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HEAT PIPES APPLIED TO FLAT-PLATE SOLAR COLLECTORS

Final Report

W. B. Bienert
D. A. Wolf

May 1976

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Dynatherm Corporation
Cockeysville, Maryland

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ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Division of Solar Energy

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FINAL REPORT
for
HEAT PIPES APPLIED TO FLAT-PLATE
SOLAR COLLECTORS

Walter B. Bienert
David A. Wolf

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ABSTRACT

The objective of this program was to investigate the use of heat pipes as heat transport elements in flat-plate solar collectors. During the early phases of the program, analytical models for heat pipe collectors were developed and configuration and integration studies were performed. The feasibility of wickless heat pipes, fabricated from commercial Roll-Bond material, was investigated in the laboratory. These heat pipes performed satisfactorily with Freon 11 as the working fluid. Two complete heat pipe solar collectors, which utilized converted Roll-Bond panels as heat pipes, were tested along with a conventional reference collector of the same basic design.

With steady solar input, the heat pipe and the reference collector yielded approximately the same instantaneous efficiency (~60%). But with intermittent insolation, such as on a partly cloudy day, the heat pipe collector collected up to twice as much energy as the reference collector. This improvement is attributable to the thermal diode capability of the heat pipes. Heat is transported only when the absorber temperature exceeds the fluid temperature in the manifold. During periods of low insolation, the heat pipe collector is automatically inactivated. This feature of a heat pipe collector can be utilized in the design of a simple stationary concentrator.

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I. INTRODUCTION

The objective of this program was to analytically and experimentally investigate the use of heat pipes in flat-plate solar collectors. Heat pipes are passive heat transport devices which utilize a closed evaporation-condensation cycle (Ref. 1). Because of their high equivalent conductance, they appear to be well suited to transport heat from the solar absorber to an air or liquid distribution system.

Several potential advantages are offered by heat pipe solar collectors. Gravity-assisted heat pipes are unidirectional conductors; that is, they behave as thermal diodes. If they are properly oriented, heat is transported only from the absorber surface to a collection manifold but never in the opposite direction. This feature reduces heat losses from shaded or defective absorber sections. Also because heat pipe panels are sealed systems, corrosion is minimized, freezing of exposed collector surfaces can be eliminated through working fluid selection, and mechanical attachment to either air or liquid collection manifolds is feasible. Conversely, potential disadvantages may include increased collector cost and lower effectiveness resulting from additional thermal interfaces.

The program consisted of the following tasks: (I) Configuration Studies, (II) Parametric Performance Studies, (III) Economic Analysis, (IV) System Integration Studies, (V) Submodule Fabrication and Testing (in the laboratory), and (VI) Full-Scale Module Fabrication and Testing (using solar input). An additional Task VII, Feasibility Study of a Stationary Concentrator, was identified during the program and was also completed. In performing Tasks I through IV, various aspects of integrating heat pipes into flat-plate solar collectors were investigated. The results of these tasks were reported in the Annual Progress Report (Ref. 2) dated January 31, 1975. A summary of that program effort is included in the present report. The results of the experimental work conducted under Tasks V and VI are presented in this report. Under Task V, bread-board heat pipes were fabricated from sections of Roll-Bond panels and their heat transfer performance was evaluated in the laboratory. Three complete solar panels, two of which were heat pipe absorbers and one was a Roll-Bond control panel, were fabricated and solar tested during Task VI. Finally, under the new Task VII, a fea-

sibility study of a stationary concentrator using heat pipes as thermal diodes was conducted.

The results obtained during this program have shown that a heat pipe collector has considerable merits. If properly designed, its thermal performance (as expressed for instance by the Collector Efficiency Factor F') is approximately equal to that of a good conventional collector which uses integral liquid passages in the absorber panel. The performance was demonstrated with Freon as the heat pipe fluid. Freon is non-toxic, noncorrosive, and poses no freezing problem. During periods of intermittent insolation, the heat pipe collector delivered up to twice as much energy as the conventional reference unit due to the thermal diode behavior of the heat pipes. Also demonstrated was the feasibility of mechanical interfacing of the sealed collector with a liquid collection loop. This feature can be expected to reduce installation and maintenance cost.

The analytical studies indicated that the heat pipe collector can be applied with advantage to air heating systems. When combined with a relatively simple Fresnel lens, the heat pipe collector can function as a stationary concentrator resulting in a substantial increase in the useful collected energy. Application of the heat pipe collector to an air system and testing of a concentrator was beyond the scope of this program. Also, a design optimization remains to be performed with the objective of minimizing fabrication cost. Initial estimates indicate that a heat pipe collector can be produced with an incremental cost increase of about \$1.00 per square foot.

II. SUMMARY

The program effort is subdivided into the following seven tasks:

- I. Configuration Studies
- II. Parametric Performance Analysis
- III. Economic Analysis
- IV. System Integration Studies
- V. Submodule Fabrication and Testing
- VI. Full-Scale Module Fabrication and Testing
- VII. Stationary Concentrator Studies

The research conducted during calendar year 1974 was reported in the Annual Progress Report (Ref. 2). During that period, Tasks I, II, III, and IV were completed. This present report details the program effort of the remaining tasks. Highlights of the results of all program tasks are summarized in the following paragraphs.

Task I. Configuration Studies

Several collector configurations, using heat pipes as energy transport devices, were evaluated for use with either air or liquid heating systems. The following three basic approaches to incorporate heat pipes with the absorber plate were studied:

- Individual heat pipes bonded to absorber plate
- Extruded heat pipes with integral absorber fins
- Heat pipes integral with 2-piece bonded absorber plate

The last approach using the commercial "Roll-Bond panel" (Ref. 3) was selected for further evaluation because it offers the best thermal performance and the most economical method of manufacture. Roll-Bond panels consist of two sheets of aluminum or copper bonded together and expanded in certain areas to form channels. The pattern of the vapor paths in a Roll-Bond panel can be readily chosen so as to minimize interface resistances between the absorber and the transfer loop.

Thermal interfacing between heat pipes and transfer loop is the most important design consideration in a heat pipe collector. Typically, the heat flux at the interface is one order of magnitude higher than the solar flux. Both liquid and air transfer loops were considered. In the case of a liquid loop, the optimum configuration is one in which the heat is transferred to a liquid manifold which is located along the top edge of the collector. The manifold can be either an integral part of the collector or can be mechanically attached through a mounting saddle. In case of an air transfer loop, the underside of the collector is equipped with fins which extend over the upper 12 to 16 inches of the panel. Heat is transferred to the air stream by convection from these finned surfaces.

