaron collector modules are typically installed in configurations requiring flow through at least two collectors in series. Since the thermal efficiency tests were made on single modules, it is desirable to extrapolate these test results and predict the thermal efficiency of collector arrays with flow through multiple modules in series. The National Bureau of Standards (10) recognized this potential problem and has provided correction factors to apply to single collector test results to obtain the desired efficiency curve. The procedure assumes that the outlet temperature from the first collector module is the inlet temperature to the second module. Another method which may be more straight forward is outlined below. Note that for this procedure or the one outlined by NBS to apply, the Reynolds number must be the same for the single collector test and the series collector panel configurations. The collector efficiency factor, F' , is primarily a function of the absorber-to-air heat transfer coefficient and secondarily a function of the overall loss coefficient, U_L . Therefore, it is reasonably assumed that F' is a constant for both single and multiple collector arrangements for constant Reynolds number.

Using the familiar Hottel-Whillier collector model for air-collectors, with the Y and X intercept modified to compensate for the reduced aperture to gross collector area ($\frac{A_a}{A_g}$), the procedure for finding the corrected heat removal factor, F_R , $\frac{A_a}{A_g}$ is as follows:

- (1) From the Y intercept of an efficiency plot ($F_R (\tau\alpha) \frac{A_a}{A_g}$) with known values of $(\tau\alpha) \frac{A_a}{A_g}$ (0.69 for this test) $\frac{A_a}{A_g}$ solve for the heat removal factor F_R .
- (2) From the X intercept ($\frac{(\tau\alpha) A_a}{U_L A_g}$) solve for the overall loss coefficient, U_L , knowing $(\tau\alpha) \frac{A_a}{A_g}$.

- (3) From the expression for heat removal factor (F_R) solve for F' .

$$F' = - \frac{Gc_p}{U_L} \ln \left(1 - \frac{F_R U_L}{Gc_p} \right). \text{ Using the test mass flow rate } G.$$

- (4) This computed value of F' is a constant for single or multiple collector module use (assuming the Reynolds number was maintained).
- (5) Compute the value of F_R associated with the modified mass flow rate, G , which is based on multiple collector modules. Table 4-1 lists the heat removal factors for single collector modules which are based on test results and those for two collector modules in series which have been found by correcting those for single modules.

C. Adjusting Test Results to Account for Unequal Pressure Drops

In addition to correcting the efficiency curves to reflect the performance which results when there is flow through two panels in series (described in step 1 through 5 above) it is necessary to adjust the heat removal factors for the respective collectors so that they can be compared at equal pressure drops (with the baseline ΔP 0.30 cm w.g. used as the standard). The procedure for correcting the heat removal factors to account for this effect is described below:

1. Calculate the value of the heat removal factor, F_R , for the absorber design as it was built and tested. Compare this value with the value calculated to give a pressure drop equal to that of the baseline design.
2. If the calculated value of F_R differs from the measured value of F_R , a ratio of these two heat removal factors multiplied by the value calculated at the normalized pressure drop gives the corrected heat removal factor to be used for comparison purposes. The relationship described in steps (1) and (2) are expressed by the following relationship.

$$F_R \text{ (corrected to baseline } \Delta P) = \frac{F_R \text{ (measured at experimental } \Delta P)}{F_R \text{ (calculated at experimental } \Delta P)} \times F_R \text{ (calculated at baseline } \Delta P)$$

The heat removal factors calculated according to the above procedure are summarized below in Table 4-1.

Collector Design	measured heat removal factors for single collectors (F_R @ 4 scfm/ft. ²)	heat removal factors for 2 collectors in series with adjustments to account for unequal ΔP 's
Baseline	0.71	0.64
Tripwire	0.75	0.65
Fin	0.81	0.70
Dimple	0.84	0.71

TABLE 4-1

Collector Heat Removal Factors for Single Collector Modules and For Two Modules in Series with Normalized Pressure Drop

D. Flow development length and Absorber to Air Temperature Difference

Plots of the absorber and air temperatures in the flow direction are shown in Figure 4-15. It is interesting to note the thermal development lengths associated with the four absorber designs. It is apparent that the baseline and tripwire absorbers have development lengths of approximately one meter or 30 hydraulic diameters. The flow in the finned and dimpled absorbers becomes fully developed after 15-20 diameters.

The temperature difference between absorber and air flow is considerably less for the dimpled and finned absorber. In the fully developed region the temperature difference for the baseline and trip wire designs averages approximately 35-40 C, (63-72 F) whereas the dimpled and finned designs averages 10 C ΔT (18 F).

The temperature profiles measured at collector mid length in the direction perpendicular to the flow are plotted in Figures 4-16 and 4-17. The shape of the absorber temperature profile reflects the effects of edge losses. Examination of the plots for the baseline and tripwire designs indicate that there is slightly imbalanced flow in the collector duct, as is evident from the higher air temperatures at the collector perimeter. The dimpled design on the other hand has a flatter temperature profile. This is as expected since this design has a much greater resistance to flow and therefore acts as an excellent flow balancer. The temperature difference between the absorber and air flow is consistently smaller at the higher inlet temperatures. This observation is consistent with efficiency measurements as less energy is transferred to the air stream at the higher inlet temperatures.

Figure 4-15

Figure 4-16
Horizontal Temperature Profile

Figure 4-17
Horizontal Temperature Profile

Blank

4-28

SECTION V

ECONOMIC ANALYSIS AND CHOICE OF THE COST EFFECTIVE DESIGN

Of the three improvement concepts tested, the finned absorber was found to be the most cost-effective, practical, design. When comparing test results it at first appears that the dimpled design is the most efficient. However, after adjustments are made to the data to account for unequal pressure drops, the finned absorber and dimpled absorber are nearly equal (see Table 4-1). Note that the pressure loss associated with the dimpled design was 7 times greater than the baseline design and nearly 3 times greater than the finned absorber.

The Roll-Bond manufacturing process was used to manufacture the dimpled absorber tested in the present work. At the time of fabrication the 2.5 cm (1 inch) air passage was the maximum that could be formed and still maintain a good thermal bond between the two sheets of aluminum, given the spacing of the dimples. Other forming techniques that were investigated had even greater limitations. As pressure drop is directly related to the free flow area, it appears that there is no simple way of lowering the pressure drop of the dimpled design and still retain its high performance characteristics.

After correcting for pressure drop differences, the trip wire design showed an improvement by a factor of 1.01 over the baseline collector at the design flow rate. This improvement by itself is not significant but if incorporated with a fin it becomes more attractive provided that a manufacturing process is available. The pressure drop associated with the trip wire design was 50% larger than that associated the baseline design.

ECONOMIC ANALYSIS

The accompanying graph (Figure 5-1) shows the relative efficiencies of the four test absorbers after adjustment for pressure drop differences and for flow through two panels in series. The four efficiency curves are shown to be coincident on the X axis of the graph. The data verifies this in all of the absorbers except the dimpled design which differed by 6 1/2 percent. Since theory supports a coincident X axis intercept for collectors of similar construction, this difference can be attributed to experimental error.

The finned absorber is more efficient by a factor of 1.11 in comparison to the baseline absorber for all values of $(t_i - t_{amb})/H$. The allowable collector cost-increase which will give the economic break-even point is found using the analysis outlined below. The following two assumptions are made in this analysis.

1. Based on Soloron's experience, the installed cost of the present collector is \$269 per sq. m. (\$25 per sq. ft.) This figure represents collector costs as well as cap strip, hold-down hardware, gaskets and other costs that vary with collector array size. The cost does not include fixed costs such as storage, air handler or controller. This cost assumes no framing cost, i.e., the collectors are mounted on an existing frame or roof.
2. It is assumed that both collector designs provide equal annual percentages of solar heating and therefore have equal auxiliary energy requirements.

In essence assumption number 2 states that the capital costs of each system are equal as are the auxiliary energy costs. For example, both solar systems provide 70% of the annual heating requirements for an investment of \$13,000. The difference between the two arrays is the required collector area which provides 70% of the heating. The baseline collector array, for example, requires 37.2 sq. m (400 sq. ft.), the finned collector array requires 11% less or 33.4 sq. m. (360 sq. ft.). The baseline collector cost is \$269 per sq. m. (\$25.00 per sq. ft.), while the maximum allowable cost for the finned design is $\$269 \times 1.11 = \299 per sq. m. (\$27.75. per sq. ft.) The above costs are summarized below:

Collector Design	% solar heating	Cost per sq. ft. installed	Collector array sq. ft.	Fixed cost	Total Cost
baseline	70%	\$25.00	400	\$3,000	\$13,000
fin	70%	\$27.75	360	\$3,000	\$13,000

The incremental cost is: $\$299 - 269 = \30.00 per sq. m ($\$27.75 - 25.00 = \2.75 per sq. ft.)

The customer can afford to spend \$29.60/sq. m (\$2.75/sq. ft.) more over the cost of the baseline design and still realize the same economic payback. The cost multiplier between Solaron's cost and the customers' is approximately 3, which includes markups by Solaron, the distributor and the installing dealer. Therefore, Solaron can spend approximately $\$2.75/3 = \$9.80/\text{sq. m}$ (\$0.92/sq. ft) above the cost of the present design. The present absorber design costs \$6.46 per sq. m (\$0.60 per sq. ft.) including coating cost. Therefore, the maximum cost of a finned absorber including coating is:

$$\$9.80 + \$6.46 = \$16.26 \text{ per sq. m.}$$

$$(\$0.92 + \$0.60 = \$1.52 \text{ per sq. ft.})$$

MANUFACTURING PROCESSES

The following six manufacturing processes were considered as methods for manufacturing a mass-produced absorber with extended surface. The advantages and disadvantages of the various alternatives are discussed and in those cases where the designs looked practical from a manufacturing standpoint, cost estimates were obtained from qualified fabricators. Cross-sectional sketches of the products that would result from the manufacturing techniques are shown in Figure 5-2.

Several techniques for fastening fins to a flat absorber plate were compared in thermal efficiency tests at Los Alamos Scientific Laboratory. These tests showed that a substantial reduction in performance results if there is not a good thermal bond between the two surfaces. The experimental work on air-heating collectors that was conducted at the University of Delaware (13) confirms these results. Therefore, only bonding techniques that give a continuous good thermal bond similar to the types shown in designs 5 and 6, Figure 5-2, were considered. Various high temperature, thermally conducting, adhesives such as metal-filled epoxies were noted as a potential fastening method. However, since lifetime data is not available for conditions similar to that which a collector will experience over extended periods of time, the adhesives were not considered applicable for immediate use. Further research including lifetime testing is required before these materials could be considered for this application.

Design No. 1

The roll formed absorber shown has the advantage that it is made from one piece of sheet metal, therefore minimizing leakage to the upper side of the absorber surface. Although mounting flanges are easily incorporated on two sides of the rolled product the short sides require the addition of a flange in a subsequent assembly step. The crevices in the fin form spots where corrosion would occur due to moisture or chemicals used in the coating process.

Design No. 2

The tooling costs associated with the manufacture of the sectional roll-formed absorber is substantially less than that of design No. 1. Crimping the joint between sections would be a required secondary assembly step. It is extremely important to make the joints air-tight in this operation. A third assembly operation would require the addition of a flange to the short sides of the absorber plate. The corrosion problems that are inherent to design No. 1 are also a problem with design No. 2.

Design No. 3

The vee corrugated design does not have the corrosion problems that are associated with designs No. 1 and 2. This is a major advantage of the vee corrugated absorber. On the other hand because there is twice the absorber area in comparison to a flat absorber, the coating costs are higher. Hollands' (8), (9) and Elsherbiny et al (4) have studied vee corrugated surfaces extensively. The latter has shown that the thermal top losses are substantially larger than those bounded by flat surfaces. This effect is due to the increase in effective emittance and the increased natural convection heat transfer coefficient. According to Elsherbiny et al (4), for the same spacing the convective losses across air layers bounded on one side by a vee corrugated surface and a flat surface on the other side, are greater than those bounded only by flat surfaces by as much as 50%.

Design No. 4

Design No. 4 has disadvantages similar to Design No. 3. Design No. 4 is an absorber that is thermoformed from a zinc-aluminum alloy marketed by St. Joe Minerals Company. This alloy has been used very little in the solar industry up to the present time. It has the advantages of a relatively high thermal conductivity and since it can be formed into a wide variety of shapes, can be made to have an interrupted flow pattern, thereby increasing absorber to air heat transfer. Due to its relatively low melting temperature of 420 - 480 C (790 - 900 F) the available coating techniques are limited. For example, porcelain-enamel which is applied at 650 C (1200 F) cannot be used. The ability to form it to most any desirable shape and its corrosion resistance are attractive features of this design.

Design No. 5

The roll-bond process for manufacturing absorbers has been used rather widely in liquid heating collector designs. Its application to air collectors has not been exploited up to the present time. Made from aluminum, the heat transfer efficiency from the fin is high and the weight is low. Electro plating costs are generally somewhat higher when applied to aluminum due to the extra process steps involved. Its good corrosion resistance is a major advantage of design No. 5.

Design No. 6

A series of continuous seam welds between a flat absorber sheet and a corrugated back plate gives the associated cost and low thermal loss advantages of a flat absorber surface. The thermal conductivity of a steel fin is adequate to provide good heat transfer efficiency and the cost for a design that provides adequate rigidity is relatively low. A potential corrosion problem exists in the crevices between the back sheet and absorber surface. This problem can likely be overcome by applying a corrosion inhibitors on the inside of the flow passages.

No. 1 Roll-Formed

No. 2 Roll-formed sections

No. 3 Roll-Formed or stamped

No. 4 Thermo Formed-Interrupted

No. 5 Roll-Bond

No. 6 Seam weld

Figure 5-2 Manufacturing Processes

The estimated fabrication cost and tooling costs for design 3, 4, 5 and 6 are given in table 5-1 below. It was determined that designs 1 and 2 have excessive problems associated with them and therefore the costs of these designs were not determined. The costs have been supplied by reputable manufacturers but may differ depending on location, quantities, etc.

