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ANNUAL PROGRESS REPORT

for

MASTER

HEAT PIPES APPLIED TO FLAT-PLATE
SOLAR COLLECTORS

Walter B. Bienert
David A. Wolf

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Larry C. Williams
Authorizing Official

Date: 09/07/2007

Dynatherm Corporation
One Industry Lane
Cockeysville, Maryland

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I. INTRODUCTION

The objective of this program is to investigate the use of heat pipes in flat-plate solar collectors. Heat pipes are passive heat transport devices which have a very high equivalent thermal conductance. Because of their high conductance, heat pipes appear to be well suited to transport heat from the collecting surface to an air or liquid distribution system. The program consists of: (I) Configuration Studies, (II) Parametric Performance Studies, (III) Economic Analysis, (IV) System Integration Studies, (V) Submodule Fabrication and Testing in the laboratory, (VI) Full-Scale Module Fabrication and Testing using solar input.

In a flat-plate heat pipe solar collector, the heat pipes are either attached to the absorber surface or are made an integral part of it. The collected energy is transported by the heat pipes to the edge of the collector where it is transferred to a liquid or gas distribution system. The working fluid in the heat pipes recirculates primarily through the influence of gravity. As a result, the heat pipes are unidirectional conductors; that is, they behave as thermal diodes. Heat is transported only from the absorber surface to the collection manifold but never in the opposite direction. Also, by virtue of being sealed systems, corrosion can be minimized and a nonfreezing working fluid can be selected. Since heat exchange with the distribution system occurs only along one edge of the collector, a mechanical attachment to the manifold is feasible and this results in simplified field installation and maintenance.

II. SUMMARY

The program is subdivided into the following six tasks:

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

During the first five months (the period covered by this report), the research was concentrated on the first four tasks. Most of the feasibility studies and analytical evaluations have been completed. Highlights of the results to date 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" was selected for further evaluation because it offers the best thermal performance and the most economical method of manufacture. The pattern of the vapor paths in a Roll-Bond Panel can be readily chosen so as to minimize interface resistances between absorber and 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'). 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 $1000 \text{ Btu/Hr-Ft}^2\text{-}^\circ\text{F}$ 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. However, it would be premature to conclude that heat pipe collectors are only suitable for use in air heating systems. Advantages such as those that accrue with sealed systems, ease of their mechanical integration, and their freedom from freezing should override the slight penalty in collection efficiency and make them also attractive for use in liquid systems.

Under this task, the effects of unidirectional heat transfer (diode action) of the

heat pipe was also evaluated. The diode property can be utilized to slightly increase the total daily collection rate ($\sim 10\%$) and, more importantly, improve the collection efficiency during periods of low insolation (early morning and late afternoon).

Task III. Economic Analysis

This task is not yet completed. Preliminary cost evaluation indicates that heat pipe processing has the greatest impact on cost effectiveness of a heat pipe collector. For this reason, Roll-Bond Panels have been selected as a baseline configuration for a heat pipe absorber plate. This fabrication technique allows interconnection of the heat pipe vapor spaces and reduces fabrication cost over that of a design consisting of several individual heat pipes within one panel.

Task IV. System Integration Studies

In cooperation with our 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 transport 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.

III. TECHNICAL SECTIONS

A. Task I. Configuration Studies

Under this task, a large number of flat-plate heat pipe collectors were evaluated qualitatively. Various techniques for incorporating heat pipes into the absorber plate were considered. The basic design approaches are shown in Figure III-1 and are:

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

Individual heat pipes are the most straight forward design approach. Typically, 1/2-inch O.D. pipes are bonded, clamped, or brazed to the absorber panel. Heat is collected along the major portion of the heat pipe's length and delivered from a short condenser section (typically 2 inches to 20 inches) to a transport system. This approach is technically feasible but requires the processing of a fairly large number of pipes (6 to 10) for a typical solar panel of 4-foot by 8-foot size. Heat pipes incorporated into extrusions have already been used in solar collectors (Ref. 1). This geometry eliminates thermal contact resistances between the heat pipe and the absorber. However, the heat flux at the absorber surface is relatively low and contact resistances at the evaporator have, therefore, only negligible influence on overall performance. Thus, the improvement in contact resistance does not seem to warrant the use of extrusions. In addition, this approach also requires the processing of many individual heat pipes.