Task II. Parametric Performance Studies

An analytical model was developed which permits rapid comparison of different collector and interface geometries in terms of heat transfer between absorber plate and transport fluid. The effectiveness of heat transfer is usually expressed in terms of the Collector Efficiency Factor (F') (Ref. 4). Analytical expressions were derived for F' as a function of heat pipe geometry, spacing, interfaces between collector and transport loop, and parameters of the transport loop. For heat pipe collectors serving a liquid transport system, the predicted F' factors are in the range of 0.85 to 0.90. These values also apply for mechanically attached manifolds provided that an interface conductance of approximately 1000 Btu/Hr-Ft²-°F is achievable. In the case of air heating systems, similar values for the F' factors were calculated; but collection efficiency is usually of less importance in air heating systems, since large temperature gradients occur inherently because of the much smaller mass flow rates which are commonly used. Thus, if the heat pipe collector introduces a small penalty in collection efficiency (F' /factors of 0.85 to 0.90 versus 0.95 for a good conventional collector), this penalty is least apparent in an air heating system. Various wick geometries and working fluids of the heat pipes were evaluated. Since flat-plate collectors are almost always tilted with respect to the horizontal, gravity provides sufficient driving potential for the condensate of the working fluid to return to the evaporators. Thus, the heat pipes can be wickless. With regard to the working fluid, the Freons provide the best choice. They are nontoxic, they are not corrosive with most metals including aluminum, and their freezing point is suffi-

ciently low (Ref. 5). But, a potential problem is pressure containment at high temperatures ($>150^{\circ}\text{C}$) which occur during stagnation conditions (no coolant flow). Careful design is required and the problem can be solved even with Roll-Bond as the heat pipe envelope.

Task III. Economic Analysis

Most components of a heat pipe collector are identical to those of an equivalent conventional collector panel. Therefore, an economic analysis is best based on the incremental cost associated with incorporating heat pipes. Since heat pipe processing is the largest cost item associated with heat pipes, a design which allows simultaneous processing of many channels is likely to be the most cost effective. For this reason, Roll-Bond panels have been selected as the baseline design. It is estimated that the incremental cost of such a heat pipe collector would be $\$8.80/\text{m}^2$ ($\$0.82/\text{ft}^2$) of collector area for a 3 m^2 (4' x 8') panel and $\$12.60/\text{m}^2$ ($\$1.17/\text{ft}^2$) for a 1.5 m^2 (2' x 8') panel. The above values apply to a heat pipe collector which is detachable from the fluid loop. If the manifold is fabricated as an integral part of the collector, the incremental cost is about 10% lower.

More important than the cost of the collector per unit area is the total cost of collected energy. This includes not only the cost of all components of a solar heating system but also installation, operation, and maintenance costs. The features of the heat pipe collector are expected to favorably impact the cost of several of the above items. However, a detailed systems analysis is required in order to quantify the cost savings.

Task IV. System Integration Studies

In cooperation with Dynatherm's consultants (the architectural firm of Wilson, Magruder, Webb and Ratych, of Towson, Maryland), integration of heat pipe solar collectors with typical roof constructions and HVAC systems was evaluated. Installation of panels between roof rafters and on top of standard sheathing was considered. Clean interfacing with a preinstalled liquid or air transport system is best achieved if the collector is recessed between rafters. This would also protect a water trans-

port line from freezing and, in combination with a freeze-proof collector, facilitate system design.

Another consideration in overall system design is the manifolding required to interconnect multiple collectors. In order to take full advantage of the modularization afforded by heat pipe collectors, a single straight manifold running across the entire roof is desirable. Analytical results show that, in a liquid system, up to eight collectors (each 4 feet by 8 feet) can be connected in series without exceeding recommended water flow rates and pipe sizes. With air heating systems, the series connection is limited to four collectors. If simple manifolding is an important objective, a vertical airflow over the collectors is the preferred arrangement with one supply manifold at the bottom and one collection manifold at the top of roof.

Task V. Submodule Fabrication and Testing

The studies performed during the earlier tasks in this program indicated that heat pipes fabricated from Roll-Bond panels represent the best choice for a heat pipe flat-plate solar collector. Under this task, a typical standard Roll-Bond panel was obtained and sectioned to yield two heat pipe configurations. One configuration consisted of a single flow channel of the Roll-Bond panel, and the other was made up of three of the panel's flow channels connected to a common manifold. Each heat pipe was fabricated by welding the flow channels closed on one end, attaching a fill tube and valve to the other end, and then cleaning, outgassing, and charging each unit with Freon-11.

As a result of extensive testing performed on these two heat pipe configurations, several conclusions regarding the feasibility of utilizing Roll-Bond panels as heat pipe absorber panels were reached. These conclusions are summarized below:

- Roll-Bond panels offer a feasible and economically attractive technique for incorporating heat pipes into the absorber plate.
- Roll-Bond flow channels possess sufficient heat transport capability without employing a wick, provided the heat pipe panel is favorably oriented.

- Heat pipe absorber panels utilizing a common manifold act as a series of individual and identical heat pipes under uniform panel illumination. For nonuniform illumination, temperature gradients between heat pipe legs may result. However, these gradients are not excessive and total panel operation is stable.
- The overall thermal resistance at the absorber panel-collection loop interface for typical interface configurations concurs with analytical predictions. The measured resistance appears to be acceptable for efficient operation of a heat pipe collector.

Task VI. Full-Scale Module Fabrication and Testing

Under this task, three solar absorber panels were fabricated from a typical Roll-Bond panel configuration. One panel was fabricated with no significant changes to the Roll-Bond configuration. It served as a reference to establish a comparative basis for the performance evaluation of a heat pipe solar collector. Another Roll-Bond panel was modified to yield a multi-legged heat pipe absorber panel. This panel featured a mechanically attached transport fluid manifold. Modifications to a third Roll-Bond panel resulted in a second heat pipe absorber panel characterized by individual heat pipes, expanded vapor chambers for condensation, and an integral transport fluid manifold. Three flat-plate solar collectors were constructed which were identical in all respects, except each contained one of the described absorber panels.

Based on the analytical model, it was recognized that the first of the two heat pipe absorbers would have a substantially higher thermal resistance than the reference Roll-Bond absorber. It was fabricated primarily to demonstrate the feasibility of a heat pipe absorber utilizing commercially available components. The second heat pipe panel was intended to be representative of the performance attainable with an optimized Roll-Bond absorber panel.

Extensive outdoor tests were conducted in both the steady-state and transient (intermittent insolation) mode of operation. Conclusions regarding the steady-state performance characteristics were as follows:

- The overall thermal resistance of the multi-legged heat pipe absorber panel was about six times greater than that of the reference panel. This resistance difference resulted in approximately a 5% reduction in the instantaneous collector efficiency.
- The overall thermal resistance of the heat pipe absorber panel characterized by individual heat pipe channels, expanded vapor chambers, and an integral transport fluid manifold was approximately twice as large as the resistance of the reference panel. This resistance difference resulted in only a 1% reduction in the instantaneous efficiency.