Absorber Design	Fabrication Cost \$/sq. m (\$/sq. ft.)	Tooling Cost
Stamped Vee Corrugated	10.76 (1.00)	\$16,200
Thermoformed Vee Corrugated	16.14 (1.50)	\$15,000
Roll Bond	15.28 (1.42)	\$ 500
Seam Weld (Tranter absorber)	10.56 (0.98)	\$ 4,000

Costs of Proposed Designs
Table 5-1

Choice of Absorber for Second Phase of Testing

Based on the analysis presented in this section, Design No. 6 was chosen as the most cost-effective design. Four panels were fabricated and tested in the second series of thermal tests. A detailed drawing of the seam welded absorber that was fabricated by Tranter Inc. is shown in Figure 5-3.

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5-10

SECTION VI

TEST RESULTS OF SECOND TEST SERIES

Thermal Performance

Four of the absorbers shown in Figure 5-3 were fabricated and assembled into standard Solaron collector bodies. A 2 X 2 collector array, the 4 panels on the left portion of the test facility (Figure 6-1), was tested for thermal performance in the manner outlined in Section IV of this report. The measured thermal efficiency is shown plotted in Figure 6-2. The Y intercept of the 2 X 2 array agrees with the Y intercept of the finned absorber tested in the single panel test when corrected for flow through two in series (Figure 5-1). The X intercept of the 2 X 2 array has shifted to the right by $0.014 \text{ Cm}^2/\text{w}$. The shift to the right on the X axis can be explained by the following:

1. The single collector tested in the first test series had a larger ratio of edge to surface area in comparison to the 2 X 2 array. Therefore, the single collector had larger edge losses than that associated with the 2 X 2 array.
2. The 2 X 2 array had more average back insulation than the single panel test due to the manifolding arrangement in the two different collector bodies used.

Time Constant

The thermal time constant of the 2 X 2 collector array was measured and is shown plotted in Figure 6-3. Due to the lighter gauge steel used in fabricating the absorbers of the 2 X 2 array, the time constant is $2/3$ of what it was for the single panel test. The faster time response of the 2 X 2 collector array permitted the test to come to a steady-state in less time which resulted in less scatter in the test data. Short time constants are particularly desirable when the test facility has a fixed orientation such as Solaron's.

Figure 6-1
Solaron Collector Test Facility
With 2 X 2 Collector Array

FIGURE 6-2 THERMAL EFFICIENCY OF A 2x2 COLLECTOR ARRAY
SECOND TEST SERIES

FIGURE 6-3 TIME CONSTANT FOR FINNED 2x2 COLLECTOR
ARRAY TESTED IN SECOND SERIES OF TESTS

Absorber to Air Temperature Difference

In the region of fully developed flow the absorber to air temperature difference averaged 15 to 18 C (28 to 32F), with an inlet temperature of 26 C (46F) above the ambient and an insolation of 788 w/m^2 (250 Btu/ hr. ft.^2). The absorber and air temperature profiles in the flow direction are plotted in Figure 6-4 for these conditions.

Measured vs. Calculated Fin Efficiency

Thermocouples were mounted at various points on the absorber surface and fin so that a temperature profile in the fin could be measured. The temperatures provide the necessary data for a comparison of the theoretical and measured fin temperature profile, if a heat transfer coefficient between the fin and air is calculated.

FIGURE 6-4 ABSORBER AND AIR STREAM TEMPERATURE PROFILES IN DIRECTION OF AIR FLOW (AT COLLECTOR CENTERLINE)

Kreith (12) gives the following relationship for calculating the temperature profile in a fin of uniform cross section with negligible heat loss from the end of the fin.

$$t - t_f = (t_a - t_f) \left(\frac{e^{mx}}{1 + e^{2ml}} + \frac{e^{-mx}}{1 + e^{-2ml}} \right) \quad (6-1)$$

$$\begin{aligned} \text{where } m &= (2h/kd)^{1/2} \\ &= \{(2)(9.65)(1000)/(45.0)(0.457)\}^{1/2} = 30.6/\text{m} \quad (9.34/\text{ft.}) \end{aligned}$$

when $x = l = 3.65 \text{ cm } (1-7/16 \text{ in.})$, $(t_l - t_f)/(t_a - t_f) = 0.590$

and the calculated temperature at the end of the fin, t_l , is $79.4 \text{ C } (175\text{F})$

$$\text{for: } t_a = 83.3 \text{ C } (182\text{F})$$

$$t_f = 73.9 \text{ C } (165\text{F})$$

The calculated value compares with the measured value of $78.3 \text{ C } (172\text{F})$. Typical temperatures of the absorber, fin and air are shown in Figure 6-5.

TEMPERATURE PROFILES IN FIN AND ABSORBER

FIGURE 6-5

SECTION VII

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

1. An improved, cost-effective absorber design has been developed. In comparison to the baseline absorber, the energy collected per unit area has been increased by a factor of 1.11 when the collector is operating with an inlet temperature equal to the ambient temperature. At $(t_i - t_{amb})/H = 0.04 \text{ Cm}^2/\text{w}$ ($0.23 \text{ hr. ft.}^2 \text{ F/Btu}$) the factor goes up to 1.23. This improvement was made while maintaining a pressure drop that is approximately equal to the baseline design.
2. Adding heat transfer surface area is an effective method of increasing air-heater thermal performance. The resistance welding techniques used to fabricate the 4 panels tested in the second test series is the simplest, most cost-effective technique found in the present work. Upon development, a thermally conducting long-life, adhesive (20 years) that can withstand sustained high stagnation temperatures would provide an acceptable alternative.
3. The improvement was made in a cost-effective manner. Following the method described in Section V for evaluating the economics of the improved absorber the break-even cost for the improved design is \$16.32/sq. m (\$1.52/sq. ft.) when operating at $(t_i - t_{amb})/H = 0$ and \$27.08/sq. m (\$2.52/sq. ft.) when operating at $(t_i - t_{amb})/H = 0.04 \text{ Cm}^2/\text{w}$ ($0.23 \text{ hr. ft.}^2 \text{ F/Btu}$). The above costs include the absorber coating costs and are based on the same assumptions that were used in Section V. The cost to manufacture the Tranter absorber was \$10.56/sq. m (\$0.98/sq. ft.) which does not include coating costs.

RECOMMENDATIONS

Based on its cost-effectiveness, the improved absorber design can justifiably be used in the Solaron air-heating collector. Its use, however, is dependent upon finding a method of providing adequate corrosion protection on the interior of the air passages which would make it suitable for use in all areas of the country.

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APPENDIX A

Details of Heat Transfer and Pressure Loss Calculations

The results of the calculations made in Appendix A are summarized in Table 3-1 (Section III). Since the different designs are compared with the baseline collector, it is logical to first compute the convective heat transfer coefficient and pressure loss associated with it. The convective heat transfer coefficient associated with the other designs are subsequently computed at pressure drops equal to that of the baseline collector. The pressure loss calculation requires an iterative computational procedure to check the friction factor as a function of Reynolds number. Usually one iteration is sufficient. A cross section of the baseline absorber/flow duct is shown in Figure A-1.

The following assumptions are made and apply to all of the designs for which calculations are made.

1. Design flow rate $G = 44.0 \text{ Kg/hr.m}^2$ (9 lbm/hr. ft.^2) = $0.01 \text{ sm}^3/\text{sm}^2$ ($2 \text{ s ft.}^3/\text{min ft.}^2$) based on gross collector area for two panels in series.
2. Air properties are evaluated at 38C (100F) and standard atmospheric pressure.
3. In those designs where radiation heat transfer between the absorber and back plate are important the values of long wavelength absorptivity and emissivity are assumed equal to $\epsilon = \alpha = 0.90$.
4. The overall collector loss coefficient, U_L , is defined as the sum of the top, back, and edge loss coefficients or

$$U_L = U_t + U_b + U_e = 6.81 \text{ w/m}^2\text{C} \text{ (1.20 Btuh/ft.}^2\text{ F)}$$

5. The collector's outside center to center dimensions are $0.915 \times 1.98 \text{ m}$ ($3 \times 6.5 \text{ ft.}$)

The following definitions are applicable to the analysis:

A_c = free flow area

P = wetted perimeter

D_H = hydraulic diameter = $4 A_c / P$
 $= (4)(b)(s) / 2(b + s) = 2s$ (for parallel plates $s \ll d$)

Q = volumetric flow

V = Q / A_c = average velocity in flow duct

f = friction factor defined by equation (3-11)

$\Delta P / L$ = pressure loss per unit length defined by the Darcy-Weibach equations as:

$$\Delta P / L = \frac{2f\rho V^2}{D_H g_c} \quad A-1$$

Reynold number = $Re = VD_H \rho / \mu = VD_H / \nu$

h = convective heat transfer coefficient for fully developed turbulent flow

h_{r12} = radiation heat transfer coefficient between infinite parallel plates

$$= \sigma F_{12} (T_1^2 + T_2^2)(T_1 + T_2) \quad A-2$$

F_{12} = gray body shape factor = $1 / (\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1)$

between surfaces 1 and 2

Design #1 - Table 3-1

Absorber description: baseline collector, two parallel plates through which the air passes, Figure A-1

a. $A_c = 0.0131 \text{ sq.m (0.141 sq. ft.)}$

b. $P = 1.75 \text{ m (5.73 ft.)}$

c. $D_H = 4 A_c / P = 0.0299 \text{ m (0.0985 ft.) or } D_H \approx 2(s) = 2(0.0159) = 0.0318 \text{ m (0.104 ft.)}$

d. $V = Q/A_c = (0.01)(3.62)(1.20/1.14)/0.0131 = 2.91 \text{ m/sec. (9.74 ft./sec)}$

(based on center to center spacing of two panels in series $A_{\text{gross}} = 1.81 \text{ sq. m (19.5 sq. ft.)}$ per panel

e. $Re = 5300$

f. $h_a = h_b = 0.0158 \text{ k/D}_H \text{ Re}^{0.8} = 13.6 \text{ w/m}^2 \text{ C (2.4 Btu/hr. ft.}^2\text{F)}$

g. $\Delta P/L = 2 f \rho V^2 / D_H g_c$ where $f = 0.079 / Re^{1/4} = 0.0093$
 $= 6.20 \text{ Pa/m}$
 $(0.03951 \text{ lb}_f/\text{ft.}^2/\text{ft.})$

h. The radiation heat transfer between the absorber and back plate is:

$$h_r = \sigma F_{a-b} (T_a^2 + T_b^2)(T_a + T_b)$$

$$\text{where } F_{a-b} = 1 / \left(\frac{1}{\epsilon_a} + \frac{1}{\epsilon_b} - 1 \right) = 0.818$$

$$t_a = 65.6\text{C} \quad t_b = 43.3\text{C}$$

$$(150\text{F}) \quad (110\text{F})$$

for $h_r = 6.5 \text{ w/m}^2\text{C}$
 $(1.15 \text{ Btu/hr. ft.}^2\text{F})$

The collector plate efficiency factor, F' , is:

$$i. \quad F' = \left[1 + \frac{U_L}{h_a + \frac{1}{\frac{1}{h_b} + \frac{1}{h_r}}} \right]^{-1} \quad \text{where } h_a = h_b = 13.6 \text{ W/m}^2\text{C} \text{ (2.4 Btu/hr.ft.}^2\text{F)}$$

$$h_r = 6.5 \text{ W/m}^2\text{C} \text{ (1.15 Btu/hr.ft.}^2\text{F)}$$

$$U_L = 6.8 \text{ W/m}^2\text{C} \text{ (1.20 Btu/hr.ft.}^2\text{F)}$$

$$F' = 0.73$$

The heat removal factor is

$$J. \quad F_R = GC_p / U_L \{1 - \exp - (F' U_L / GC_p)\} = 0.60$$

Design #2 - Table 3-1

Absorber description: baseline collector with boundary layer trip-wires attached to the absorber surface and back plate; Figure A-2.

- a. $A_c = 0.0131 \text{ sq. m (0.0141 sq. ft.)}$
- b. $P = 1.75 \text{ m (5.73 ft.)}$
- c. $D_H = 0.0299 \text{ m (0.0984 ft.)}$
- d. $V = Q/A_c = (0.01)(3.62)(1.20/1.14)/0.0131 = 2.91 \text{ m/sec (9.74 ft./sec.)}$
- e. $Re = VD_H/\nu = 5300$
- f. The convective heat transfer coefficient can be approximated from the correlation for convective heat transfer in the entrance region of a rectangular duct

$$N_u = N_{u_\infty} (1 + S D_H / L) \quad S = 6.49 \text{ for } L/D_H = 10$$

$$Nu = 0.0158 Re^{0.8} \{1 + 6.49 (0.10)\}$$

$$h_a = \frac{k}{D_H} Nu = 22.1 \text{ W/m}^2\text{C} \text{ (3.9 Btu/hr.ft.}^2\text{F)}$$

g. Since the trip-wires are spaced periodically, the friction coefficient must be evaluated experimentally. Therefore, no estimate on the resulting increased pressure drop associated with the trip-wire design will be made.

h. The radiation heat transfer coefficient, h_r , is the same as that found in design #1. $h_r = 6.5 \text{ W/m}^2 \text{ C}$ ($1.15 \text{ Btu/hr.ft.}^2 \text{ F}$)

$$i. \quad F' = \left[1 + \frac{U_L}{h_a + \frac{1}{\frac{1}{h_b} + \frac{1}{h_r}}} \right]^{-1} \quad \text{for } h_a = h_b = 19.9 \text{ W/m}^2 \text{ C} \text{ (3.5 Btu/hr.ft.}^2 \text{ F)}$$

$$U_L = 6.8 \text{ W/m}^2 \text{ C} \text{ (1.2 Btu/hr.ft.}^2 \text{ F)}$$

$$h_r = 6.5 \text{ W/m}^2 \text{ C} \text{ (1.15 Btu/hr.ft.}^2 \text{ F)}$$

$$F' = 0.78$$

$$j. \quad F_R = GC_p / U_L (1 - \{\exp - F' U_L / GC_p\}) = 0.64$$

Design #3 - Table 3-1

Absorber description: An arrangement of "spikes" on a diamond pattern act as turbulators in the flow duct, Figure A-3.