A configuration consisting of heat pipes built directly into a 2-piece bonded absorber plate (for example, a Roll-Bond Panel which is the trade name for a product manufactured by Olin Brass Corporation) is very attractive. Thermal interface resistances are eliminated; close spacings of the heat pipes become feasible; and, most importantly, the heat pipe processing cost is reduced. A typical Roll-Bond Panel configuration, as it is being used in solar collectors, is shown in Figure III-2a. A configuration suitable for a multi-pronged heat pipe is depicted schematically in Figure III-2b. All vapor spaces are interconnected through a manifold located at the bottom which

Either Side Could Be Absorber Surface

- a. Individual Heat Pipes Bonded to Absorber Plate in Preformed Grooves

Either Side Could Be Absorber Surface

- b. Individual Heat Pipes Extruded with Integral Absorber Fins

Both or Single Side Expanded

- c. Absorber Plate with Integral Heat Pipes Formed in 2-Piece Bonded Plate

FIGURE III-1

INCORPORATION OF HEAT PIPES INTO ABSORBER PLATE

a. Roll-Bond Panel with Liquid Passages

b. Roll-Bond Panel with Heat Pipe Passages

FIGURE III-2
ROLL-BOND PANEL CONFIGURATIONS

also serves as the common fluid reservoir for all branches. During heat pipe processing, one common evacuation and filling port is used. A preliminary evaluation of the heat transport capability of such a configuration has shown (see Task II) that sufficient transport should be achievable even without employing a wick. This conclusion remains to be verified through testing in subsequent phases of the program. In any case, the processes involved in fabricating Roll-Bond Panels do not preclude the incorporation of simple wicks. Roll-Bond Panels and individual heat pipes bonded to a standard absorber panel have been selected for further evaluation in the program.

The design of the thermal interface between heat pipes and the transport system presents the greatest challenge for a heat pipe solar collector. Since the width of this interfacing section should be confined to a few inches (versus several feet for the length of a typical collector), the heat flux at the interface is roughly one order of magnitude higher than the solar flux. This increased flux is compatible with the heat transfer characteristics of a heat pipe. The problem lies in efficiently transferring this flux into a liquid or gas transport system.

The general configurations which were considered are shown schematically in Figure III-3. The collection manifold could be either integral or detachable. An integral manifold is defined here as one which is a permanent part of the collector. Interconnections between collectors are made through hoses, tubing, or ducting as in conventional collectors. Detached manifolds are not part of the collectors. They are preinstalled into the supporting structure (e.g., roof of the building). The collectors themselves consist of prefabricated and sealed modules which are mechanically attached to the manifolds during installation. Typical mechanical configurations of liquid and air manifolds are shown in Figures III-4, III-5, and III-6.

Although the heat pipe condenser does not usually represent the dominating thermal interface resistance, nevertheless, an optimization of the available interface area is advantageous. Some condenser geometries, which were considered during the parametric studies, are shown in Figure III-7. A summary of the predicted performance of the various heat pipe collector configurations is presented under Task II.