The transient behavior and performance of the reference collector and the second heat pipe collector were investigated. A mathematical model was formulated to predict the performance of each panel when subjected to a step function change in the solar insolation. A series of tests were simultaneously conducted on both collectors to determine their relative transient performance characteristics. The results of these tests emphasize that:

- The diode capability of a heat pipe absorber panel increases collector performance above that of an identical nondioding absorber panel.
- The behavior of each panel is in good agreement with the predictions of the analytical model.
- The heat pipe absorber panel experiences a "start-up" period characterized by incipient localized burnout and nucleate boiling.

Additional tests conducted during natural oscillations in the solar insolation indicate that, under certain ambient conditions, the diode characteristic of a heat pipe absorber panel can provide twice the energy supplied by an identical nondioding collector.

Task VII. Stationary Concentrator Studies

A feasibility study was conducted of a stationary concentrator utilizing heat pipes as thermal diodes. The concept consists of a series of cylindrical Fresnel lenses which provide moderate concentration in the absorber plane. Diode heat pipes collect energy only from those absorber sections which are in focus at a particular time. Out-of-focus heat pipes are automatically inoperative and do not contribute to any heat losses. As the sun angle changes, different heat pipes

come into focus and transport heat. The concentrator can be employed in either of two ways:

- Generate higher temperatures than achievable with conventional flat-plate collectors.
- Yield higher collection efficiencies at the temperatures obtainable from a flat-plate collector.

A numerical evaluation of the performance of such a stationary concentrator was conducted. East-West orientation and a constant elevation equal to the local latitude was assumed. A compromise had to be made as to the location of the absorber plane with respect to the focal plane. If absorber and focal planes coincide, high concentration ratios are achieved at near normal incidence; but, at large angles, the performance becomes very poor. A separation of lens and absorber planes of between 60 and 80% of the focal length was found to give the highest integrated energy at all angles.

The calculations indicated that, at a temperature difference between fluid output and ambient of 55°C , the stationary concentrator yields, on the average, 30% more useful energy than a flat-plate reference collector having identical parameters. If the temperature difference is raised to 80°C , the concentrator will double the output per unit of collector area. The above values are averages for the entire year. Higher improvement factors can be obtained if the concentrator is optimized for particular times of the day or particular seasons of the year.

III. TECHNICAL DISCUSSION

A detailed description of the work performed under Tasks I through IV is provided in the Annual Report (Ref. 1).

A. Task V. Submodule Fabrication and Testing

Based on the configuration studies and performance evaluations of earlier tasks, a Roll-Bond panel was selected as the envelope for a heat pipe absorber. A baseline design for such a solar collector was presented in a previous report (Ref. 1). For easy reference, two figures from that report are reproduced here. The first one (Figure III-1) shows the overall design of a heat pipe solar collector utilizing a Roll-Bond panel. The second one (Figure III-2) shows details of the interfacing between the absorber and a liquid collection loop. The important feature of this design is the purely mechanical interface between the collector and the liquid loop. Also, the heat pipe channels within the Roll-Bond panel do not contain a wick; refluxing of the working fluid occurs only by gravity forces.

For the experimental evaluation, a standard panel manufactured by the Olin Brass Corporation was purchased. The dimensions of this panel are shown in Figure III-3 which is a partial reproduction of an Olin Brass drawing. Instead of the desirable vapor chambers at the condenser which are shown in Figures III-1 and III-2, the standard panel was only equipped with straight channels. Also, the panel was expanded on both sides to form the channels. Ideally, it should be flat on one side for easier interfacing with the liquid loop.

Portions of the panel were subjected to laboratory testing to qualify the use of such panels as heat pipe solar absorbers. The specific objectives of these tests included:

- Verification of sufficient heat transport capability of Roll-Bond flow channels without a wick.
- Determination of the resistance components constituting the thermal interface between the heat pipe absorber panel and the collection fluid loop.

FIGURE III-1
 BASELINE DESIGN OF HEAT PIPE FLAT-PLATE COLLECTOR
 (All Dimensions in Inches)

a. Typical Top View

b. Side View with Circular Headers

c. Side View with Rectangular Headers

FIGURE III-2
WATER HEADER CONFIGURATIONS

FIGURE III-3
STANDARD ROLL BOND PANEL
(Partial Reproduction of Olin Brass Drawing)

- Determination of the effect of common vapor chambers during non-uniform heat input.

To accomplish these objectives, two test samples were made from the Roll-Bond panel. One sample consisted of an individual flow channel centrally located in a 3.8-cm wide section of the panel. This channel was welded closed on one end, and a fill tube and valve were attached to the other end. The other sample was prepared in a similar manner, except that it consisted of three channels which were centered on a 21-cm wide panel section. Each channel was sealed on one end and connected to a common manifold on the other end. A fill tube and valve were attached to this manifold. Both samples are shown in Figure III-4. Prior to testing, each sample was purged, evacuated, and charged with Freon-11.

After fabrication was completed, each sample was prepared for testing. In order to simulate solar insolation, a heater was attached to one side of each sample along the entire length of each heat pipe evaporator and its associated fin. Each sample was insulated to minimize heat losses to the surroundings. Heat was rejected by each heat pipe condenser to either a water bath, a chill block, or a water-header manifold. Each test setup is represented schematically in Figure III-5. For the water bath tests, thermocouples were placed along the length of the evaporator (3 locations typical) and along the condenser's length (2 locations typical). For the tests with the chill block as the heat sink, additional thermocouples were placed on the chill block. When the water-header manifold was used as the heat sink, thermocouples were also placed on the water manifold and the manifold saddle.

The objective of the first series of tests was to determine the optimum charge of Freon-11 and to verify the existence of adequate heat transport capability. The condenser end of the individual heat pipe test sample was submerged in a water bath as shown in Figure III-5a. The sample was oriented at an angle of 25 degrees relative to the horizontal (condenser above evaporator). The overall thermal resistance between the evaporator and condenser, determined by the temperature gradient as a function of the net heat input, was found to be lowest at a fluid charge level of 5.3 ml as indicated by Figure III-6. For this reason, subsequent tests on this sample were performed with this fluid inventory. The net power levels during these tests were as

TEST SAMPLE #1

TEST SAMPLE #2

FIGURE III-4
ROLL-BOND HEAT PIPE TEST SAMPLES

5a. Schematic of Test with Water Bath

5b. Schematic of Test with Chill Block

5c. Schematic of Test with Clamp-On Manifold

FIGURE III-5
TEST SCHEMATICS FOR HEAT PIPE COMPONENT TESTS

FIGURE III-6
OVERALL HEAT PIPE TEMPERATURE GRADIENT FOR
DIFFERENT FLUID INVENTORIES

high as 75 watts. This corresponds to an absorbed solar flux of 1000 watts/meters².