The first calculation must be the pressure loss through the flow channel. This calculation requires an iterative solution. An example of the procedure is shown below:

Rearranging equation A-1 gives

$$\frac{V^2}{D_H} = (\Delta P / L) g_c / (2 f \rho) \quad \text{A-3}$$

$$\text{also} \quad V = Q / A_c \quad \text{A-4}$$

from Pescod (16) guess a friction coefficient, say $f = 0.02$ at $Re = 4500$

knowing that $\Delta P / L = 6.20 \text{ Pa/m}$ ($0.0395 \text{ lb}_f/\text{ft.}^2/\text{ft.}$)

try $D_H = 0.0458 \text{ m}$ (0.150 ft.) or $s = 2.29 \text{ cm}$ (0.900 in.)

which gives $V = 2.50 \text{ m/sec (8.2 ft/sec)}$ from A-3

and $V = 1.86 \text{ m/sec (6.10 ft/sec)}$ from A-4

Since the velocities do not agree, try a new hydraulic diameter.

Try $D_H = 0.0381 \text{ m (0.125 ft.)}$ or $s = 1.91 \text{ cm (0.750 in.)}$

giving $V = 2.30 \text{ m/sec (7.53 ft/sec)}$ from A-3

$V = 2.23 \text{ m/sec (7.32 ft/sec)}$ from A-4

Since the friction factor is not known exactly, these velocities are close enough for a first approximation.

Check the Reynolds number based on a $V = 2.26 \text{ m/sec (7.42 ft/sec)}$

$$Re = 5140$$

The corresponding friction coefficient is $f \approx 0.02$ which was the original guess. The remaining parameters are listed below.

a. $A_c = 0.0175 \text{ sq. m (0.188 sq. ft.)}$

b. $P = 1.87 \text{ m (6.13 ft.)}$

c. $D_H = 2s = 0.0381 \text{ m (0.125 ft.)}$

d. $V = 2.26 \text{ m/sec (7.42 ft/sec)}$

e. $Re = 5140$

f. The convective heat transfer coefficient, h_a , is found from Pescod's (16) experimental plots of $St P_r^{2/3}$ versus Reynolds number. At $Re = 5100$

$$St P_r^{2/3} = 0.015$$

$$h/\rho V C_p = 0.015/0.72^{0.666}$$

$$h_a = h_b = 48.3 \text{ W/m}^2\text{C (8.5 Btu/hr. ft.}^2\text{ F)}$$

g. The pressure drop is as calculated.

h. Although not exact, the radiation heat transfer coefficient can be approximated by the infinite parallel plate relationship.

$$h_r = 6.5 \text{ W/m}^2\text{C} \quad (1.15 \text{ Btu/hr. ft.}^2 \text{ F})$$

$$i. \quad F' = \left[1 + \frac{U_L}{h_a + \frac{1}{\frac{1}{h_b} + \frac{1}{h_r}}} \right]^{-1} \quad \text{for } h_a = h_b = 48.3 \text{ W/m}^2\text{C} \quad (8.5 \text{ Btu/hr.ft}^2\text{F})$$

$$U_L = 6.8 \text{ W/m}^2\text{C} \quad (1.20 \text{ Btu/hr.ft.}^2\text{F})$$

$$h_r = 6.5 \text{ W/m}^2\text{C} \quad (1.15 \text{ Btu/hr.ft.}^2\text{F})$$

$$F' = 0.89$$

$$j. \quad F_R = GC_p/U_L (1 - (\exp - F' U_L/GC_p)) = 0.70$$

Design #4 - Table 3-1

Absorber description: A staggered arrangement of dimples on a diamond pattern, Figure A-4.

A procedure similar to that used to determine the heat transfer coefficient of the spiked design is used for the dimpled design.

A hydraulic diameter of $D_H = 0.104 \text{ m} (0.340 \text{ ft.})$ or $s = 5.18 \text{ cm} (2.04 \text{ in.})$ gives velocities of,

$$V = 0.842 \text{ m/sec} \quad (2.76 \text{ ft/sec}) \quad \text{from equation A-3}$$

$$V = 0.820 \text{ m/sec} \quad (2.69 \text{ ft/sec}) \quad \text{from equation A-4}$$

with an assumed friction coefficient $f = 0.40$ at $Re = 4500$

Checking the friction coefficient for a $V = 0.831 \text{ m/sec} (2.73 \text{ ft/sec})$

it is found that $Re = V\rho D_H/\mu = 5160$

$f = 0.40$ checks

the parameters are listed below

a. $A_c = 0.0474 \text{ sq. m (0.510 sq. ft.)}$

b. $P = 1.87 \text{ m (6.13 ft.)}$

c. $D_H = 2s = 0.104 \text{ m (0.340 ft.)}$

d. $V = 0.833 \text{ m/sec (2.73 ft/sec)}$

e. $Re = 5160$

f. The convective heat transfer coefficient, h_a , is found from Pescod's (16) experimental plots of $St P_r^{2/3}$ versus Reynolds number.

At $Re = 5100$

$$St P_r^{2/3} = 0.045$$

$$h_a = h_b = \rho V C_p (0.045)/0.72^{0.666} = 52.6 \text{ W/m}^2\text{C (9.27 Btu/hr.ft.}^2\text{F)}$$

g. The pressure drop is as calculated.

h. As a first approximation the radiative heat transfer coefficient is estimated to be the same as that for infinite parallel plates or

$$h_r = 6.53 \text{ W/m}^2\text{C (1.15 Btu/hr.ft.}^2\text{F)}$$

i.
$$F' = \left[1 + \frac{U_L}{h_a + \frac{1}{\frac{1}{h_b} + \frac{1}{h_r}}} \right]^{-1} \quad \text{for } h_a = h_b = 54 \text{ W/m}^2\text{C (9.5 Btu/hr.ft.}^2\text{F)}$$

$$U_L = 6.8 \text{ W/m}^2\text{C (1.2 Btu/hr.ft.}^2\text{F)}$$

$$h_r = 6.5 \text{ W/m}^2\text{C (1.15 Btu/hr.ft.}^2\text{F)}$$

$$F' = 0.90$$

j. $F_r = GC_p/U_L (1 - \exp - F' U_L/GC_p) = 0.71$

Design #5 - Table - 3-1

Absorber description: A fin having a good thermal bond to the absorber plate is interrupted periodically by staggering its placement; the fin length is such that the boundary layer never becomes fully developed

($L/D_H \leq 10$); the flow is generally parallel to the direction of flow; assume a 28-gauge steel fin and absorber with the fin length, equal to the fin spacing; Figure A-5.

Computing the required free flow area to give a pressure drop equal to the baseline collector.

From Kays and London (11) the friction coefficient corresponding to a Reynolds number of, $Re = 2500$, for $L/D_H = 10$ is $f = 0.012$. An L/D_H of 10 or less is desirable as 10 diameters is generally considered the development length for turbulent flow. A hydraulic diameter of 2.03 cm (0.0665 ft.) equates to a fin spacing or fin length equal to 2.03 cm (13/16. in.).

The air velocities computed from equations A-3 and A-4 are:

$$V = 2.15 \text{ m/sec (7.05 ft/sec) from A-3}$$

$$V = 2.10 \text{ m/sec (6.88 ft/sec) from A-4}$$

Based on average of the above velocities, the Reynolds number is,

$$Re = 2580$$

which checks the friction coefficient.

The design parameters for the interrupted fin are listed below.

a. $A_c = 0.0185 \text{ sq. m (0.199 sq. ft.) with } s = 2.03 \text{ cm (13/16 in.)}$

b. $P = 3.66 \text{ m (12 ft.)}$

c. $D_H = 4A_c/P = 2.03 \text{ m (0.0665 ft.)}$

d. $V = 2.13 \text{ m/sec (6.97 ft/sec)}$

e. $Re = 2580$

- f. The convective heat transfer coefficient, h_a , is found from Kays and London (11) where $St P_r^{2/3}$ is plotted as a function of Reynolds number and is,

$$h_a = \rho V C_p (0.0055/0.72)^{0.666} = 16.6 \text{ W/m}^2\text{C} (2.93 \text{ Btu/hr.ft.}^2\text{F})$$

- g. The pressure drop through the collector is as calculated.
- h. The radiative heat transfer between the absorber plate and the back plate is negligible in comparison to the convective heat transfer.

i. $F' = \left\{ 1 + \frac{U_L}{\eta_o h_a A} \right\}^{-1}$

The fin efficiency is computed as follows:

$$m = \left(\frac{2h}{kd} \right)^{1/2} = \left(\frac{2(16.6)}{(4/5.0)(0.000381)} \right)^{1/2} = 44.0/\text{m} (13.4/\text{ft.})$$

$$l = 2.03 \text{ cm} (13/16 \text{ in.})$$

$$ml = 0.909$$

For a rectangular fin with flow on both sides of the fin, the fin efficiency is given by,

$$\eta_{fin} = \left(\frac{e^{ml} - e^{-ml}}{e^{ml} + e^{-ml}} \right) / ml = 0.793$$

Assuming the absorber surface is 100% efficient, the overall surface effectiveness is,

$$\eta_o = \frac{A_{fin}}{A_t} (\eta_{fin} - 1) + 1$$

$$\text{where } A_{fin}/A_t = 2/3 \quad \text{and} \quad A = A_t/A_a = 3$$

$$\eta_o = 0.862$$

Then $F' = 0.86$ for $U_L = 6.8 \text{ W/m}^2\text{C}$ ($1.20 \text{ Btu/hr.ft.}^2\text{F}$)

$$h_a = 16.6 \text{ W/m}^2\text{C} \text{ (2.93 Btu/hr.ft.}^2\text{F)}$$

$$j. \quad F_R = GC_p/U_L (1 - (\exp - F' U_L/GC_p)) = 0.69$$

Design #6 - Table 3-1

Absorber description: A flat sheet is corrugated so that the included angle between corrugations is 60° consequently doubling the effective heat transfer area; flow is generally parallel to the direction of the vee corrugations; Figure A-6.

The pressure drop through a vee-corrugated flow passage is the first calculation to be made. As a first approximation, the surface is assumed to be smooth and the appropriate heat transfer and pressure loss correlations used for asymmetrically heated parallel plates are used for the vee-corrugated design. Assume a friction coefficient, $f = 0.01$, which corresponds to a Reynolds number of 3500.

A hydraulic diameter of 0.0233 m (0.0764 ft.) or a vee depth of 3.49 cm (1-3/8 in.) provides the free flow area to give the following resulting velocities:

$$V = 2.52 \text{ m/sec (8.26 ft./sec) from equation A-3}$$

$$V = 2.45 \text{ m/sec (8.02 ft/sec) from equation A-4}$$

Based on an average of the above two velocities, the Reynolds number is,

$$Re = 3460$$

which checks the friction coefficient used in the pressure loss calculations. The design parameters for the vee-corrugated absorber are listed and are as follows:

- a. $A_c = 0.0160 \text{ sq. m (0.172 sq. ft.)}$ with $s = 3.49 \text{ cm (1-3/8 in.)}$
- b. $P = 2.75 \text{ m (9 ft.)}$
- c. $D_H = 4A_c/P = 0.0233 \text{ m (0.0764 ft.)}$
- d. $V = 2.49 \text{ m/sec (8.14 ft/sec)}$
- e. $Re = 3460$
- f. The convective heat transfer coefficient, h_a , is found using the following correlation,

$$Nu = 0.0158 Re^{0.8} \quad \text{or } h = \frac{k}{D_H} 0.0158 Re^{0.8}$$

$$h_a = h_b = 12.2 \text{ W/m}^2\text{C} \quad (2.15 \text{ Btu/hr.ft.}^2\text{F})$$

If the entire absorber surface is irradiated there is no fin effect and the effective heat transfer area is two times the projected area.

$$h_a A = 12.2 \times 2 = 24.4 \text{ W/m}^2\text{C} \quad (4.30 \text{ Btu/hr.ft.}^2\text{F})$$

- g. The pressure drop through the collector is as calculated.
- h. The radiative heat transfer from the absorber to the back plate is enhanced slightly due to the directional qualities of the adjacent corrugations. Hollands (8) showed that the effective emittance of a 60° vee-corrugation increases from $\epsilon = 0.90$ for a flat surface to $\epsilon = 0.97$ for the corrugated surface. For absorber and back plate temperatures of $t_a = 65.5 \text{ C (150 F)}$ and $t_b = 43.3 \text{ C (110 F)}$ respectively, the radiative heat transfer coefficient is,

$$h_r = \frac{\sigma(T_a^2 + T_b^2)(T_a + T_b)}{1/\epsilon_a + 1/\epsilon_b - 1} = 6.98 \text{ W/m}^2\text{C} \quad (1.23 \text{ Btu/hr.ft.}^2\text{F})$$

$$\text{for } \epsilon_a = 0.97$$

$$\epsilon_b = 0.90$$

$$\begin{aligned}
 \text{i. } F' &= \left\{ 1 + \frac{U_L}{h_a A + \frac{1}{\frac{1}{h_r} + \frac{1}{h_b}}} \right\}^{-1} \quad \text{for } h_a A = 24.4 \text{ W/m}^2\text{C} \text{ (4.30 Btu/hr.ft.}^2\text{F)} \\
 &\quad U_L = 6.8 \text{ W/m}^2\text{C} \text{ (1.20 Btu/hr.ft.}^2\text{F)} \\
 &\quad h_b = 13.6 \text{ W/m}^2\text{C} \text{ (2.4 Btu/hr.ft.}^2\text{F)} \\
 &= 0.81
 \end{aligned}$$

$$\text{j. } F_r = GC_p / U_L (1 - (\exp - F' U_L / GC_p)) = 0.65$$

Design #7 - Table 3-1

Absorber description: A corrugated sheet that is commonly used in the construction of agricultural or industrial buildings is used as an absorber surface; flow is generally transverse to the direction of the corrugation; Figure A-7.