A baseline design which uses a Roll-Bond Heat Pipe Panel is shown in Figure III-8. The selected overall collector size is 24 inches by 96 inches. Two tempered

FIGURE III-3
TYPICAL CONFIGURATIONS OF THERMAL INTERFACES BETWEEN
HEAT PIPE ABSORBER AND HEAT TRANSPORT SYSTEM

FIGURE III-4
MECHANICAL ATTACHMENT OF HEAT PIPE
ABSORBER TO LIQUID MANIFOLD

FIGURE III-5
HEAT PIPE COLLECTOR WITH INTEGRAL AIR DUCT SECTION

FIGURE III-6
HEAT PIPE COLLECTOR ATTACHMENT TO
PREINSTALLED AIR DUCTING

a. Condenser Vapor Chamber Formed in 2-Piece Bonded Plate

b. "L" Shaped Individual Heat Pipes Bonded to Absorber

c. Straight Individual Heat Pipes Bonded to Absorber

FIGURE III-7
TYPICAL HEAT PIPE CONDENSER GEOMETRIES

FIGURE III-8
BASELINE DESIGN OF HEAT PIPE FLAT-PLATE COLLECTOR

cover-glasses are spaced $3/8$ inch above the absorber and have a $3/8$ inch spacing between them. A sheet-metal edge structure has been selected which permits replacement of cover-glasses after installation. The absorber is backed by 5 inches of glass wool insulation. At the condenser end, the heat pipe passages expand into rectangular vapor chambers. An extruded saddle is bonded to the backside of the absorber and is located directly behind the heat pipe vapor chambers. The saddle, which has a semi-circular groove, is mechanically clamped to a $3/4$ -inch copper pipe which serves as a water manifold.

The above baseline design represents only one typical configuration of a heat pipe collector. Considerable optimization, breadboard testing, and design effort remain before a final configuration can be selected. Also, the decision as to whether liquid or air heating applications will be given first priority will depend on the results of the breadboard tests.

An important potential advantage of a heat pipe collector is the unidirectional heat transport capability of these particular heat pipes. Unlike regular heat pipes, in which the condensate return is accomplished with a capillary structure, the heat pipes in a solar collector rely on gravity for liquid return. Therefore, heat transport occurs only in an upward direction which requires that the collection manifold be located near the top of the collector. Such heat pipes are frequently referred to as "thermal diodes".

One consequence of this diode action is that heat is transported only if the absorber is at a slightly higher temperature than the collection manifold. Most conventional solar collector installations use temperature sensors which activate the circulation pumps or blowers whenever the collector temperature is above that of the rest of the system. Conversely, circulation is stopped when the collector is cooler than the fluid in the manifold. Failure to stop circulation under this condition would cause a reverse flow of heat; that is, heat from the absorber would be rejected rather than collected. The thermal diode heat pipes automatically provide this same control; that is, they turn off whenever the adiabatic temperature of a panel drops below the fluid loop temperature. The important feature of this type of control is that it works on individual panels. Whenever a panel drops below the useful collection temperature, it is automatically removed from the system. Differences in collection rate between

panels can occur for several reasons, such as:

- Selective shading
- Breakage of individual panels
- Differences in heat losses
- Differences in orientation

A conventional control system cannot easily adapt to these factors since it is economically impractical to monitor and control the flow through individual panels.

Special collector designs can be devised in which the diode feature increases the collector efficiency during periods of low insolation. Two examples of such designs are shown in Figures III-9 and III-10. The first design (Figure III-9) utilizes small reflectors which are located at right angles to the collector surfaces. The absorber consists of a multitude of parallel individual heat pipes which are thermally decoupled from each other. During normal incidence, the reflectors do not interfere with the sun's rays and the collector efficiency is identical to that of a conventional collector. During oblique incidence (other than noon), part of the absorber surface is shaded by the reflectors and part receives additional illumination via the reflectors. The illuminated segments of the surface contribute to the heat collection in accordance with the local incident flux. Due to the diode action of the heat pipes, the shaded portions do not contribute to heat losses. This results in increased efficiency during periods of oblique incidence. The instantaneous efficiency can then be expressed as (Ref. 2):

$$\eta = \frac{1}{I} \sum_i \zeta_i (a\tau I_i - U_1 \Delta T) \quad (1)$$

where I is the instantaneous flux intercepted by the collector ($I = 300 \cos \omega t$ Btu/hr-ft²); ζ_i the fraction of the total area with a flux I_i ; $a\tau$ the product of collector absorptance and transmissivity of the covers ($a\tau = 0.85$); U_1 the heat loss coefficient ($U_1 = 0.5$ Btu/hr-ft²-°F); and ΔT the difference between local absorber and ambient temperature ($\Delta T = 100^\circ\text{F}$).