The next series of tests were conducted in order to determine the thermal resistance of the heat pipe and of the various heat sink interface configurations. The extent of the instrumentation employed during this test series enabled the determination of the significant components of the overall thermal resistance. These components are schematically represented in Figure III-7. Throughout these tests, the fluid charge in the individual heat pipe test sample was 5.3 ml of Freon-11.

The test results from this series of tests are summarized in Table III-1. With the condenser immersed in a water bath, the overall resistance of the heat pipe (combination of evaporator and condenser resistances) was measured to be 0.39°C/watt. The thermal interface resistance between the heat pipe absorber panel and an external heat sink was determined by the use of a chill block clamped to the heat pipe condenser. A thermal grease coating was applied to the condenser/chill block interface. The test setup is shown in Figure III-5b. The sample was tested in a vertical reflux orientation and at 25 degrees relative to the horizontal (condenser above evaporator). Heat pipe performance was identical at both orientations. The resistance at the thermal grease interface was measured to be 0.22°C/watt, and the overall heat pipe resistance during this test was only 0.26°C/watt. One possible explanation of the lower overall heat pipe resistance measured during this test is that the indicated condenser temperature during the previous test was influenced by the water bath.

Another test was conducted on this sample to determine the resistance associated with the implementation of a water manifold at the heat pipe condenser. This manifold, shown in Figure III-5c, consisted of a 1.91 cm copper pipe enclosed by an aluminum saddle (5.1 cm in width) extending over the width of the test sample. A thermal grease coating was applied to the interface between the manifold saddle and the heat pipe condenser and to the saddle/water pipe interface. The test results are also given in Table III-1. The overall heat pipe resistance and the resistance at the heat pipe/manifold saddle interface obtained during this test were in good agreement with the results obtained during the tests with the chill block. In addition, the resistance between the manifold saddle and the transport fluid was measured to be 0.14°C/

FIGURE III-7
SCHEMATIC OF RESISTANCES IN
HEAT PIPE FLAT-PLATE SOLAR COLLECTOR

TEST DESCRIPTION	Condenser Length (cm)	RESISTANCE ($^{\circ}\text{C}/\text{Watt}$)			
		Heat Pipe	Condenser-Saddle Interface	Saddle-Fluid Loop	Total
Test Sample #1, Water Bath	5.1	0.39	0.0	0.0	0.39
Test Sample #1, Chill Block	5.1	0.26	0.22	0.0	0.48
Test Sample #1, Water Manifold, Thermal Grease at Panel-Saddle Interface and Saddle-Fluid Loop Interface	5.1	0.28	0.24	0.14	0.66
	10.2	0.18	0.12	0.083	0.38
Test Sample #1, Water Manifold, Eccobond at Panel-Saddle Interface and Thermal Grease at Saddle-Fluid Loop Interface	5.1	0.26	0.23	0.14	0.63
Test Sample #2, Water Bath	5.1	0.38	0.0	0.0	0.38

*Two Water Manifolds

TABLE III-1
SUMMARY OF HEAT PIPE AND INTERFACE RESISTANCES

watt. This test was conducted at heat pipe vapor temperatures of 49°C and 71°C , and no noticeable effect on the overall resistance was observed. The water manifold test was repeated using "Eccobond" at the heat pipe/manifold saddle interface. A slight decrease in the overall resistance resulted. This test sample was also tested with two water manifolds attached to the heat pipe condenser. As the results in Table III-2 show, the overall resistance is decreased markedly.

Summarizing the test results (reported in Table III-1) for the individual heat pipe test sample with a condenser length of 5.1 cm, the overall heat pipe resistance is $0.27^{\circ}\text{C}/\text{watt}$, the heat pipe/manifold saddle interface resistance is $0.23^{\circ}\text{C}/\text{watt}$, and the resistance between the manifold saddle and the transport fluid is $0.14^{\circ}\text{C}/\text{watt}$. If the condenser length is doubled (by using two manifolds), the overall heat pipe resistance is reduced to $0.18^{\circ}\text{C}/\text{watt}$, the heat pipe/manifold saddle interface resistance becomes $0.12^{\circ}\text{C}/\text{watt}$, and the manifold saddle/transport fluid resistance is reduced to $0.08^{\circ}\text{C}/\text{watt}$.

It should be pointed out that the magnitude of these resistances are not easily related to equivalent resistances in full-scale heat pipe absorbers. The reason for this is that resistances in a heat pipe absorber do not scale with absorber area. The overall resistances depend on evaporator and condenser area, heat pipe spacing, and the length-to-width ratio of the absorber. But the measured resistances were correlated with the analytical model of Task II, and good agreement with the assumptions of that model was obtained. It was found that the thermal performance of such a heat pipe absorber would be acceptable.

The series of tests conducted on the individual heat pipe test sample resulted in the accomplishment of several objectives. First, it has been demonstrated that heat pipes constructed using this particular Roll-Bond channel geometry are capable of meeting the transport requirements associated with an absorbed solar flux exceeding $1000 \text{ watts}/\text{m}^2$ for heat pipe lengths up to at least 1.22 meters. Furthermore, this capability is realized without employing a wick (provided that the heat pipe is favorably oriented). Heat pipe orientations tested included a vertical reflux position and an inclination of 25 degrees with respect to the horizontal (condenser above evaporator). Tests were conducted at vapor temperatures ranging from 49°C to 71°C

without observing a noticeable effect on heat pipe performance. In addition to establishing the performance capabilities of the heat pipe, the tests also resulted in the determination of the thermal resistances at various junctions in a typical mechanical interface between the heat pipe and the fluid collection loop.

An attractive feature of a Roll-Bond heat pipe panel is that all vapor spaces are interconnected through a common manifold. This common manifold can reduce heat pipe panel processing costs because only a single evacuation and filling port is required for the entire panel. During normal heat pipe panel operation, this manifold will serve as a common liquid reservoir for each heat pipe leg. Each heat pipe of a properly charged panel will have a small liquid slug at the bottom of its evaporator to prevent vapor communication between the branches. In periods of nonuniform illumination, significant temperature gradients may exist across the panel which are accompanied by vapor pressure gradients. If temperature gradients exist between heat pipe legs which are connected to a common manifold, pressure stabilization across the panel will occur only if the free surface level of the working fluid in the shaded heat pipes is increased above that of the illuminated heat pipes. This tendency could conceivably result in reduced fluid inventory and excessive temperature gradients in the evaporator sections of the illuminated heat pipes.