Based on experimental work at Michigan State University, Malik (15) found that the friction coefficient associated with corrugated surfaces is increased by 40% over that of smooth, flat surface. The convective heat transfer coefficient on the other hand was increased approximately 20% over that associated with a flat surface. A hydraulic diameter of $D_H = 0.0366 \text{ m}$ (0.110 ft.) corresponding to an absorber to back plate spacing of $s = 1.67 \text{ cm}$ (0.660 in.) provides the free flow area to give the following air velocities:

$$V = 2.66 \text{ m/sec (8.71 ft/sec) from equation A-3}$$

$$V = 2.55 \text{ m/sec (8.35 ft/sec) from equation A-4}$$

The Reynolds number based on an average of the above velocities is:

$$Re = 5210$$

which checks the assumed friction coefficient. The design parameters associated with the corrugated surface are listed below:

$$\text{a. } A_c = 0.0153 \text{ sq. m (0.165 sq. ft.)}$$

$$\text{b. } P = 1.86 \text{ m (6.11 ft.)}$$

- c. $D_H = 4A_c/P = 0.0329 \text{ m (0.108 ft.)}$
- d. $V = 2.61 \text{ m/sec (8.53 ft/sec)}$
- e. $Re = 5210$
- f. The convective heat transfer coefficient is increased by 20% over that of a smooth duct or $h_a = 0.019 \frac{k}{D_H} Re^{0.8} = 14.5 \text{ W/m}^2\text{C (2.55 Btu/hr.ft.}^2\text{F)}$
- g. The pressure drop through the collector is as calculated.
- h. The radiative heat transfer coefficient can be approximated as that between parallel plates, or, $h_r = 6.5 \text{ W/m}^2\text{C (1.15 Btu/hr.ft.}^2\text{F)}$
- i.
$$F' = \left\{ 1 + \frac{U_L}{h_a + \frac{1}{\frac{1}{h_b} + \frac{1}{h_r}}} \right\}^{-1} \quad \text{for } h_a = 14.5 \text{ W/m}^2\text{C (2.5 Btu/hr.ft.}^2\text{F)}$$

$$h_b = 12.0 \text{ W/m}^2\text{C (2.1 Btu/hr.ft.}^2\text{F)}$$

$$U_L = 6.8 \text{ W/m}^2\text{C (1.20 Btu/hr.ft.}^2\text{F)}$$

$$= 0.74$$
- j. $F_r = GC_p/U_L (1 - (\exp - F' U_L/GC_p)) = 0.60$

Design #8 - Table 3-1

Absorber description: A continuous fin having a good thermal bond to the absorber plate runs parallel to the direction of air flow; assume a 28 gauge, steel, fin and absorber with the fin depth equal to the fin spacing; Figure A-8.

First, computing the required fin length and flow area for a pressure drop equal to the baseline collector gives:

from Kays and London (11) at an assumed Reynolds number of
 $Re = 2300$

The corresponding friction coefficient, f , is $f = 0.007$

A hydraulic diameter of $D_H = 0.0155 \text{ m}$ (0.0508 ft.) equates to a fin spacing or fin length of 1.59 cm (5/8 inch).

The air velocities computed from equations A-3 and A-4 are:

$$V = 2.49 \text{ m/sec (8.16 ft/sec) from A-3}$$

$$V = 2.69 \text{ m/sec (8.82 ft/sec) from A-4}$$

Based on an average of the two above velocities the Reynolds number is,

$$Re = 2380$$

The friction coefficient assumed in the initial computation checks.

The design parameters for the continuous fin absorber are listed below.

a. $A_c = 0.0141 \text{ sq. m (0.152 sq. ft.)}$ where $s = 1.59 \text{ cm (0.625 in)}$

b. $P = 3.66 \text{ m (12 ft.)}$

c. $D_H = 4A_c/P = 0.0155 \text{ m (0.0507 ft.)}$

d. $V = 2.59 \text{ m/sec (8.49 ft/sec)}$

e. $Re = 2380$

f. The convective heat transfer coefficient, h_a , is found from Kays and London (11) and is,

$$h_a = \rho V C_p (0.0026/0.72)^{0.666} = 9.6 \text{ W/m}^2\text{C (1.69 Btu/hr.ft.}^2\text{F)}$$

g. The pressure drop through the collector is as calculated.

h. The radiative heat transfer between the absorber plate and the back plate is negligible in comparison to the dominant convective heat transfer coefficient.

$$i. \quad F' = \left\{ 1 + \frac{U_L}{\eta_o h_a A} \right\}^{-1}$$

the fin efficiency is computed as follows,

$$m = \left(\frac{2h}{kd} \right)^{1/2} = \left\{ \frac{(2)(9.60)}{(45.0)(0.000381)} \right\}^{1/2} = 33.4/\text{m} \quad (10.2/\text{ft})$$

$$l = 1.59 \text{ cm} \quad (5/8 \text{ in})$$

$$ml = 0.530$$

For a rectangular fin with flow on both sides of the fin, the fin efficiency is given by

$$\eta_{fin} = \left\{ \frac{e^{ml} - e^{-ml}}{e^{ml} + e^{-ml}} \right\} / ml$$

$$\text{or } \eta_{fin} = 0.916$$

Assuming the absorber surface is 100% efficient, the overall surface effectiveness is,

$$\eta_o = \frac{A_{fin}}{A_t} (\eta_{fin} - 1) + 1$$

$$\text{where } A_{fin}/A_t = 2/3$$

$$\text{or } \eta_o = 0.944 \quad A = A_t/A_a = 3$$

$$\text{then, } F' = 0.80 \quad \text{for } U_L = 6.8 \text{ W/m}^2\text{C} \quad (1.20 \text{ Btu/hr.ft.}^2\text{F})$$

$$h_a A = 28.8 \text{ W/m}^2\text{C} \quad (5.0 \text{ Btu/hr.ft.}^2\text{F})$$

$$j. \quad F_R = GC_p/U_L (1 - (\exp - F' U_L/GC_p)) = 0.65$$

Design #9 - Table 3-1

Absorber description: A matrix type absorber plate made from a porous medium such as expanded metal; solar energy is absorbed in depth; air flows through the porous absorber; Figure A-9.

By their nature matrix collectors absorb and transfer energy to air in quite a different manner in comparison to parallel plate air-heaters. Therefore, Chiou (3) et. al has redefined many of the parameters that characterize matrix absorbers. Chiou's study concluded that there were substantial differences among the various matrix screens that he tested. Referring to the original paper, his type III screen had the best ratio of heat transfer to pressure drop. The physical characteristics of the Type III screen mesh are given below:

foil thickness - 0.05 mm (0.002 in)
voidage (p) - 97.4 percent
extinction
coefficient - 14.5/m (47.4/ft)
Hydraulic diameter - 0.381 cm (0.0125 ft)
mesh depth (L) - 3.81 cm (1.5 in)

The pressure drop through the matrix collector is given by

$$\Delta P/L = \frac{f \rho V^2}{2 g_c r_h} = \frac{2 f \rho V^2}{g_c D_H} \quad A-5$$

where V = air velocity inside matrix

$$f = \text{friction factor} = 12.7 \text{ Re}^{-1.09} + 1.06$$

$$\text{Re} = G D_H / \mu$$

G = mass flow rate based on free flow area inside the matrix (Kg/hr. m²)

rearranging equation A-5 gives

$$\frac{V^2}{D_H} = \frac{\Delta P/L \ g_c}{2 f \rho} \quad A-6$$

also $V = Q/pA$

A-7

where $A = 3.62 \text{ sq. m (39 sq. ft.)}$
 $Q = 0.01 \text{ Sm}^3/\text{sec m}^2 \text{ (2 scfm/ft.}^2\text{)}$
 $p = \text{voidage}$

The Reynolds number for this collector is

$$Re = 4.86 \quad \text{and the corresponding friction factor is}$$

$$f = 12.7 Re^{-1.09} + 1.06 = 3.33$$

Equation A-6 gives the velocity internal to the matrix and is

$$V^2 = (0.00381)(645)/(2)(3.33)(1.14)$$

$$V = 0.569 \text{ m/sec (1.87 ft/sec)}$$

Since the face velocity is only 1.22 m/min (4 ft/min) at the design flow rate for two panels in series $\{0.01 \text{ sm}^3/\text{sec m}^2 \text{ (2 scfm/ft.}^2\text{)}\}$ then a velocity as large as 0.569 m/sec (1.87 ft/sec) would require a much smaller voidage than that specified (97.4 percent). This velocity calculation is an indication of the comparatively low pressure drop associated with matrix absorbers.

Another comparison can be made by computing the heat transfer coefficient at the design flow rate. This calculation is done under the following subheading f. First computing the face velocity (for two panels in series)

$$V = Q/A = 1.29 \text{ m/min (4.22 ft/min)}$$

The remaining design parameters are calculated below:

- a. $A_C = 1.81 \times 0.974 = 1.77 \text{ sq. m (19.0 sq.ft.)}$
- b. $P = \text{unknown}$

c. $D_H = 2 \times \text{voidage} \times \text{thickness of foil} / (1 - \text{voidage})$
 $= 0.381 \text{ cm (0.150 in)}$

d. $V = \text{face velocity} / \text{voidage}$
 $= 1.29 / 0.974 = 1.32 \text{ m/min (4.33 ft/min)}$

e. $Re = GD_H / \mu = 4.86$

f. The volumetric heat transfer coefficient, h_v , is found from

$$Nu = h_v D_H^2 / k = 0.0878 Re^{0.97}$$

$$h_v = \frac{k}{D_H} 2 \quad 0.0878 (4.86)^{0.97} = 746 \text{ W/m}^2 \text{ C/m (40.1 Btu/hr.ft.}^2\text{F/ft.)}$$

multiplying by the thickness gives an $h = 28.4 \text{ W/m}^2 \text{ C (5.0 Btu/hr.ft.}^2\text{F)}$

g. The pressure drop through the matrix screen is found from equation A-5 and is

$$\Delta P / L = 2f\rho V^2 / g_c D_H$$

$$= 0.964 \text{ Pa/m (0.0061 lb}_f\text{/ft.}^2\text{/ft.)}$$

$$\text{where } f = 12.7 Re^{-1.09} + 1.06 = 3.33$$

Comparing the pressure drop of a 3.8 cm (1-1/2 inch) thick mesh absorber and a 1.98 m (6-1/2 foot) long parallel plate collector at the design flow rate gives the following ratio,

$$\frac{\Delta P_{\text{parallel plate}}}{\Delta P_{\text{mesh}}} = 0.257 / 0.00076$$

$$= 338$$

h. The radiative heat transfer coefficient between the matrix and back plate is quite small as the bottom surface of the matrix should be at a relatively low temperature.

i. Whillier (20) gives the following approximate relationship for the plate efficiency factor of a matrix absorber.

$$F' = (1 + U_c/h)^{-1} \quad \text{A-8}$$

$$\text{where } U_c = \left[\frac{1}{h_{reg}} + \frac{1}{U_{ga}} \right]^{-1}$$

As a first approximation assume parallel plates and compute h_{rag}
(See Figure A-9)

with $\epsilon_g = 0.88$, $\epsilon_a = 0.90$, $t_a = 65.6 \text{ C (150 F)}$,

$$t_{\text{glass (inner)}} = 37.8 \text{ C (100 F)}$$

$$h_{rag} = \frac{\sigma(338^2 + 311^2)(338 + 311)}{(1/0.88) + (1/0.90) - 1} = 6.25 \text{ W/m}^2\text{C (1.10 Btu/hr.ft.}^2\text{F)}$$

$$U_{ga} = 6.25 \text{ W/m}^2\text{C (1.10 Btu/hr.ft.}^2\text{F)}$$

$$U_c = 3.12 \text{ W/m}^2\text{C (0.550 Btu/hr.ft.}^2\text{F)}$$

$$F' = \{1 + 3.12/28.4\}^{-1} = 0.90$$

j. The overall loss coefficient, U_L , is the sum of U_a , U_c and U_b .

where U_b = rear loss coefficient = $0.57 \text{ W/m}^2\text{C}$

($0.10 \text{ Btu/hr.ft.}^2\text{F}$)