The summation in Equation 1 is extended only over those area segments for which

A. Geometry

B. Comparison of Efficiency with Identical Flat-Plate Collector

FIGURE III-9
HEAT PIPE COLLECTOR WITH PERPENDICULAR REFLECTORS

A. Geometry

B. Comparison of Efficiency with Identical Flat-Plate Collector

FIGURE III-10
CONVOLUTED HEAT PIPE COLLECTOR

the net collected heat (the expression in the parenthesis in Equation 1) is positive. Using the indicated numerical values, the efficiency versus time curve shown in Figure III-9 was generated. The ratio of reflector height-to-collector width for this particular example was 0.134, but the results are not very sensitive to this ratio. The same numerical values were applied to the conventional reference collector for which the result is also plotted in Figure III-9.

Another design approach is shown in Figure III-10. Here the collector surface is convoluted with individual segments forming an included angle of 90° . Again, the objective is to increase collection efficiency during periods of oblique incidence. At normal incidence, the efficiency is somewhat lower than that of a conventional flat-plate collector. This is due to the increased area (increased by $\sqrt{2}$) which causes higher heat losses than those of an equivalent flat-plate collector. However, during periods of oblique incidence, the illumination is concentrated on a smaller area. Because of the diode action of the heat pipes, thermal losses occur only from the illuminated sections and the efficiency of energy collection is substantially increased.

These are two examples illustrating the use of diode action of the heat pipes to spread out high efficiency energy collection over large portions of the day. It can also be shown that, for the given set of collector and insolation parameters, the total rate of daily collection is increased by approximately 10%. A more detailed evaluation must include incremental losses at the reflectors, increased reflection by the cover-glasses, and an optimization of geometry and orientation.

B. Task II. Parametric Performance Studies

The generalized performance equation for a solar collector is often written as (Ref. 3):

$$q_u = F'F'' q_a - U_1 (T_i - T_a) \quad (2)$$

This equation relates the useful heat collection rate per square foot of collector area (q_u), to the absorbed solar flux (q_a), the collector heat loss coefficient (U_1), ambient temperature (T_a), and inlet fluid temperature (T_i). This equation utilizes two performance factors -- the Collector Efficiency Factor (F') and the Flow Factor (F''). The Flow Factor (F'') can be written as:

$$F'' = \frac{1 - \exp (U_1 F' / GC_p)}{U_1 F' / GC_p} \quad (3)$$

where GC_p is the product of the collection fluid flow rate per collector unit area and the specific heat of the collection fluid.

The Collector Efficiency Factor is the ratio of the heat transfer coefficient from the heat transport fluid to ambient and the heat transfer coefficient from the absorber plate and ambient. For a heat pipe collector panel, this efficiency can be expressed as a function of the thermal resistances due to heat collection at the absorber surface, the transport of this heat to the interface between the heat pipe fluid and transport fluid, and the transport of the heat across this fluid interface. The pertinent dimensions of a typical Roll-Bond Heat Pipe Panel is shown in Figure III-11. For this panel geometry, the Collector Efficiency Factor can be written:

$$F' = \frac{1}{\frac{B}{b + 2 L' F} + \frac{U_1 B}{C h} + U_1 H B R_{\text{head}}} \quad (4)$$

where:

F = Fin efficiency of collector plate

h = Evaporator film coefficient

FIGURE III-11
CROSS-SECTION OF HEAT PIPE ABSORBER PANEL
AND CHARACTERISTIC DIMENSIONS

C = Evaporator circumference

H = Total height of panel

R_{head} = Resistance of the interface between
heat pipes and transport fluid

The resistance at the fluid interface is critical to the overall collection efficiency; hence, parametric performance studies have been made on header configuration for air and water collection.