To determine the effect of nonuniform heat input on a multi-chambered common manifold heat pipe panel, a second test sample was made from the sample Roll-Bond panel. This sample (shown in Figure III-4) consisted of three Roll-Bond channels, each sealed at one end and connected to the Roll-Bond manifold at the other end. To each heat pipe channel and its absorber fin, a heater was attached along the evaporator length. The condenser end of the panel was submerged in a water bath. The panel was charged with 24 ml of Freon-11. The panel was oriented at an inclination of 25 degrees relative to the horizontal (condenser above evaporator). A preliminary test was conducted with all heaters powered equally. The overall heat pipe resistance demonstrated during this test compared very well with the overall heat pipe resistance obtained from the water bath condenser tests conducted on the first test sample. These results are shown in Table III-2. The sample was then tested with one of the heaters shut off. The shutoff heater was located on one of the outer branches of the three-

legged fork. Figure III-8 shows the overall temperature gradient of each heat pipe leg as a function of the power into each of the other two legs. For comparison, the average temperature gradient across the heat pipes when all legs were powered is also shown. Although the heat pipe farthest from the unpowered leg does exhibit a larger temperature gradient than the middle heat pipe, its gradient is approximately the same as that exhibited by a uniformly illuminated panel. In addition, the temperature gradient along both powered heat pipes is a linear function of the heat transport, even at gross power inputs of 80 watts. The difference in the behavior of the powered heat pipes is apparently due to the difference in net heat transport (of the heat pipes) resulting from conduction through the heat pipe fin. Subsequent tests indicate that this performance is characteristic of vapor temperatures in the range from 49°C to 71°C. Similar tests at decreased charge levels (17 ml of Freon-11) indicated that this performance is independent of fluid charge (at least over the range of charge levels tested).

B. Task VI. Full-Scale Module Fabrication and Testing

During this task, three solar collectors were fabricated and tested. Two of the collectors had heat pipe absorber panels, and the third one was a reference collector.

1. Design and Fabrication

The absorbers of all three collectors were fabricated from a standard Roll-Bond panel shown schematically in Figure III-3. After fabrication, the panels were treated identically; that is, the same absorbent coating (3M Black Velvet) was applied and the panels were placed in similar insulated frames and covered with two glass plates. Common details of the collector designs are shown in Figure III-9. Simultaneous testing of the heat pipe solar collectors and the reference solar collector permitted direct performance comparisons of the absorber panel design.

The reference absorber panel was fabricated with no significant changes to the standard Roll-Bond panel. In this panel, the transport fluid is circulated directly through the Roll-Bond channels.

FIGURE III-8
EFFECT OF NONUNIFORM HEATING IN
MULTI-LEGGED HEAT PIPE

FIGURE III-9
COMMON DETAILS OF EXPERIMENTAL COLLECTORS

In the first of the heat pipe absorber panels, the standard Roll-Bond panel was altered to create the multi-legged vapor chamber shown in Figure III-10. To accomplish this, one of the common manifolds of the standard panel was removed and the fluid channels were welded closed. A fill tube and valve were attached to the remaining common manifold to serve as an evacuation and filling port. The panel was then purged, outgassed, and charged with Freon-11. A manifold saddle was fabricated and attached to the underside of the heat pipe panel using Eccobond. This saddle contacted each of the heat pipe legs over a 10.2 cm length at its end. Two transport fluid manifolds were clamped to the manifold saddle using a mating saddle. A thermal grease coating was applied to the transport fluid manifold/saddle interface.

After fabrication of the reference absorber panel and the first of the heat pipe panels was completed, each panel and its collector box was instrumented with chromel-alumel thermocouples. Nine thermocouples were placed on the underside of each absorber panel. Another thermocouple was used to measure the inlet collection fluid temperature for each collector. Two thermocouples were placed on the manifold saddle of the heat pipe absorber. Several more thermocouples were placed on the edges and underside of each collector box. A thermopile was used to indicate the transport fluid temperature difference between inlet and outlet of each collector. In addition to the thermocouple instrumentation, the solar flux incident on the collector plane was measured with a Sol-A-Meter MK-14 pyranometer. A water-glycol mixture was pumped through both collectors, and fluid mass flow rate through each collector was continuously measured. Ambient temperature was indicated by a thermometer. The test setup on the roof of Dynatherm's facility at Cockeysville, Maryland, is shown in Figure III-11.

The second heat pipe absorber was also fabricated from the standard Roll-Bond panel design. The major objective of this second design was to reduce the overall thermal resistance of a heat pipe absorber. Figure III-12 schematically represents the resulting panel. Both common manifolds were removed from the standard Roll-Bond panel. The remaining channel pattern was sectioned to yield individual flow channels, each centered on an absorbing fin. The channels were

FIGURE III-10
FIRST ROLL BOND HEAT PIPE ABSORBER

FIGURE III-11
FLAT PLATE SOLAR COLLECTOR TEST INSTALLATION

FIGURE III-12
SECOND ROLL BOND HEAT PIPE ABSORBER

welded closed on one end and then welded to a specially prepared heat pipe condenser block. This block contained six separate heat pipe vapor chambers similar to those shown in the baseline design configuration (Figure III-1). The block also contained an integral transport fluid manifold. This condenser block configuration offered increased condenser areas and hence reduced heat pipe condenser resistance, eliminated the resistance associated with a mechanical manifold attachment, and promised to reduce the fluid film resistance by increasing the manifold surface area. One other important modification to the second heat pipe absorber panel eliminated contact between the absorber fin and the heat pipe condenser block. After the absorber assembly was completed, the individual heat pipes were purged, outgassed, and charged with Freon-11. The same absorbent coating used previously was applied to the panel. Fifteen thermocouples were attached to the panel, and two thermocouples were placed on the transport fluid manifold. The panel was then placed in the insulated collector box (which previously housed the first heat pipe absorber panel) and was then covered with two glass plates.