U_a = loss coefficient between the air flow and the ambient

$$= (1/h_g + 1/U_{ga})^{-1}$$

$$= (1/8.5 + 1/6.3)^{-1} = 3.61 \text{ w/m}^2 \text{ C}$$

$$(0.64 \text{ Btu/hr.ft.}^2\text{F})$$

$$U_L = U_a + U_c + U_b$$

$$= 3.61 + 3.12 + 0.57 = 7.3 \text{ w/m}^2\text{C} (1.29 \text{ Btu/hr.ft.}^2\text{F})$$

$$F_R = GC_p/U_L \{1 - (\exp - F' U_L/GC_p)\}$$

$$= 0.70$$

Figure A-1
Baseline Absorber/Flow Duct

Figure A-3
Spiked Absorber

Figure A-4
Pocketed Absorber

Figure A-5
Underside of Interrupted Fin Absorber

Figure A-6
Vee Corrugated Absorber

Figure A-7
Corrugated Absorber Transverse Flow

Section A-A
Figure A-8
Underside of Continuous Fin Absorber

Figure A-9
Matrix Absorber

COLLECTOR PARAMETERS

AMBIENT CONDITIONS

	$\dot{m}$ (Kg/hr.m ²)	t_{in} (C)	t_{out} (C)	$\frac{t_{in} - t_{amb}}{m^2 H C/W}$	η	$\bar{t}_{amb}$ (C)	$\bar{H}$ (w/m ²)	$\bar{V}_{wind}$ m/sec	Wind Dir.	R.H. %
date - 9/1/77										
test 1:15										
interval 1:30										
<u>Collector</u>										
Baseline	79.28	62.11	76.00	0.0301	0.320	33.22	960.3	2.6-	S	25
Tripwire	79.71	61.94	75.78	0.0299	0.320			3.6		
Fin	79.43	62.50	78.17	0.0305	0.362					
Dimple	79.43	61.78	78.22	0.0298	0.380					
date - 9/14/77										
test 2:03										
interval 2:18										
Baseline	79.53	101.6	105.3	0.0792	0.0882	27.1	879.0	3.6	E	13
Tripwire	78.74	102.9	105.9	0.0807	0.0686					
Fin	79.35	103.3	107.1	0.0810	0.0891					
Dimple	73.30	101.4	106.3	0.0791	0.108					
date - 9/20/77										
test 12:22										
interval 12:37										
Baseline	82.91	70.44	81.56	0.0410	0.263	30.11	981.1	2.6	S	16
Tripwire	82.85	70.00	82.06	0.0407	0.284					
Fin	82.67	70.72	83.61	0.0414	0.303					
Dimple	77.09	69.56	84.94	0.0402	0.337					

APPENDIX B (Data - First Test Series)

COLLECTOR PARAMETERS

AMBIENT CONDITIONS

	$\dot{m}$ (Kg/hr.m ²)	t_{in} (C)	t_{out} (C)	$\frac{t_{in} - t_{amb}}{(\frac{H}{m^2 C/W})}$	η	$\bar{t}_{amb}$ (C)	$\bar{H}$ (W/m ²)	V_{wind} m/sec	Wind Dir.	$\bar{R}.H.$ %
date - 9/20/77										
test	1:08									
interval	1:23									
Baseline	80.56	98.11	102.6	0.0683	0.101	30.50	986.7	3.6	S	16
Tripwire	78.90	98.94	104.3	0.0692	0.119					
Fin	78.83	99.39	105.0	0.0697	0.124					
Dimple	73.36	97.78	105.8	0.0680	0.166					
date - 9/20/77										
test	1:45									
interval	2:00									
Baseline	79.68	108.7	110.9	0.0808	0.0500	30.33	969.8	3.4	SE	14
Tripwire	79.64	109.1	111.3	0.0812	0.0510					
Fin	79.58	109.5	111.9	0.0815	0.0550					
Dimple	72.75	107.4	112.2	0.0794	0.101					
date - 9/20/77										
test	2:22									
interval	2:37									
Baseline	81.84	81.83	90.89	0.0546	0.228	32.39	904.5	3.6	SE	14
Tripwire	81.73	82.11	91.89	0.0550	0.247					
Fin	81.50	83.11	94.00	0.0560	0.274					
Dimple	75.29	82.33	94.50	0.0551	0.282					

COLLECTOR PARAMETERS

AMBIENT CONDITIONS

	$\dot{m}$ (Kg/hr.m ²)	t_{in} (C)	t_{out} (C)	$\frac{t_{in} - t_{amb}}{H}$ (m ² C/W)	η	$\bar{t}_{amb}$ (C)	$\bar{H}$ (w/m ²)	$\bar{V}_{wind}$ m/sec	Wind Dir.	R.H. %
date - 9/20/77										
test	3:15									
interval	3:30									
Baseline	82.60	77.89	84.22	0.0583	0.187	32.50	778.8	4.9	S	12
Tripwire	82.44	80.28	85.67	0.0613	0.159					
Fin	82.38	80.56	86.17	0.0616	0.166					
Dimple	75.24	79.00	87.11	0.0597	0.218					
date - 9/21/77										
test	12:47									
interval	1:02									
Baseline	87.28	26.44	46.89	0.000	0.501	24.78	995.3	4.9	S	15
Tripwire	87.32	24.72	46.61	0.000	0.536					
Fin	87.09	24.61	48.33	0.000	0.571					
Dimple	79.27	24.56	51.39	0.000	0.597					
date - 9/21/77										
test	1:15									
interval	1:30									
Baseline	86.85	31.33	50.11	0.00599	0.455	25.28	998.1	4.5	S	30
Tripwire	86.87	30.11	49.94	0.00475	0.482					
Fin	86.60	30.22	51.94	0.00493	0.526					
Dimple	78.94	29.94	54.11	0.00475	0.533					

Data - First Test Series

COLLECTOR PARAMETERS

AMBIENT CONDITIONS

	$\dot{m}$ (Kg/hr.m ²)	t_{in} (C)	t_{out} (C)	$\frac{t_{in} - t_{amb}}{H}$ (m ² C/W)	η	$\bar{t}_{amb}$ (C)	$\bar{H}$ (w/m ²)	$\bar{V}_{wind}$ m/sec	Wind Dir.	R.H. %
date - 10/18/77										
test	1:06									
interval	1:21									
Baseline	42.20	63.72	79.11	0.0382	0.180	25.22	1008	3.6	S	19
Tripwire	42.19	63.72	83.56	0.0382	0.232					
Fin	42.02	63.00	87.33	0.0375	0.283					
Dimple	43.40	61.72	88.06	0.0363	0.316					
date - 10/18/77										
B-4 test	1:58									
interval	2:13									
Baseline	41.44	84.00	94.44	0.0592	0.125	27.00	962.2	3.6	E	19
Tripwire	42.33	83.33	95.17	0.0585	0.145					
Fin	42.23	81.72	97.00	0.0569	0.187					
Dimple	42.89	79.39	96.94	0.0544	0.218					
date - 10/18/77										
test	2:37									
interval	2:52									
Baseline	42.60	94.72	99.28	0.0777	0.0630	27.11	869.6	3.6	E	19
Tripwire	42.83	91.67	98.00	0.0741	0.087					
Fin	42.77	89.56	99.56	0.0719	0.137					
Dimple	42.72	87.06	99.11	0.0689	0.166					

COLLECTOR PARAMETERS

AMBIENT CONDITIONS

	$\dot{m}$ (Kg/hr.m ²)	t_{in} (C)	t_{out} (C)	$\frac{t_{in} - t_{amb}}{(m^2 H C/W)}$	η	t_{amb} (C)	$\dot{A}$ (w/m ²)	V_{wind} m/sec	Wind Dir.	R.H. %
date - 10/19/77										
test	1:11									
interval	1:26									
Baseline	41.18	90.44	99.78	0.0629	0.107	27.94	993.4	2.7	SE	9
Tripwire	41.41	86.94	99.00	0.0593	0.139					
Fin	41.36	85.11	101.1	0.0576	0.183					
Dimple	42.79	82.67	101.2	0.0551	0.220					
date - 10/19/77										
test	2:00									
interval	2:15									
Baseline	42.06	107.7	110.2	0.0805	0.0300	29.17	975.4	2.7	SE	9
Tripwire	42.37	102.1	107.7	0.0748	0.0650					
Fin	42.34	99.83	108.4	0.0724	0.105					
Dimple	42.41	96.61	108.2	0.0692	0.140					
date - 10/26/77										
test	12:53									
interval	1:08									
Baseline	42.80	69.00	89.89	0.0380	0.236	29.00	1049	2.8	S	8
Tripwire	42.51	69.22	90.89	0.0384	0.245					
Fin	42.28	68.61	95.44	0.0377	0.302					
Dimple	42.60	67.44	94.39	0.0366	0.306					

Data - First Test Series

COLLECTOR PARAMETERS

AMBIENT CONDITIONS

	$\dot{m}$ (Kg/hr.m ²)	t_{in} (C)	t_{out} (C)	$\frac{t_{in} - t_{amb}}{H}$ (m ² C/W)	η	$\bar{t}_{amb}$ (C)	$\bar{H}$ (W/m ²)	V_{wind} m/sec	Wind Dir.	R.H. %
date - 10/26/77										
test	1:48									
interval	2:03									
Baseline	42.86	81.78	96.78	0.0534	0.183	29.50	981.1	2.8	S	8
Tripwire	42.13	82.11	97.50	0.0535	0.187					
Fin	42.71	80.89	100.6	0.0523	0.240					
Dimple	43.29	79.56	99.56	0.0511	0.246					
date - 10/27/77										
test	1:50									
interval	2:05									
Baseline	43.04	20.06	53.00	0.00229	0.419	17.83	944.2	3.1	NE	33
Tripwire	43.82	18.61	52.83	0.000881	0.443					
Fin	43.45	18.78	58.89	0.00106	0.515					
Dimple	44.58	18.83	57.50	0.00106	0.509					
date - 10/28/77										
test	12:20									
interval	12:35									
Baseline	43.33	28.78	63.61	0.00546	0.417	23.33	1010	3.1	SE	17
Tripwire	43.30	28.56	62.89	0.00511	0.411					
Fin	42.94	28.44	69.44	0.00511	0.487					
Dimple	44.07	28.11	67.67	0.00476	0.483					

Data - First Test Series

COLLECTOR PARAMETERS

AMBIENT CONDITIONS

	$\dot{m}$ (Kg/hr.m ²)	t_{in} (C)	t_{cut} (C)	$\frac{t_{in} - t_{amb}}{H}$ (m ² C/W)	η	t_{amb} (C)	$\bar{H}$ (w/m ²)	$\bar{V}$ m/sec	Wind Dir.	$\bar{R.H.}$ (%)
date - 5/9/78 test 1:19 interval 1:34	40.69	20.22	58.99	0.000	0.489	20.28	900.5	2.65	S.E.	23
date - 5/13/78 test 11:27- interval 11:42	39.93	26.94	22.78	0.00507	0.472	22.78	820.5	2.50	S.E.	25
date - 5/14/78 test 12:00- interval 12:15	39.53	69.17	89.56	0.0472	0.263	28.83	855.8	2.94	S.E.	17
date - 5/14/78 test 12:45- interval 1:04	39.42	83.11	99.22	0.0620	0.206	29.76	861.1	4.08	S.E.	17
date - 5/14/78 test 1:50 interval 2:05	39.90	98.50	108.4	0.0801	0.131	30.75	846.2	4.21	S.E.	17
date - 5/15/78 test 13:27- interval 13:42	39.51	80.17	98.50	0.0556	0.233	31.97	866.9	1.78	S	19
date - 5/16/78 test 11:35- interval 11:54	39.92	49.44	76.61	0.0304	0.350	25.02	802.5	3.72	N.E.	46
date - 5/16/78 test 12:16- interval 12:36	39.76	51.94	79.50	0.0296	0.362	26.96	845.3	4.15	N.E.	46

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B-8

NOMENCLATURE

- A - Area; A_a absorber area; A_c free flow area; A_{fin} extended surface area; A_g gross collector area; $A_t = A_a + A_{fin}$; $A = A_t/A_a$; A_n nozzle area
- a - aspect ratio of duct b/s
- b - duct width
- C - total cost of flow duct
- c - nozzle discharge coefficient
- c_p - specific heat of air
- D - jet diameter; D_H hydraulic diameter; D nozzle throat diameter (in.)
- d' - honeycomb depth
- d - fin thickness
- F_{12} - gray body shape factor between surfaces 1 and 2
- F' - plate efficiency factor
- F_R - heat removal factor
- f - friction factor
- G - mass flow rate per unit area = $\dot{m}/A_g$
- g_c - gravitational constant
- h - heat transfer coefficient; h_a convective coefficient between absorber and air flow; h_b convective coefficient between back plate and air flow; h_g convective coefficient; h_o overall coefficient between absorber/back plate and air; between glass and air flow; h_w wind coefficient; h_v volumetric heat transfer coefficient; h_r radiative heat transfer coefficient; h_{rag} radiative heat transfer coefficient between the absorber and glass
- H - total solar radiation on a tilted surface
- K_1 - $0.0182 k_{air} \Delta T_{pa} Pr^{0.4}$ ΔT_{pa} - temperature difference between the absorber and air stream

- k - thermal conductivity
- L - collector length or matrix depth
- L' - honeycomb length
- l - fin length
- m - $(2h/kd)^{1/2}$
- $\dot{m}$ - mass flow rate
- Nu - Nusselt number hD_H/k ; Nu_∞ Nusselt number for fully developed turbulent flow; Nu_{ave} average Nusselt number in the entrance region of a flow duct
- P' - pressure drop across a flow nozzle (in. w.g.); P_n absolute pressure at the nozzle throat (in. Hg absolute).
- P - wetted perimeter
- p - voidage
- Pr - Prandtl number = $c_p \mu / k$
- Q - volumetric flow
- Q_u - useful energy collected
- Q_s - volumetric flow of standard air
- q - energy per unit area per unit time; q_a energy striking the absorber; q_L energy lost to the ambient; q_u useful energy collected
- Re - Reynolds number = $V \rho D_H / \mu$
- r_h - hydraulic radius = $D_H / 4$
- St - Stanton number = $h / \rho V c_p = Nu / Re Pr$
- s - flow duct depth
- T - absolute temperature
- t - temperature; t_a mean absorber temperature; t_{amb} ambient air temperature; t_b mean back plate temperature; t_f bulk fluid temperature; t_i inlet bulk air temperature; t_o outlet bulk air temperature; t_1, t_2 mean temperature of surfaces 1 and 2; t_{OT} outlet air temperature as a function of time