In a hot-water solar collector, the resistance at the fluid interface is chiefly dependent on the heat pipe condenser film coefficient and condenser area, the water film coefficient and area, and the resistance between the two fluid loops. Collector Efficiency Factors for a basic collector unit, for various header configurations, are summarized in Table III-1. It is assumed that a basic unit is 4 feet wide by 8 feet high. The heat pipes are assumed to be 0.5 inch wide and located on 2-inch centers, as shown in Figure III-12. The length of the heat pipe condenser is assumed equal to the width of the saddle(s) of the water pipe. It is also assumed that these saddles are bonded to the absorber panel but are mechanically fastened to the water pipe resulting in a contact conductance of 1000 Btu/hr-ft²-°F.

The efficiency factors in Table III-1 are presented for a water film coefficient based on the recommended unit flow rate of 15 lbm/ft²-hr (Ref. 2) and for an infinite film coefficient. Actually, the efficiency for an infinite film coefficient is obtained when the product of the film coefficient and the surface area within the water tube is infinite. This condition could be approached through the use of internally-finned water tubes or series couplings of additional collectors. For example, suppose a round header tube was replaced with a number (N) of smaller tubes which had the same cross-sectional area. The area of each configuration can be written:

$$A_1 = \pi r_1^2 \text{ (Large Tube)} \quad ; \quad A_s = N (\pi r_s^2) \text{ (Small Tubes)} \quad (5)$$

Since $A_1 = A_s$, then:

$$r_s = \frac{r_1}{\sqrt{N}} \quad (6)$$

The surface area for each configuration can be written:

$$S_1 = 2 \pi r_1 \quad ; \quad S_s = N (2 \pi r_s) \quad (7)$$

The film coefficient is directly proportional to:

$$h \propto \frac{Re^{0.8}}{r} \propto \frac{r^{0.8}}{r} \propto r^{-0.2} \quad (8)$$

Hence:

$$h_1 \propto r_1^{-0.2} \quad ; \quad h_s \propto r_s^{-0.2} \quad (9)$$

The product of the film coefficient and surface area for each configuration is:

$$(h S)_1 \propto (2 \pi r_1) r_1^{-0.2} \quad (10a)$$

$$\begin{aligned} (h S)_s &\propto N (2 \pi r_s) r_s^{-0.2} = N \left(2 \pi \frac{r_1}{\sqrt{N}} \right) \left(\frac{r_1}{\sqrt{N}} \right)^{-0.2} \\ &= N^{0.6} (h S)_1 \end{aligned} \quad (10b)$$

Therefore:

$$\frac{(h S)_s}{(h S)_1} = N^{0.6} \quad (11)$$

Commercial tubes are available which provide numerous flow channels within the external envelope of the header sizes shown in Figure III-12.

Another important effect on the Collector Efficiency Factor is the interface resistance between the water pipe and its saddle. This relationship is shown in Figure III-13, where the efficiency factor is plotted as a function of the inverse bond conductance for the header configuration employing one flattened water pipe with a heat pipe condenser width of 1.75 inches (Table III-1). To demonstrate the relative importance of the components of the entire thermal resistance network, the resistances of each

HEADER CONFIGURATION	Configuration #1				Configuration #2
	1		2		1		2	
Number of Headers								
Condenser Width (inches)	0.50	1.75	0.50	1.75	0.50	1.75	0.50	1.75
Efficiency Factor 1	0.848	0.914	0.912	0.949	0.817	0.894	0.893	0.938
Efficiency Factor 2	0.786	0.842	0.849	0.881	0.743	0.806	0.816	0.853

NOTE: Efficiency Factor 1 is based on an infinite heat transfer coefficient in the water.
Efficiency Factor 2 is based on a water film coefficient in accordance with a flow rate of 15 lbm/ft²-hr.