2. Steady-State Testing

Simultaneous testing of a heat pipe collector and the reference collector (side-by-side) enabled relative as well as absolute thermal performance evaluations. Typical test data obtained with the first heat pipe collector is shown in Figure III-13. The instantaneous energy collection rate for each collector is shown as a function of time during one day's testing. The instantaneous insolation is also shown in this figure. By integrating the energy collection of each collector and the insolation over the duration of the test, the average daily efficiency of each collector was obtained. The average efficiency of the reference collector during this test was 57%, while the average efficiency of the heat pipe collector was 52%. This performance difference can be attributed to the overall thermal resistance of each absorber panel. Referring to Figure III-7, the major components of the resistance network for the heat pipe absorber panel are the fin resistance, the resistance at the heat pipe evaporator and condenser, the resistance at the heat pipe condenser/manifold saddle interface, and the fluid film resistance. For the reference solar collector, the overall

FIGURE III-13

INSTANTANEOUS ENERGY COLLECTION OF
REFERENCE AND FIRST HEAT PIPE SOLAR COLLECTORS

resistance consists of the fin resistance and the resistance of the transport fluid flowing through the Roll-Bond panel. Assuming the instantaneous energy collection rate of the heat pipe collector results from equal contributions of individual heat pipes, the measured overall resistance per heat pipe is $0.40^{\circ}\text{C}/\text{watt}$. This resistance is in good agreement with the component test results for a condenser length of 10.2 cm (Table III-1). A similar treatment of the instantaneous energy collection rate of the reference collector indicates that the overall resistance per Roll-Bond channel is $0.07^{\circ}\text{C}/\text{watt}$. Both resistances are based on the recommended transport fluid mass flow rate of $73 \text{ Kg}/\text{m}^2\text{-hr}$ (Ref. 6). Generally, comparative testing of the reference collector and the first heat pipe collector shows that the nearly six-fold difference in the overall thermal resistance between the two absorber panels results in approximately a 5% reduction in instantaneous collection efficiency.

The second heat pipe solar collector and the reference solar collector were also simultaneously tested to determine the performance characteristics of both collectors under identical conditions. Figure III-14 shows the instantaneous energy collection rate for each collector and the solar insolation during February 19, 1976 testing. The average daily efficiency of the heat pipe solar collector was 65%, and the average daily efficiency of the reference collector was 66%. Overall resistance measurements on the reference collector were in good agreement with the previous test results ($0.07^{\circ}\text{C}/\text{watt}$ per Roll-Bond channel), while the overall resistance per heat pipe was measured to be $0.15^{\circ}\text{C}/\text{watt}$. It should be noted that the measured efficiency of the reference collector was higher during this test than it was during January 6, 1976 testing (Figure III-10). This can be attributed to the difference in ambient temperature on the two test days. The performance of the reference collector and the second heat pipe solar collector are defined in Figures III-15 and III-16, respectively, as a function of the temperature difference between the transport fluid and ambient divided by the incident solar insolation. Comparison of these two figures indicates that the resistance difference which exists between the two panels negligibly influences relative performance.

The test results summarized thus far indicate that under steady-state test conditions (that is, when the solar insolation, ambient temperature, and transport

FIGURE III-14

INSTANTANEOUS ENERGY COLLECTION OF
REFERENCE AND SECOND HEAT PIPE SOLAR COLLECTORS

FIGURE III-15
EFFICIENCY OF REFERENCE SOLAR COLLECTOR

FIGURE III-16
 EFFICIENCY OF SECOND HEAT PIPE SOLAR COLLECTOR

fluid temperature experience no rapid fluctuations) a heat pipe solar collector can be designed and fabricated with performance characteristics similar to a conventional (reference) solar collector.

3. Transient Testing

All solar collectors are frequently exposed to transient fluctuations of the solar insolation. To determine the relative transient performance of a heat pipe solar collector and a conventional collector, another series of tests were conducted simultaneously on the reference collector and the second heat pipe solar collector.

The objective of the first of these tests was to determine the performance of each collector when subjected to a step function change in the solar input. These tests were conducted on days when the solar input was reasonably constant. The collectors were allowed to achieve a steady-state condition under full illumination. Both collectors were then totally shielded from the sun for approximately a ten minute period. Circulation of the water-glycol transport fluid was unchanged. Then the collectors were uncovered and allowed to readjust to the incident insolation. Typical test data is shown in Figure III-17. At the beginning of this test, the fluxes absorbed by each collector were equivalent and equal to $385 \text{ watts/meter}^2$. At time = 0 minutes, both collectors were covered. The temperature of each absorber panel decreased rapidly for approximately two minutes. During this time period, the sensible heat surrendered by each panel is evidenced by the absorbed flux. From time = 2 minutes until the collectors were uncovered, the performance of the panels differ markedly. The temperature of the reference panel quickly stabilized approximately 1.5°C lower than the inlet temperature of the transport fluid. At this temperature, the heat loss from the collector was approximately $250 \text{ watts/meter}^2$. During the same time period, the temperature of the heat pipe absorber panel continued to decrease. The thermal diode nature of the heat pipe panel automatically decoupled the panel from the transport fluid. Consequently, the heat losses from the heat pipe collector (strictly a function of the conduction path between the absorber panel and the transport fluid manifold) were approximately 70 watts/meter^2 . At time = 9.8

FIGURE III-17
PERFORMANCE OF REFERENCE AND SECOND HEAT PIPE SOLAR COLLECTORS
DURING INTERMITTENT INSOLATION

minutes, both collectors were uncovered. The temperature of the reference panel increased for approximately two minutes and then stabilized. The absorbed flux reflects this temperature history; that is, during the first two minutes, the absorbed flux gradually increased after which time it became reasonably stable.

The readjustment of the heat pipe solar collector to the solar flux is more complicated. For approximately 1.5 minutes, the heat pipe absorber panel temperature increased but did not exceed the inlet temperature of the transport fluid. During this time, the corresponding absorbed heat flux remained unchanged from the steady-state value established during the covered period. Following this 1.5 minute period, the absorber panel temperature continued to increase, overshooting its final steady-state value. At the same time, the absorbed flux increased, although at a rate noticeably lower than that associated with the reference collector. Nonetheless, after approximately eight minutes of exposure to illumination, the absorbed flux of the heat pipe solar collector nearly equaled the absorbed flux of the reference collector.

A quantitative comparison of the transient performances of the second heat pipe solar collector and the reference collector can be obtained by integrating the instantaneous absorbed fluxes for each solar collector over the duration of the test. The energy collected by the reference solar collector during this 26-minute test was 46 watt-hours/meter², and the energy collected by the heat pipe solar collector was 56 watt-hours/meter². The conduction path from the transport fluid manifold to the heat pipe absorber panel has an important effect on the energy collected by the heat pipe collector. For example, if the heat pipes acted like ideal thermal diodes (reverse mode conduction equal to zero), the heat pipe collector absorbed energy would have been 66 watt-hours/meter². Ideal thermal diode behavior can be approached by minimizing the transport fluid manifold/absorber panel conduction path and by increasing absorber panel size (which favorably affects the energy collection capability of the collector without increasing its heat loss potential).