- U - heat transfer coefficient; U_a air to ambient coefficient; U_b back loss coefficient; U_c matrix absorber to ambient coefficient; U_e edge loss coefficient; U_{ga} glass to air coefficient; U_L overall loss coefficient; $U_L = U_t + U_b + U_e$; U_t top loss coefficient between the absorber plate and ambient air
- V - mean velocity; V_a approach velocity; V_n nozzle throat velocity; V_{wind} wind velocity
- v' - specific volume of air at nozzle ($\text{ft.}^3/\text{lb}_m$ air-water mixture)
- v - specific volume of air at dry and wet bulb temperature conditions existing at nozzle at standard barometric pressure ($\text{ft.}^3/\text{lb}_m$ dry air)
- w' - honeycomb width
- X - spacing between jets
- x,y,z - coordinates in 3-dimensional space
- Z - criterion function = annual cost/useful energy collected
- Z' - absorber plate to jet plate spacing

Greek letters

- α - absorptance
- β - extinction coefficient
- γ - opening angle between corrugations
- δ - cell wall overcoat thickness
- ϵ - emittance
- η - collector efficiency = $Q_u/H A_g$
- η_{fin} - fin efficiency
- η_o - overall surface efficiency
- μ - viscosity of air

- ν - Kinematic viscosity = μ/ρ
- ρ - density of air
- σ - Stefan - Boltzman constant
- τ - transmittance
- T - collector time constant
- $(\tau\alpha)_e$ - effective transmittance absorptance product
- ΔP - pressure loss through a collector
- ΔT - temperature difference

REFERENCES CITED
(SECTIONS III THROUGH VII, APPENDIX A)

1. ASHRAE STANDARD 93-77, "Methods of Testing to Determine the Thermal Performance of Solar Collectors".
2. Beckman, W.A., "Radiation and Convection Heat Transfer in a Porous Bed" Journal of Engineering for Power, Trans. ASME, Series A. Vol. 90, Jan. (1965).
3. Chiou, J.P., M.M. El-Wakil, J.A. Duffie, "A Slit-and-Expanded Aluminum Foil Matrix Solar Collector", Solar Energy, Vol. 9, No. 2, 1965.
4. Elsherbiny, S.M., Hollands, K.G.T., Raithby, G.D., "Free Convection Across Inclined Air Layers With One Surface V-Corrugated", was to be presented at ASME Winter Annual Meeting, November 27, 1977, Atlanta, Georgia.
5. Gupta, G.L., H.P. Garg, "Performance Studies on Solar Air Heaters", Solar Energy, Vol. 11, No. 1, 1967.
6. Hamid, Y.H., W.A. Beckman, "Transpiration Cooling of Radiatively Heated Porous Bed", ASME publication number 69-WA/Sol-6, (1971).
7. Hamid, Y.H., W.A. Beckman, "Performance of Air-Cooled Radiatively Heated Screen Matrices", Journal of Engineering for Power, April, (1971).
8. Hollands, K.G.T., "Directional Selectivity, Emittance, and Absorptance Properties of Vee Corrugated Specular Surfaces", Solar Energy, Vol. 7, (1963).

9. Hollands, K.G.T., T.E. Unny, G.D. Raithby, L. Konicek, "Free Convective Heat Transfer Across Inclined Air Layers", Journal of Heat Transfer, Vol. 98, No. 2, May, (1976).
10. Jones, Dennis, "Calculation of Performance of N Collectors in Series from Test Data on a Single Collector" Unpublished paper, NBS, Thermal Engineering Section, October, 1977.
11. Kays, W.M., A.L. London, Compact Heat Exchangers, McGraw Hill, (1955).
12. Kreith, F., Principles of Heat Transfer, Second Edition, 1965; International Textbook Company.
13. Kuzay, T.M., M.A.S. Malik, S.M. Ridemour, K.W. Boer, part 4: "Experimentation with Solar Collectors", Institute of Energy Conversion, University of Delaware, (1974).
14. Loth, J.L., "A Grooved Foámglas Flat-Plate Solar Collector", Proceedings: 1977 Flat-Plate Solar Collector Conference, February 28 - March 2, 1977, Orlando, Florida.
15. Malik, M., "Hydrodynamic and Heat Transfer Characteristics of A Solar Heated Air Duct", PhD. Thesis, Michigan State University, (1967).
16. Pescod, D., "Effects of Turbulence Promoters on the Performance of Plate Heat Exchangers", Heat Exchangers: Design and Theory Sourcebook, McGraw-Hill Co., 1974.
17. Rask, D.R., Mueller, L.J., Pejsa, J.H., "Low Cost Solar Air-Heater", Final Report July 22, 1977, ERDA Contract No. EY-76-C-02-2929.

18. Tan, H.M., W.W.S. Charters, "Effect of Thermal Entrance Region on Turbulent Forced-Convective Heat Transfer for an Asymmetrically Heated Rectangular Duct with Uniform Heat Flux", Solar Energy, Vol. 12, (1969).
19. Tan, H.M., Charters, W.W.S., "An Experimental Investigation of Forced Convective Heat Transfer for Fully-Developed Turbulent Flow in a Rectangular Duct with Asymmetric Heating", Solar Energy, Vol. 13, pp. 121-125, 1970.
20. Whillier, A., Chapter VIII, Applications of Solar Energy for Heating and Cooling of Buildings, "Prediction of Performance of Solar Collectors", ASHRAE GRP 170, 1977.

BIBLIOGRAPHY

1. Angilletta, D.J., "Venetian Blind Air-Heater", Patent No. 4,002,159., Nov. 7, 1975.
2. ASHRAE Standard 37-69, "Methods of Testing Unitary Air Conditioning and Heat Pump Equipment", American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 345 East 47th St., New York, N.Y. 10017, July 1964.
3. ASHRAE Standard 41.1, "Standard Measurement Guide: Section on Temperature Measurements", American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. 345 E. 47th Street, New York, N.Y. 10017, March, 1974.
4. ASHRAE Standard 93-77, "Method of Testing to Determine the Thermal Performance of Solar Collectors", American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. 345 E. 47th Street, New York, N.Y. 10017, February, 1977.
5. Beckman, W.A., "Radiation and Convection Heat Transfer in a Porous Bed", Journal of Engineering for Power, Trans. ASME, Series A, Vol. 90, Jan. (1965).
6. Beville, V.D., Brandt, H., "A Solar Energy Collector for Heating Air", Solar Energy, Vol. 12, pp. 19-29, 1968.
7. Bourne, R.C., "Louvered Air-Heater", Patent No. 3,971,359. July 27, 1976.
8. Buchberg, H., Catton, I., Edward, K.K., "Natural Convection in Enclosed Spaces: A Review of Application to Solar Energy Collection", ASME paper No. 74-WA/Ht-12., August, 1974.

9. Charters, W.W.S., "Some Aspects of Flow Duct Design for Solar Air-Heater Applications", Solar Energy, Vol. 13, pp. 283-288, 1971.
10. Charters, W.W.S., Lim, T.M., Tong, Y.C., "The Design Evaluation of Flat Plate Solar Air-Heater Absorbers by Digital Computer", Mechanical and Chemical Engineering Transactions, I.E. Australia, Vol. MC8, No. 1, pp. 55-58, May, 1972.
11. Charters, W.W.S., R.W.G. MacDonald, "Heat Transfer Effects in Solar Air Heaters", Publication unknown.
12. Charters, W.W.S., "Theory and Performance of Solar Air-Heaters", Unpublished paper.
13. Chiou, J.P., M.M. El-Wakil, J.A. Duffie, "A Slit-and-Expanded-Aluminum Foil Matrix Solar Collector", Solar Energy, Vol. 9, No. 2, (1965).
14. Close, D.J., "Solar Air Heaters for Low and Moderate Temperature Applications", Solar Energy, Vol. No. 2, (1963).
15. Cole-Appel, B.E., R.D. Haberstroh, "Performance of Air-Cooled Flat Plate Collectors", Proceedings of the Winnipeg Conference American Section of ISES, August, 1976.
16. Cope, W.F., "The Friction and Heat Transmission Coefficients of Rough Pipes", Proc. Inst. Mech. Engrs., Vol. 145, (1941).
17. Duffie, J.A., W.A. Beckman, Solar Energy Thermal Processes, Wiley-Interscience, (1974).
18. Eckert, R.G., A.J. Diaguila, "Convective Heat Transfer for Mixed Free and Forced Flow Through Tubes", Trans. ASME, Vol. 76 (1954).

19. Hewitt, H.C., "Performance Tests of a Solar Energy Collector Used to Heat Air", Proceedings of the 1977 Annual Meeting, American Section of ISES, June 6-10, 1977, Vol. 1, Section 2.
20. Hollands, K.G.T., "Directional Selectivity, Emittance, and Absorptance Properties of Vee Corrugated Specular Surfaces", Solar Energy, Vol. 7, (1963).
21. Hollands, K.G.T., T.E. Unny, G.D. Raithby, L. Konicek, "Free Convective Heat Transfer Across Inclined Air Layers", Journal of Heat Transfer, Vol. 98, No. 2, May, (1976).
22. Hottel, H.C., Woertz, B.B., "The Performance of Flat-Plate Solar Heat Collectors", ASME Transactions, Vol. 64, p. 91, 1942.
23. Johnston, J.G., "Matrix Air-Heater", Patent No. 3,875,925., April 8, 1975.
24. Jones, Dennis, "Calculation of Performance of N Collectors in Series from Test Data on a Single Collector", Unpublished paper, NBS, Thermal Engineering Section, October, 1977.
25. Jordan, R.C., Liu, B.Y.H., Applications of Solar Energy for Heating and Cooling of Buildings, ASHRAE GRP 170, 1977.
26. Elsherbiny, S.M., Hollands, K.G.T., Raithby, G.D., "Free Convection Across Inclined Air Layers with One Surface V-Corrugated", was to be presented at the ASME Winter Annual Meeting, Atlanta, Georgia, Nov. 27, 1977.
27. Edwards, D.K., Solar Collector Design, Solar Energy Services, Academic Press, 1977.
28. Gardner, J.G., "Embossed Absorber", Patent No. 3,561,425., Feb. 9, 1971.

29. Gupta, C.L., H.P. Garg, "Performance Studies on Solar Air Heaters", Solar Energy, Vol. 11, No. 1, (1967).
30. Hamid, Y.H., W.A. Beckman, "Transpiration Cooling of Radiatively Heated Porous Bed", ASME publication number 69-WA/Sol-6, (1971).
31. Hamid, Y.H., W.A. Beckman, "Performance of Air-Cooled Radiatively Heated Screen Matrices", Journal of Engineering for Power, April, (1971).
32. Hedstrom, J.C., "Single Solar Collector Results", Unpublished paper, Los Alamos Scientific Laboratory, May 1976.
33. Henderson, S.M., Khe, C.V., "Performance of a Matrix Solar Collector for Heating Air", American Society of Agricultural Engineers Annual Meeting, June 22-25, 1975.
34. Kays, W.M., A.L. London, Compact Heat Exchangers, McGraw Hill, (1955).
35. Karaki, S., G.O.G., Lof, C.C. Smith, "Comparative Performance of Solar Heating with Air and Liquid Systems", Interim Progress Report for the period February 1, 1976 to August 30, 1976. Solar Energy Applications Laboratory, Colorado State University, Fort Collins, Colorado.
36. Keyes, J.H., Strickland, C.I., Strickland, R.G., "Cup Air-Heater", Patent No. 3,946,720., March 20, 1976.
37. Kreider, J.F., Kreith, F., Solar Heating and Cooling: Engineering Practical Design and Economics, Scripta Book Company, 1975.
38. Kreith, F., Principles of Heat Transfer, International Textbook Co., Second Edition, May, 1967.

39. Kuzay, T.M., A.S. Malik, S.M. Ridenour, K.W. Boer, "Solar One: First Results, Part 4: Experimentation with Solar Collectors", Unpublished Paper, Institute of Energy Conversion, University of Delaware.
40. Lalude, O., H. Buchberg, "Design and Application of Honeycomb Porous-Bed Solar-Air Heaters", Solar Energy, Vol. 13, (1971).
41. Lof, G.O.G., "Solar Energy Utilization for House Heating", W.P.B. Report No. PB 25357, Office of the Publication Board, Dept of Commerce, 1946.
42. Lof, G.O.G., "Performance of Solar Energy Collectors of Overlapped-Glass-Plate Type", Proceedings Space Heating with Solar Energy, MIT, 1950, pp. 72-86.
43. Lof, G.O.G., Irish, E., "Heat Transfer from Parallel Plates to Air Flowing at Low Velocity", AICh E Annual Meeting, Columbus, 1950.
44. Lof, G.O.G., Nevens, T.D., "Heating of Air by Solar Energy", Ohio Journal Science, 53, pp. 272, 1953.
45. Lof, G.O.G., "Performance of Solar Energy Collectors of Overlapped-Glass-Plate Type", MIT Solar Heating Symposium, Open Space Heating with Solar Energy, MIT, 1954.
46. Lof, G.O.G., "House Heating and Cooling with Solar Energy", Solar Energy Research, pp. 33-46, U. of Wisconsin, 1955.
47. Lof, G.O.G., "Domestic Hot Air Heating with Overlapped-Plate Solar Collector", Proceedings World Symposium on Applied Solar Energy, Stanford Research Institute, 1956.