TABLE III-1
COLLECTOR EFFICIENCY FACTORS (F') FOR HOT WATER SOLAR COLLECTOR

a. Typical Top View

b. Side View of Configuration #2

c. Side View of Configuration #1

FIGURE III-12
WATER HEADER CONFIGURATIONS

FIGURE III-13
 EFFECT OF BOND CONDUCTANCE ON COLLECTOR EFFICIENCY FACTOR

component of the above configuration are given as a percentage of the total resistance in Table III-2.

The Collector Efficiency Factor for a heated-air solar collector is also defined by Equation 4. The resistance at the fluid interface (R_{head}) is chiefly determined by the heat pipe condenser-film coefficient and area and by the air-film coefficient and effective fin area. Collector Efficiency Factors for a basic collector unit and various header configurations are given in Table III-3. The geometrical assumptions concerning the basic unit are shown in Figure III-14. It is assumed that the fins are bonded to the absorber surface. The Collector Efficiency Factors (Table III-3) are based on a recommended air-volume flow rate for each unit collector area of 2 CFM/ft² (Ref. 2). The airflow velocities used in the computations are congruent with this flow rate and with the cross-sectional flow area described by the fin geometry. The pressure drops (inches of water) associated with the basic collector unit are given in Table III-3 for each header configuration. For this basic unit, the resultant total volume flow rate is 64 CFM and the flow length is 4 feet.

Solar collector performance cannot be totally described by Collection Efficiency Factors. The Flow Factor, defined by Equation 3, must also be used to express the useful energy collection as a function of the inlet temperature of the collection fluid. For water collection loops, where flow rates tend to be high (~ 15 lbm/hr-ft²), the Flow Factor is in the range of 95%. For air collection loops, pressure drops usually prohibit high flow rates and, consequently, Flow Factors are approximately 85% for typical volume flow rates of 2 CFM/ft². For this reason, the design of heated-air solar collectors should be based on the product of the Flow Factor and the Collector Factor, whereas the design of hot-water solar collectors is more dependent on the Collector Efficiency Factor.

As an important part of the overall collector performance, heat pipe transport was also examined. An analytical model was developed to predict heat pipe transport capability as a function of fluid charge. This model assumes that heat pipe performance is limited by the hydrodynamic pressure balance on a heat pipe elevated at an angle of 25° with respect to the horizontal plane. Performance predictions were made with this model for an 0.50-inch heat pipe and a typical heat pipe in a Roll-Bond Heat Pipe Panel.

WATER FILM COEFFICIENT	∞	500 Btu/hr-ft ² -°F
Absorber Plate to Heat Pipe Evaporator	11.2%	5.6%
Heat Pipe Evaporator Resistance	3.3%	1.6%
Heat Pipe Condenser Resistance	35.7%	17.9%
Interface Resistance (Saddle-Water Pipe)	49.8%	25.0%
Water Pipe Flow Resistance	0.0%	49.9%
TOTAL	100.0%	100.0%
EFFICIENCY FACTOR	0.914	0.842

NOTE: The water film coefficient of 500 Btu/hr-ft²-°F is based on a flow rate of 15 lbs/ft²-hr.

TABLE III-2
RELATIVE CONTRIBUTIONS OF INDIVIDUAL RESISTANCES
TO EFFICIENCY FACTOR

		CONFIGURATION #1		CONFIGURATION #2		CONFIGURATION #3		CONFIGURATION #4	
		H = 4 inches P = 0.25 inch t = 0.0625 inch		H = 4 inches P = 0.167 inch t = 0.0625 inch		H = 2 inches P = 0.167 inch t = 0.0625 inch		H = 1 inch P = 0.0833 inch t = 0.01 inch	
W-Condenser Width (inches)		0.50	1.75	0.50	1.75	0.50	1.75	0.50	1.75
L-Condenser Length = 12"	F'	0.890	0.904	0.916	0.931	0.892	0.906	0.918	0.933
	ΔP (in. H ₂ O)	0.044	0.044	0.165	0.165	0.346	0.346	1.030	1.030
L-Condenser Length = 16"	F'	0.906	0.916	0.929	0.940	0.908	0.919	0.931	0.942
	ΔP (in. H ₂ O)	0.012	0.012	0.124	0.124	0.260	0.260	0.773	0.773