The behavior of the heat pipe solar collector, which is suddenly exposed to solar insolation, warrants special attention. The nature of the energy source requires that the heat pipe collector starts up at least once daily at sunrise and at

numerous other times on partly cloudy days. Figure III-19 characterizes the transient startup behavior of the second heat pipe solar collector. On the morning of February 19, 1976, the collector was shielded from the solar illumination until 08:25. During this period, the transport fluid was circulated through the manifold; and, since its temperature exceeded the absorber temperature, the heat pipes were shut off as a result of their diode capability. After the collector was uncovered, the temperature of the entire absorber panel increased rapidly for approximately 20 minutes. After this, the temperature at the center of the panel decreased slightly and began to track the inlet temperature of the transport fluid. The noticeable peak in the temperature history of the center of the panel results from an initial lack of working fluid at this location. During startup, evaporation first occurs only at the bottom of the heat pipe panel because all of the heat pipe fluid is located in a puddle at the bottom of the heat pipes. All other illuminated portions of the panel are supplied with the heat pipe fluid through gravity-induced condensate flow. The first evidence that this condensate flow has reached the center of the panel occurs at 08:55. The temperature near the bottom of the heat pipe absorber panel continued to increase until 10:35. The temperature shown in Figure III-19 actually corresponds to a location 15 cm from the bottom of the panel. The behavior at this location could be influenced in the same manner as the center of the heat pipe panel. It is also possible that a liquid puddle existed at the bottom of the panel and that the rapid decrease in the temperature of this location corresponds to the onset of nucleate boiling.

Even more important than the explanation of transient heat pipe behavior during startup is a description of its effects on the overall thermal performance of the heat pipe collector. Extensive testing indicates that the flux collected by the panel is directly related to the temperature at the center of the panel. The collected flux of the heat pipe panel during transient startup often illustrates step-like increases associated with individual heat pipe startup (sudden drop of the temperatures at the center). On the other hand, the performance of the heat pipe collector is not noticeably influenced by the temperature at the bottom of the absorber panel. This fact supports the suspicion that the temperature history at this location is due to the existence of a puddle rather than an initial lack of working fluid.

FIGURE III-19
HEAT PIPE BEHAVIOR DURING START-UP TRANSIENT

A final test was conducted concurrently on the reference collector and on the second heat pipe collector to compare their relative performance during natural transients such as occur on a partly cloudy day. As a result of intermittent cloud cover, the solar insolation (measured in the collector plane) varied from 120 watts/meter² to 1080 watts/meter². Throughout the test, the ambient temperature was 13.5°C and the transport fluid temperature was 66°C. Figure III-20 shows the recorded total insolation during the particular day. Figure III-21 gives the flux collected by each collector as a function of time. Generally, during maximum flux conditions (12:30 to 12:50), the performances of the collectors were nearly identical. This observation supports the conclusion of the steady-state test results. During periods of low insolation, both collectors began to lose heat. The losses from the reference collector exceeded those of the heat pipe collector. Integration of the absorbed fluxes from 12:24 to 14:42 indicates that the average collected flux of the heat pipe collector is twice as large as the average collected flux of the reference collector. These results emphasize that, under certain ambient conditions, the diode characteristic of a heat pipe solar collector can provide significant performance enhancements over an otherwise identical collector.

Throughout the above test, the fluid circulation in both panels was unchanged. In an actual solar installation, a sensor coupled with a control of the pump would be used to stop circulation whenever the collector temperature drops below the fluid temperature. This is certainly adequate for long-term fluctuations. Rapid fluctuations cannot be handled as easily using an electromechanical control system. Furthermore, the diode characteristics of the heat pipes provide automatic control, thus an auxiliary system is no longer needed.

C. Task VII. Stationary Concentrator Studies

A solar collector using heat pipes as absorber elements has the unique capability of self-control. By virtue of the one-way or diode heat transfer capability of gravity pumped heat pipes, heat is transported only from the collector surface to the transport manifold but never in the opposite direction (Ref. 7). This diode action prevents heat losses from unilluminated collector elements and provides automatic

Date: 4/1/76

FIGURE III-20
RECORD OF INSOLATION DURING TRANSIENT TEST

Date: 4/1/76

FIGURE III-21

PERFORMANCE OF REFERENCE AND SECOND HEAT PIPE
SOLAR COLLECTORS ON A PARTLY CLOUDY DAY

shutoff of a collector whose temperature is less than that of the collection loop.

An analytical investigation was conducted to determine the feasibility of utilizing the diode action of the heat pipes in a stationary concentrator. The objective of the concentrator is to increase collector efficiency in order to collect at higher temperatures. The principle of this stationary concentrator is very simple. A series of cylindrical Fresnel lenses provides moderate concentration in the absorber plane. Diode heat pipes collect energy only from those absorber sections which are in focus at a particular time. Heat pipes which are out of focus are automatically turned off and do not contribute to heat losses. As the focal line moves (due to daily or seasonal progression of the sun vector) different heat pipes move in and out of focus. Figure III-22 shows an exploded perspective view of a single concentrator element. A schematic cross-section of an array of elements is shown in Figure III-23. Each heat pipe with its extended surface (fin) represents an individual collector element. It will transport heat to the collection manifold if its temperature exceeds that of the manifold. Nonilluminated elements (or those whose temperature is less than the manifold temperature) are inoperative. The concentrator can be operated in either of the following modes:

- Generate higher quality energy (higher temperatures than achievable with conventional flat-plate collectors).
- Achieve higher efficiency at standard output temperatures ($<100^{\circ}\text{C}$). The effective instantaneous absorber area consists only of the active elements; hence the area subject to heat losses is substantially reduced.

A stationary concentrator of similar design was recently reported in a NASA TECH BRIEF (Ref. 8). But the reported system does not utilize heat pipes for automatic control of individual absorber elements. Instead, it uses thermostatically controlled valves to release fluid from individual channels. An ideal concentrator would have a high concentration ratio (e.g., $>10:1$). But it is difficult, if not impossible, to design a stationary concentrator in which the focus is always of high quality. Daily and seasonal changes of the direction of the sun vector are too great (solid angle of nearly 2π) to achieve a sharp focus at all times. Thus, the approach taken in the

FIGURE III-22
SCHEMATIC OF STATIONARY CONCENTRATOR ELEMENT

FIGURE III-23
SCHEMATIC CROSS-SECTION OF STATIONARY CONCENTRATOR

analytical evaluation represents a compromise.

- For reasons of economy a flat absorber surface was used. Other absorber configurations may yield better concentrations over wider angles, but the added complexity was judged to be not economical.
- The absorber plane was moved out of the focal plane of the Fresnel lens. This results in some sacrifice in the concentration at normal incidence, but reasonable concentration is gained at off-normal angles of solar incidence. This is demonstrated in Figure III-24. It shows a typical calculated ray pattern for two angles of incidence (0° and 30°). While for 0° the best concentration is obtained when the absorber is located 25 cm from the plane of the lens, the optimum location of the plane shifts to approximately 19 cm for 30° incidence angle. Obviously, the location for optimum concentration for any angle between 0 and 30° lies between the two extremes.