48. Lof, G.O.G., El-Wakil, M.M., Chiou, J.P., "Residential Heating with Solar Heated Air - The Colorado Solar House", Trans. ASHRAE, October, 1963, p. 77-86.
49. Lof, G.O.G., "Solar Air-Heaters", Proceedings of the Solar Heating and Cooling of Buildings Workshop, Washington, D.C., March 21-23, 1973.
50. Lof, G.O.G., "Air-Heater with Turbulator", Patent No. 4,072,142. Feb. 7, 1978.
51. Lof, G.O.G., "Dimpled Air-Heater", Patent No. (has not been issued to date).
52. Los Alamos Scientific Laboratory, Solar Energy Group, University of California, Los Alamos, New Mexico, ERDA's Pacific Regional Solar Heating Handbook, 2nd Edition, November, 1976.
53. Loth, J.L., "A Grooved FoamglasTM Flat Plate Solar Collector" Proceedings: 1977 Flat-Plate Solar Collector Conference, February 28 - March 2, 1977, Orlando, Florida.
54. Miller, K.W., "Overlapped Plate Collector" Patent No. 2,680,437. Patent No. 2,680,437., June 8, 1954.
55. Moan, K.L., Pei, Y.K., "Evacuated Tube Air-Heater", Patent No. 3,960,136., June 1, 1976.
56. Moan, K.L., "An Analysis of the Low Loss Evacuated Tubular Collector Using Air as the Heat Transfer Fluid", paper presented at the 1977 Annual Meeting of the American Section of ISES, June 10, 1977, Orlando, Florida.

57. Moreau, E., "Wire Gauze Air-Heater", Patent No. 201,439. March 19, 1878.
58. Parker, L.W., "Nested Absorber Air-Heater", Patent No. 3,919,998. Nov. 18, 1975.
59. Parker, B.F., "A Solar Panel for Residential Use", Proceedings of the 1977 Annual Meeting, American Section of ISES, June 6-10, 1977, Vol. 1, Section 2.
60. Pescod, D., "Effects of Turbulence Promotors on the Performance of Plate Heat Exchangers", Reprinted from Afgan, N.H. and Schlunder, E.V., "Heat Exchangers: Design and Theory Sourcebook", McGraw-Hill Book Co., (1974).
61. Pyle, D.L., "Box Style Air-Heater", Patent No. 3,902,474., Sept. 2, 1975.
62. Rankine, A.D., Charters, W.W.S., "Combined Convective and Radiative Heat Losses from Flat-Plate Solar Air Heaters", Solar Energy, Vol. 12, pp. 517-523, 1969.
63. Rask, D.R., Mueller, L.J., Pejisa, J.H., "Low Cost Solar Air-Heater", Final Report, ERDA Contract No. EY-76-C-02-2929. June 29, 1976 through July 22, 1977.
64. Rom, F.E., "Plastic Film Air-Heater", Patent No. 3,908,631., Sept. 30, 1975.
65. Safdari, Y.B., "Finned Absorber Air-Heater", Patent No. 4,019,494. Aug. 11, 1976.
66. Satcunanathan, S., Deonaraine, S., "A Two Pass Solar Air-Heater", Solar Energy, Vol. 15, pp. 41-49, May, 1973.

67. Sayigh, A.A.M., Solar Energy Engineering, Academic Press, 1977.
68. Schoenfelder, J.L., "Gang-Nail Air-Heater", Patent No. 3,863,621., Feb. 4, 1975.
69. Selcuk, K., "Thermal and Economic Analysis of the Overlapped-Glass Plate Solar Air-Heater", Solar Energy, Vol. 13, pp. 165-191, 1971.
70. Shoemaker, M.J., "Notes on a Solar Collector with Unique Air Permeable Media", Solar Energy, Vol. No. 4, p. 138, 1961.
71. Shoemaker, M.J., "Matrix Type Air-Heater", Patent No. 3,102,532., Sept. 3, 1963.
72. Suri, R.K., Saini, J.S., "Performance Prediction of Single and Double Exposure Solar Air-Heaters", Solar Energy, Vol. 12, pp. 525-530, 1969.
73. Sobel, A.T., Buelow, F.H., "Galvanized Steel Roof Construction for Solar Heating", Agricultural Engineering, June, 1973.
74. Tabor, H., "Radiation, Convection and Conduction Coefficients in Solar Collectors", Bull. Res. Council of Israel, Vol. 6C, 1958.
75. Tan, H.M., W.W.S. Charters, "Effect of Thermal Entrance Region on Turbulent Forced-Convective Heat Transfer for an Asymmetrically Heated Rectangular Duct with Uniform Heat Flux", Solar Energy, Vol. 12, (1969).
76. Tan, H.M., Charters, W.W.S., "An Experimental Investigation of Forced Convective Heat Transfer for Fully-Developed Turbulent Flow in a Rectangular Duct with Asymmetric Heating", Solar Energy, Vol. 13, pp. 121-125, 1970.
77. Thomason, H.E., "Trickle Collector Absorbers", Patent No. 3,145,707., August 25, 1964.

78. Ward, J.C., Lof, G.O.G., "Long-Term (18 Years) Performance of a Residential Solar Heating System", Solar Energy, Vol. 18, No. 4, 1976.
79. Whillier, A., "Black-Painted Solar Air Heaters of Conventional Design", Solar Energy, Vol. 8, No. 1, 1964.

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5/30/78

Confidential

D. K. Nowlin, Director, Special Programs Division
ATTN: GRACE

CONTRACT NO. EY-76-C-04-3713 Solaredon

Enclosed for your review and evaluation is:

- ☒ Contractor Report No 9
- ☒ Contractor Voucher \$21,112.00 (No 3437)
- ☐ Contractor Proposal _____
- ☐ Contractor Modification _____
- ☐ Other _____

At your earliest convenience, please advise and/or furnish this office:

- ☒ SPD's concurrence that the attached report is acceptable.
- ☒ SPD's concurrence that the above billing is compatible with work performed by the Contractor to date.
- ☐ SPD's programmatic evaluation and cost estimate of the attached contractor proposal. Your response should include, if applicable, an evaluation of technical skills by man-hours, materials, travel, consultants, subcontractors, etc.
- ☐ SPD's concurrence that the attached contract modification is compatible with programmatic requirements and agreements of the parties.

If the attached document is acceptable to SPD, please indicate your concurrence by signing and returning to this office one copy of this memo.

Charles E. Thurman
Contract Administrator
Contracts & Procurement Division

SPD Approval:

(Name)

(Title)
7/12/78

(Date)

May 26, 1978

Mr. Charles Thurman
Contract Administrator
Contracts and Procurement Division
Albuquerque, New Mexico 87115

Dear Mr. Thurman:

Attached is a copy of Report No. 9 per DOE Contract EY-76-C-04-3713. Report No. 9 is also Section VI of the final report and can be inserted into the rough draft for continuity. I am presently preparing Section VII which will conclude the final report. Upon my receipt of the rough draft from Los Alamos I will revise the final report and it will be submitted for publication by NTIS. I have attached the invoice for Report No. 9. If you have questions regarding this matter please call me at (303) 759-0101.

Sincerely,

Bruce E. Cole-Appel
Product Development Engineer

BECA/lcm
Encl.

UNITED STATES
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
ALBUQUERQUE OPERATIONS OFFICE
ALBUQUERQUE, NEW MEXICO 87115

W. P. Jones
Contract

D. K. Nowlin, Director, Special Programs Division

ATTN: *P. Grace*

CONTRACT NO. *EY-76-(0)-04-3713 - SOLARIM!*

Enclosed for your review and evaluation is:

- ☒ Contractor Report *No 6 (Voucher 24,078 paid)*
☐ Contractor Voucher _____
☐ Contractor Proposal _____
☐ Contractor Modification _____
☐ Other _____

At your earliest convenience, please advise and/or furnish this office:

- OK*
mt
- ☒ SPD's concurrence that the attached report is acceptable.
☐ SPD's concurrence that the above billing is compatible with work performed by the Contractor to date.
☐ SPD's programmatic evaluation and cost estimate of the attached contractor proposal. Your response should include, if applicable, an evaluation of technical skills by man-hours, materials, travel, consultants, subcontractors, etc.
☐ SPD's concurrence that the attached contract modification is compatible with programmatic requirements and agreements of the parties.

If the attached document is acceptable to SPD, please indicate your concurrence by signing and returning to this office one copy of this memo.

Charles E. Sumner

Contract Administrator
Contracts & Procurement Division

SPD Approval:

W. P. Jones
(Name)

Sr. Prog. Coord.
(Title)

12-27-77
(Date)

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Report No. 5

Instrumentation and Test Set-up

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INSTRUMENTATION & TEST SET UP

Introduction

The comparison of different collector designs is best accomplished by simultaneous, side-by-side testing, shown schematically in Figure 1. Since collector performance is a function of ambient conditions; insolation, wind speed and ambient temperature, and these parameters are not controllable in an outdoor test, it is highly desirable to compare performance under identical operating conditions. A simultaneous test is especially important since the expected increase in collector efficiency is 10 to 15 percent in comparison to the baseline collector, and the data are likely to have scatter of this same magnitude.

Testing Objective

Testing will be done in two phases as prescribed in the contract. The first phase is designed to determine which absorber concept has the greatest benefit-to-cost ratio, in comparison to the baseline collector. Single collector modules will be tested in this first testing phase. For comparison purposes, this type of testing is adequate as long as the forced convective heat transfer coefficients between the air and absorbers are maintained at the values which would be realized under field conditions. In other words, the Reynolds number associated with a particular design must be the same as that which a field installed collector would experience. The second and final phase of testing will be a 2 x 2 array (4 collector modules) of the collector type (or combination of types) shown to be the most promising of those tested in the first phase. This second phase of testing will provide the performance curve necessary for system sizing and evaluation.

The test absorbers were designed to fit into the collector bodies of the current Solaron collector (Figure 2). The inlet and outlet conditions, and edge and back losses will therefore be nearly identical forcing the comparison to be based only on the difference in heat transfer effectiveness between the different absorber designs. The test absorbers were painted with 3M-Black Velvet paint and baked according to the paint manufacturer's specifications. Black Velvet was chosen for its ease of application and uniformity.

○ AVERAGE AIR TEMPERATURE
MEASUREMENT POINTS

TEST RIG FLOW SCHEMATIC

FIGURE 1

EXPLODED VIEW OF BASELINE SOLARON COLLECTOR

FIGURE 2

AVERAGE INLET & OUTLET AIR TEMPERATURES

FIGURE 3

Pretest Conditioning

Since air leakage affects the results of a collector test a thorough examination of the collectors and the test loop was made. A helpful method of finding leaks is to use a soap and water solution applied on areas where leaks are likely. The collectors were also subjected to several days of exposure under no-flow conditions to determine if there were any off-gassing problems.

Test Description

The first phase of testing consists of the following two tests: (1) A determination of the collector time constant and (2) A determination of the collector thermal performance curve. The test methods to be followed are those outlined in ASHRAE 93-77, "Methods of Testing to Determine the Thermal Performance of Solar Collectors" and its subsequently referenced standards (See list of References). The basic test procedures are outlined below.

(a) Test No. 1 - Collector time constant

In order to minimize the transient effects on the collector test results, it is necessary to determine the collector time response. The collector time constant (τ) is a measure of this phenomenon. The test procedure for finding τ is outlined below:

1. Air is circulated through the collector at an inlet temperature (t_{fi}) equal to the ambient temperature (t_{amb}). The air is continuously circulated until the outlet air temperature (t_{fe}) is constant and remains so for at least 15 minutes.
2. Under a steady solar flux of at least 250 Btuh/ft.², the collectors are abruptly shielded from the incident solar radiation.
3. With t_{fi} held constant, the exit air temperature (t_{fe}) is monitored as a function of time (t_{feT}) until

$$\frac{t_{feT} - t_{fi}}{t_{fe_{initial}} - t_{fi}} = 0.368 \quad (1)$$

4. The time required for the above equality to hold true is designated as the collector time constant.

(b) Test No. 2 - Collector Thermal Performance

The steady-state thermal performance of the collector can be described by the following governing equation.

$$q_u = A_a F_R (\tau\alpha)_e H - F_R U_L A_g (t_{fi} - t_{amb}) = mc_p (t_{fe} - t_{fi}) \quad (2)$$

The collector efficiency is defined as $\eta = q_u / HA_g$

$$\text{or } \eta = \frac{mc_p (t_{fe} - t_{fi})}{HA_g} = \frac{A_a}{A_g} F_R (\tau\alpha)_e - F_R U_L \frac{(t_{fi} - t_{amb})}{H} \quad (3)$$

Equation (3) shows that if the collector efficiency (η) is plotted as a function of $(t_{fi} - t_{amb})/H$, the y-intercept will be equal to $\frac{A_a}{A_g} F_R (\tau\alpha)_e$, the x-intercept equal to $\frac{A_a}{A_g} (\tau\alpha)_e / U_L$ and the slope equal to $F_R U_L$.

The test procedure for determining the performance line is as follows:

1. Air at constant inlet temperature is circulated through the collector at constant temperature for at least 15 minutes (or one time constant) prior to the period which will be used to compute one data point.
2. Data points symmetric with solar noon are taken with sufficiently different inlet air temperatures to determine the thermal performance curve.
3. The data points that establish the performance curves are efficiency values determined by averaging the air temperature rise $(t_{fe} - t_{fi})$ and the solar input (H) over a time period equal to or greater than the time constant for the particular collector.

4. The flow rate through the collectors is held constant for all data points that determine the performance curve. The ΔP associated with the collector at the test flow rate is recorded.

Collector Test Stand

The collector test stand is a simple A frame type structure with a 53° south sloping roof line. The south facing roof is capable of holding 10 collector modules (5 wide x 2 high). (See photograph accompanying this report, Figure 4.)