TABLE III-3
COLLECTOR EFFICIENCY FACTORS AND PRESSURE DROPS
FOR HOT-AIR SOLAR COLLECTOR

FIGURE III-14
TYPICAL AIR HEADER CONFIGURATION
WITH CHARACTERISTIC DIMENSIONS

Each heat pipe contained a 50% volumetric charge of Freon-11. These predictions, along with the required transport for each heat pipe, are shown as a function of heat pipe temperature in Figure III-15. In each case, the predicted performance exceeds the expected requirements. However, performance testing must be implemented to insure that heat pipe transport capability is not limited by flux limitations, entrainment, or in any other manner.

FIGURE III-15
 HEAT PIPE TRANSPORT CAPABILITY AND REQUIREMENTS

C. Task III. Economic Analysis

At this early stage of the program, a detailed economic analysis of a heat pipe solar collector is premature. Because of the sensitivity of any design to cost, an estimate of the incremental cost associated with the heat pipes was made. As a basis for the estimate, the baseline design discussed in Task I was used. It consists of a liquid heating collector using a Roll-Bond Heat Pipe Panel as the absorber. A single, unfinned water pipe serves as the collection manifold. The latter is detachable; i.e., the collector is mounted mechanically to the manifold.

The incremental costs are defined as those which can be attributed to heat pipes. That is, the basic collector components such as Roll-Bond Panel, cover-glasses, insulation, frames, etc., are not included. The incremental costs consist of additional materials; that is,

- Interfacing saddle
- Water manifold
- Heat pipe working fluid

(It is assumed that a wickless heat pipe system can be developed using the available Roll-Bond technology). Another part of the incremental costs is additional labor; for example,

- Heat pipe processing
- Fabrication of interfacing saddle
- Bonding of saddle to Roll-Bond Panel

The materials estimates were based on using aluminum extrusion at \$0.80/lb, copper tubing at \$1.50/lb, and Freon working fluid at \$1.00/lb.

The most crucial labor cost is associated with heat pipe processing. The estimate was based on using Dynatherm's actual experience with heat pipe heat exchangers which employ large numbers of individual heat pipes. Standard industry practices were used to determine labor costs for fabricating saddles and mounting hardware.

The results of this analysis of incremental costs are shown in Table III-4. A burdened labor rate of \$12.00 per hour was used. Material costs included a 32% markup for general and administrative costs (G&A). A nominal profit was included. The indicated cost represents, therefore, the manufacturer's increment associated with the heat pipes over and above the cost of basic solar collector of the same design.

MODULE SIZE		4' x 8'	2' x 8'
LABOR COST	Heat Pipe Processing	\$8.20	\$8.20
	Saddle Fabrication	6.00	3.00
	Saddle Attachment	3.00	3.00
MATERIAL COST	Interfacing Saddle	5.00	2.50
	Water Manifold	1.30	0.65
	Freon Working Fluid	2.80	1.40
TOTAL COST		\$26.30	\$18.75
Cost per ft ² of Area		\$ 0.82	\$ 1.17

NOTE: Based on reference design identified in Task I.
See text for synthesis of cost details.

TABLE III-4
INCREMENTAL COST OF HEAT PIPE SOLAR COLLECTOR

D. Task IV. Integration Studies

The integration of heat pipe collector panels with the requirements of typical HVAC systems was studied. As a baseline installation, an area which is 35 feet wide and 20 feet high was selected. This provides sufficient area to install sixteen 4 feet by 8 feet collector modules which are arranged in two rows with eight modules to the row. Heat pipe collectors permit series connection irregardless of whether an integral or a detachable manifold is used. Because the manifolds are conveniently aligned near the top of the panels, the series connection applies to collectors which are located within one horizontal row.