A reasonably detailed analysis was made to assess the performance of a stationary heat pipe solar concentrator. The following assumptions were made in the analysis.

- The Fresnel lens is infinitely thin, has an equivalent index of refraction of 1.5 and an absorptance of 0.9 (Ref. 9).
- Each heat pipe and its associated fin is independent of the other heat pipes. Heat losses from the collector assembly are proportional to the number of operating elements.
- The axis of the collector is oriented east-west, and the collector is elevated above the horizontal by an angle equal to the local latitude. (East-west orientation and the indicated elevation are not essential to the performance. They simply were convenient choices for the analysis.)
- The heat pipe concentrator and a reference conventional collector are compared for the following parameters:

Absorptance-Transmittance Product : $\alpha\tau = 0.85$

Heat Loss Coefficient : $\mu_1 = 4 \times 10^{-4} \text{ watt/cm}^2\text{-}^\circ\text{C}$

Solar Flux : $q_s = 0.063 \text{ watt/cm}^2$
(200 Btu/hr-ft²)

(Note that for the concentrator an additional absorptance of 0.9 was used to account for the Fresnel lens.)

FIGURE III-24
CONCENTRATION OF REPRESENTATIVE RAYS AT
TWO DIFFERENT ANGLES OF INCIDENCE

- The Fresnel lens has a focal length of 25 cm and each lens element is 12.5 cm wide. The absorber consists of five heat pipe elements (each 2.5 cm wide) for each lens element.
- As explained above, the integrated energy collected over one day depends on the location of the absorber plane with reference to the focal plane of the lens. The analysis was performed for two locations:

15.6 cm (63% of focal length)

18.75 cm (75% of focal length)

Using the above assumptions, the performance of a stationary heat pipe solar concentrator and a reference conventional collector (with the same parameters) was calculated. Figures III-25 through III-28 show the resultant collected energy (normalized with respect to $\alpha \tau q_s$) for two solar declinations and two temperature differences between collection manifold and ambient. The absorber plane is located (in all cases) at 15.6 cm from the lens plane.

Figure III-25 $\Delta T = 55^\circ\text{C}$, Declination = 0 (Equinox)

Figure III-26 $\Delta T = 55^\circ\text{C}$, Declination = $\pm 23.45^\circ$ (Solstices)

Figure III-27 $\Delta T = 80^\circ\text{C}$, Declination = 0 (Equinox)

Figure III-28 $\Delta T = 80^\circ\text{C}$, Declination = $\pm 23.45^\circ$ (Solstices)

A substantial increase of collected energy is indicated in all cases, even with the 10% additional loss attributed to the Fresnel lens. A numerical comparison between the heat pipe concentrator and the reference conventional collector is shown in Table III-2. It gives the ratios of total collected energy per day between concentrator and reference collector. In addition to the four cases shown in the above figures, four more cases for a different location of the absorber plane are listed. As expected, the greatest improvement occurs at the higher collector temperatures. At a collector-ambient temperature difference of 55°C , the year-round improvement averages 32% and at 80°C it averages 100%. Also, the effect of location of the absorber plane is interesting. If it is close to the focal plane of the lens, the greatest improvement is obtained at near normal incidence. (During the equinoxes the sun vector is, on the average, closer to the normal of the collector than during the solstices.) Deliberate defocusing, by moving the collector plane closer to the lens, shifts the improvement towards larger angles of incidence.

FIGURE III-25
COMPARISON OF CONCENTRATOR AND REFERENCE COLLECTOR
DURING EQUINOX ($T_{\text{coll}} - T_{\text{amb}} = 55^{\circ}\text{C}$)

FIGURE III-26
COMPARISON OF CONCENTRATOR AND REFERENCE COLLECTOR
DURING SOLSTICES ($T_{coll} - T_{amb} = 55^{\circ}\text{C}$)

FIGURE III-27
COMPARISON OF CONCENTRATOR AND REFERENCE COLLECTOR
DURING EQUINOX ($T_{\text{coll}} - T_{\text{amb}} = 80^{\circ}\text{C}$)

FIGURE III-28
COMPARISON OF CONCENTRATOR AND REFERENCE COLLECTOR
DURING SOLSTICES ($T_{\text{coll}} - T_{\text{amb}} = 80^{\circ}\text{C}$)

Separation of Lens and Absorber	15.6 cm		18.75 cm	
Declination	0°	23.45°	0°	23.45°
$\Delta T = 55^\circ\text{C}$	1.27	1.38	1.39	1.24
$\Delta T = 80^\circ\text{C}$	1.84	2.31	2.13	1.80

TABLE III-2
TOTAL DAILY ENERGY OF CONCENTRATOR DIVIDED BY
TOTAL DAILY ENERGY OF REFERENCE COLLECTOR

Details of the calculations used to obtain the above results cannot be presented here. The basic approach is summarized below. At first, a simple numerical ray tracing technique was used to calculate the optical concentration within a selected absorber plane. For this calculation, the lens was assumed to be infinitely thin and was divided into 20 segments. The absorber plane was divided into five segments for each lens. This corresponds to a heat pipe (plus fin) width of 2.5 cm which appears to be a reasonably fine division. The total collected energy was calculated by performing a heat balance for each of the absorber elements as shown below for the i th element:

$$q_i = 0.9 \alpha \tau \cos \theta \frac{n_i}{20} - \frac{\eta_1 \Delta T}{5} \quad (1)$$

where: q_i is the flux collected by the i th element

$\cos \theta$ is the angle between the sun vector and the normal of the collector

n_i is the number of rays (out of 20) impinging on the i th element

$\alpha \tau$, η_1 , ΔT have the usual meaning (Ref. 4)

A summation over all five absorber elements was then performed, whereby only positive values of q_i were included, that is:

$$q_n = \sum_{i=1}^5 q_i \quad (2)$$

with:

$$q_i = q_i \text{ for } q_i \geq 0$$

$$q_i = 0 \text{ for } q_i < 0$$

The same type of analysis was applied to the conventional reference collector, except that uniform illumination of all absorber elements was used; that is:

$$n_i = 4 \text{ for all } i \text{ values}$$

Also, the reference collector was not penalized by the added absorption of 0.9 due to the Fresnel lens. It should be pointed out that the lens could replace at least one conventional cover; the assumed additional absorption of 0.9 may thus be pessimistic.

With regard to the economics of the investigated system, the Fresnel lens and the heat pipes represent added cost elements. However, this increased cost will most likely be cancelled by the increased efficiency which, in turn, results in smaller collectors for the same requirement. In addition, the stationary heat pipe concentrator offers sufficient design flexibility to tailor the collector for specific needs; e.g.:

- High temperatures and efficiency at preferred times of day or season, or
- Spreading out moderate collection rates over long periods

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