FIGURE 4
SOLARON TEST FACILITY

Instrumentation

The instrumentation used in the tests follows the guidelines set forth in ASHRAE 93-77. The measurement techniques are discussed in more detail in the paragraphs that follow. Three deviations from 93-77 have been made. They are: (1) The measurement of the direct beam radiation has been deleted. (2) The measurement of the wet bulb temperature in the nozzle throat has been deleted. (3) The accuracy of the temperature difference between inlet and outlet air is limited by the calibration accuracy of the recorder, i.e., $\pm 0.7F$.

A. Temperature Measurement.

All temperature measurements are made with copper-constantan, type T, special limit, thermocouple wire. Temperature measurements are made following the guidelines of ASHRAE 41.1 - 74, "Standard Measurements Guide, Section on Temperature Measurement". The limits of error are specified by the wire manufacturer to be as follows:

Temperature Range	Limits of Error
15 - 200F	$\pm 3/4F$
200 - 700F	$\pm 3/8\%$ (of temperature being measured)

The temperature difference between the collector inlet and outlet air is being measured by an averaging arrangement of thermocouples (Figure 3). The inlet and outlet ducts are each 8 inches in diameter in which there are eight thermocouples placed in equal cross sectional areas of the duct. The thermocouples are connected in parallel to a common terminal point from which thermocouple wires lead to the recording device. The measurement points are approximately 1 foot from the inlet and outlets of the collector at all of the test stations, and the ducts are insulated with 1-1/2 inch foil backed fiberglass insulation to minimize heat loss or gain from the ducts.

Surface measurements are made at the points shown in Figures 2 and 5. The thermocouples are located in shallow surface penetrations and then attached with a metal-filled cement. This cement is then painted with Black Velvet paint to match the rest of the absorbing surface.

Additional air temperatures are being measured in the air passageways (Figure 5). These measurement points are located directly opposite the surface thermocouples so that a surface to air ΔT may be determined. These temperature measurements will be useful in the determination of the flow development length.

FIGURE 5. - TEMPERATURE MEASUREMENT POINTS

Ambient air temperature is measured approximately five feet from the test structure at approximately mid-roof height. The thermocouple has been installed in a shaded, ventilated, environment which has been painted white to reflect the incident solar radiation.

B. Flow Measurement

The flow is measured independently at each of the five test stations. The nozzle boxes were constructed according to ASHRAE standard 37-69, "Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment," and a sketch of the box is shown in Figure 6.

FIGURE 6 - NOZZLE BOX

The nozzles are 2 inches in diameter (D), which meet the size specifications of ASHRAE 37-69. The pressure drop across the nozzle is measured with a 4 inch manometer subdivided into minor scale divisions of 0.02 in. of water. The temperature at the nozzle is measured with a thermocouple located directly below the inlet diffusion baffle (Figure 6).

Flow to the individual collectors is controlled by dampers placed downstream of the outlet averaging thermocouple. Flow is controlled by balancing the ΔP through the parallel flow circuits. Additional flow control is obtained by dampers located on the inlet and outlet sides of the blower.

The air flow rate through a nozzle can be found from the following simplifications of the Bernoulli equation, and have been extracted from ASHRAE 37-69.

$$Q = 1096 C A_n (\Delta P V')^{0.5} \quad (4)$$

$$\text{where } V' = 29.92 V / P_n (1+W) \quad (5)$$

The volumetric flow of standard air can be found from

$$Q_s = Q / (0.075 V') \quad (6)$$

The discharge coefficient from the nozzle based on air temperature and Reynolds number can be found from figures 7 and 8.

$$\text{where } Re = f V_n D$$

TEMPERATURE FACTOR
VS. DEGREES (F)
FIGURE 7

DISCHARGE COEFFICIENT VS.
REYNOLDS NUMBER
FIGURE 8

C. Insolation

Insolation is measured with an Eppley model PSP pyranometer. The instrument has the following characteristics as provided by the manufacturer.

sensitivity	10.58 microvolts/watt m ⁻²
impedance	650 ohms (approx.)
temperature dependence	± 1 percent for ambient temperatures between - 20C to 40C.
linearity	± 0.5% 0 - 2800 W/m ²
response time	1 second

cosine	$\pm 1\%$ from normalization 0-70° zenith angle) $\pm 3\%$ 70-80° zenith
orientation	no effect
mechanical vibration	up to 20g - no damage
calibration	approx. 1 cal/cm ² min

The pyranometer is mounted in a plane parallel to the test collectors.

D. Recorder

The data recording device is a Doric model 220 recorder. It has the following characteristics.

resolution	0.1 (F or C)
calibration accuracy	$\pm 0.7F$ or $\pm 0.4C$
stability for 6 months	$\pm 0.5F$ or $\pm 0.3C$
temperature stability	$\pm 0.0025\% \times \text{RDG} \pm 1.7 \mu\text{v}/^\circ\text{C}$
repeatability at a point	± 1 count (max.) during a 20 minute period

The recorder has been used only 5 months since it was recalibrated at the factory.

E. Wind Speed

Wind speed is measured with a Stewart Electronic Wind Speedometer. The Stewart speedometer is a continuous running integrator giving an indication of the immediate past average wind speed (integrated over the

past 5 minutes) measured by a 1/60th mile contacting anemometer. The scale range is 0-100 mph using a 0-1 milliampere DC meter of 2% accuracy.

Miscellaneous Equipment

A. Air Reconditioning Apparatus

Air leaving the collectors passes to a common air duct where it is collected and reconditioned to the proper temperature. If the air is to be cooled, water is circulated through a four row coil which will remove the required amount of heat. The heat removal is controlled by metering the proper quantities of hot and cold water through a mixing valve, thereby setting the water temperature which feeds the heat exchanger. This is shown schematically in Figure 9.

FIGURE 9
AIR RECONDITIONING APPARATUS

At higher inlet air temperatures it is necessary to add auxillary heat to the air to offset losses in the duct system. For example, to obtain a datum point of $(t_{in} - t_{amb})/H = 0.6$, with a value of $H = 250$ BTU/hr. ft.² and $t_{amb} = 50F$ an inlet temperature of 200F is required. To meet conditions such as this an industrial quality resistance duct heater (4.8 KW) is used and is controlled with a silicon controlled rectifier which is turned on-off at zero voltage. This device controls the inlet air temperature to within 1°C (1.8F).

B. Blower

The blower/motor used to circulate air through the collectors is a direct-drive, 1 HP, Dayton model No. 4C108. This fan is rated by the manufacture to deliver 610 cfm at 4 inches of static pressure. (A high pressure fan is required in a test loop such as this to overcome the relatively large ΔP across the flow nozzle.)

Measurement of Optical Properties

A. Transmittance

The glazing transmittance is measured using an Eppley 8-48 Black and White Pyranometer. The transmittance can be found by measuring the ratio of solar energy transmitted by the glazing to the total incident solar energy, or

$$\tau = \frac{\text{voltage, transmitted radiation}}{\text{Voltage, incident radiation}}$$

A reading is simply taken with the pyranometer behind the glazing and then with the pyranometer in front of the glazing and a ratio of output voltages is found.

The following specifications apply to the model 8-48 pyranometer:

sensitivity - approximately 11 microvolts/watt meter⁻².

linearity - $\pm 1\%$.

cosine response - $\pm 2\%$ from 0-70°.

B. Absorptance

The absorber plate absorptance (α) is measured with a model 1A, Devices and Services miniature pyranometer. The following relationship is used to compute the absorptance.

$$\alpha = 1 - \frac{\text{voltage, absorber plate reflectance}}{\text{voltage, incident radiation}}$$

This measurement is made by holding the pyranometer so that it receives the reflected solar radiation from the absorber. A subsequent reading of the total incident radiation is then measured, and the above formula applied.

The following specifications apply:

cosine response - $\pm 2\%$ 0-60°

spectral response - 0.3 - 2 μ m

C. Emittance

The absorber plate emittance (ϵ) is measured with a model AE, Devices and Services Emissometer. The emissivity of the unknown surface is measured after calibration of the detector with a sample of known emissivity, which is maintained at the same temperature as the unknown sample. The standard for comparison is a piece of aluminum supplied by the manufacturer and has a long wavelength emissivity of 0.93. The emissivity of the sample is calculated from the following equation.

$$\epsilon_{\text{sample}} = \frac{V_{\text{sample}}}{V_{\text{standard}}} \times 0.93$$

The detector output is linear with emissivity to within ± 0.01 units according to the manufacturers specifications.

Closure and Error Analysis

Since computation errors have a compounding effect on the final results, it is desirable to determine the expected limits of accuracy on a plot of efficiency (η) versus $(t_{fi} - t_{amb})/H$. The following equations for computing the resulting errors (R) were taken from (4).

$$(A \pm a) + (B \pm b) \quad R = \pm (a^2 + b^2)^{0.5}$$

$$(A \pm a)(B \pm b) \quad R = \pm \{(Ab)^2 + (Ba)^2\}^{0.5}$$

$$(B \pm b)/(A \pm a) \quad R = \pm \left\{ \frac{(Ba/A)^2 + b^2}{A^2} \right\}^{0.5}$$

where A,B are the measured quantities and a,b are the tolerances on the individual measurements.

For the following typical test conditions (and specified limits of error) the resulting errors using the above equations are as follows:

Case 1: $t_{fi} = 85 \text{ F} \pm 3/4 \text{ F}$

$$t_{amb} = 80 \text{ F} \pm 3/4 \text{ F}$$

$$t_{fo} = 118 \text{ F} \pm 3/4 \text{ F}$$

$$H = 275 \text{ Btu/hr. ft.}^2 \pm 1\%$$

$$\Delta P = 1.06 \text{ i.w.g.} \pm 0.01$$

$$T_{noz} = 570 \text{ R} \pm 3/4 \text{ R}$$

$$G = 18 \text{ lbm/hr. ft.}^2$$

$$\eta = q_u/HA_g = 0.518 \pm 0.018 \quad (t_{fi} - t_{amb})/H = 0.018 \pm 0.0038$$

Case 2:

$$t_{fi} = 200 \text{ F} \pm 3/4 \text{ F}$$

$$t_{fo} = 206.4 \text{ F} \pm 3/4 \text{ F}$$

$$t_{amb} = 85 \pm 3/4 \text{ F}$$

$$H = 275 \text{ Btu/hr. ft.}^2 \pm 1\%$$

$$\Delta P = 1.22 \text{ i.w.g.} \pm 0.01$$

$$T_{noz} = 654 \text{ R} \pm 3/4 \text{ R}$$

$$G = 18 \text{ lbm/hr. ft.}^2$$

$$\eta = 0.10 \pm 0.017 \quad (t_{fi} - t_{amb})/H = 0.418 \pm 0.0038$$

The examples are for the two extreme cases of collector efficiency, i.e., the efficiencies associated with a low air inlet temperature and a high air inlet temperature. It is noted that the overall resulting errors are nearly constant for both cases.

It should be noted that the ASHRAE requirement for measuring the temperature rise of the fluid to 0.2F may not be as applicable to the testing of air collectors as it is to the testing of water heating collectors. The reason for this is that air collectors typically have a temperature rise of at least four times that of an equivalent water heater operating under identical conditions. For an equivalent percent error, the air collector test could withstand a tolerance of $\pm 0.80\text{F}$ on the individual temperature measurements.

NOMENCLATURE

A_a	- net absorber area (ft. ²)	Re	- Reynolds number
A_g	- gross collector area (ft. ²)	T	- temperature (F, R)
A_n	- nozzle throat area (ft. ²)	t_{fi}	- fluid inlet temperature (F)
c_p	- specific heat of air (Btu/lbmF)	t_{fe}	- fluid outlet temperature (F)
C	- nozzle Discharge coefficient	t_{amb}	- ambient temperature (F)
D	- nozzle throat diameter (in.)	t_{feT}	- fluid temperature at time T (F)
f	- temperature factor	U_L	- overall top loss coefficient based on absorber plate temperature (Btu/hr. ft. ² F)
F_R	- heat removal factor $F_R = \frac{Gc_p}{U_L} (1 - e^{-F'U_L/Gc_p})$	V	- specific volume of air at nozzle at standard barometric pressure (ft. ³ /lb.)
F'	- plate efficiency factor (see Duffie and Beckman)	V'	- Specific volume of air (ft. ³ /lbm)
G	- $\dot{m}/A_g$ (lbm/hr. ft. ²)	V_n	- velocity at nozzle throat (fpm)
H	- insolation (Btu/hr. ft. ²)	W	- humidity ratio at nozzle (lbs. moisture/lb. dry air)
$\dot{m}$	- mass flow (lbm/hr.)		
P_n	- pressure at nozzle throat (in. Hg)		
Q	- measured volumetric flow (cfm)		
q_u	- useful energy collected (Btu/hr.)		
Q_s	- air flow, standard air (cfm)		

GREEK LETTERS

α - absorptivity

ϵ - emmissivity

τ - transmittance

$(\tau\alpha)_e$ - effective transmittance - absorptance product

η - collector efficiency

ΔP - pressure difference across flow nozzle (i.w.g.) or across collector

ΔT - temperature difference between absorber plate and airflow (F)

LIST OF REFERENCES

1. "Methods of Testing to Determine the Thermal Performance of Solar Collectors", ASHRAE Standard 93-77, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 345 East 47th Street, New York, N.Y. 10017, February, 1977.
2. "Standard Measurements Guide, Section on Temperature Measurement", ASHRAE 41. 1-74, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 345 East 47th Street, New York, N.Y. 10017, March, 1974.
3. "Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment", ASHRAE Standard 37-69, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., April, 1969.
4. ASHRAE Handbook of Fundamentals, Section on measurements, p. 197, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1972.

BIBLIOGRAPHY

1. Benedict, Robert P., Fundamentals of Temperature, Pressure, and Flow Measurements, John Wiley and Sons Inc., 1969.
2. Duffie, J.A., W.A. Beckman, Solar Energy Thermal Processes, John Wiley & Sons, New York, 1974.