A series connection of collectors within one row and a parallel connection of the two rows are shown schematically in Figure III-16A. The technical feasibility of this type of interconnection was evaluated. For the case of liquid heating systems, it was found that the optimum flow rate associated with an individual collector results in a poor film coefficient in the water manifold. As observed in Task II, individual heat pipe collectors should employ multiple water manifolds or internal finning within the manifold. In a series connection of a number of collectors, the increased flow rate obviates the need for internal finning. For example, if eight collectors are connected in series, the Collector Efficiency Factor achievable with one ordinary straight water pipe approaches that obtainable with infinite heat transfer in the water manifold. The penalty of series connection is, of course, a higher pressure drop in the manifold. Standard engineering practice recommends (Ref. 4) that the frictional pressure drop in HVAC hydraulic systems will not exceed a head of 3 to 5 feet in order to make it small when compared to typical elevation differences.

The pressure drops associated with an 8-module series connection are listed in Table III-5 for the basic configurations examined under Task II. The table also lists the corresponding Collection Efficiency Factors based on one header serving as a manifold. A comparison with Table III-1 shows that series connection increases the Collector Efficiency Factors even when unfinned manifolds are used. The Collector Efficiency Factors are increased almost to the point where the heat transfer within the manifold has no influence on performance. With regard to pressure drop, it is seen from Table III-5 that circular headers meet the recommended limitations on ΔP .

A. Series Connection of Horizontal Rows - Parallel Connection of Rows

B. Series - Parallel Connection of Horizontal Rows

FIGURE III-16
INTERCONNECTION OF MULTIPLE HEAT PIPE COLLECTORS

		RECTANGULAR HEADER (Configuration #1)		CIRCULAR HEADER (Configuration #2)	
NUMBER OF HEADERS		1	2	1	2
8-Module Series Connection (Figure III-16A)	ΔP (ft)	47.8	13.8	5.4	1.5
	F'	--	--	0.877	0.921
4-Module Series Parallel-Connection (Figure III-16B)	ΔP (ft)	6.9	2.1	--	--
	F'	--	--	--	--

NOTE: Based on configuration listed in Table III-1.
Condenser width = 1.75 inch.

TABLE III-5
PRESSURE DROPS AND EFFICIENCY FACTORS OF
MULTIPLE WATER-HEATING HEAT PIPE COLLECTORS

The rectangular headers, on the other hand, result in excessive ΔP 's in an 8-module series connection. However, their thermal performance is slightly better. Therefore, for the rectangular headers, a series-parallel interconnection (Figure III-16B) was investigated. This results in acceptable pressure drops and good thermal performance.

In air heating systems a series connection of eight collectors is not as easily achievable. Table III-6 lists the air velocities for an individual collector for each of the configurations discussed under Task II. It is seen that the air velocity ranges from approximately 3 to 15 FPS. Standard engineering practice limits the velocity in trunk ducts to 15 FPS (Ref. 4). With this limitation, a maximum of five collectors of the Configuration 1 and fewer of the other configurations can be connected in series. Thus, the arrangement of Figure III-16A cannot be used in air heating systems. However, two of the collector configurations (marked with an asterisk in the Table III-6) can be used in a series-parallel connection.

An arrangement in which air flows vertically over the collectors was also investigated. Its implementation is shown in Figure III-17. Because of the larger available flow area, the velocities are sufficiently reduced, such that all four configurations considered under Task II can be used. This vertical airflow arrangement is, of course, also applicable to conventional air-heating collectors. The heat pipe collector, however, avoids air flowing in contact with the collector surface, allows prefabrication of complete modules, and uses thermal diodes.

Integration of heat pipe collectors with conventional roof constructions was also considered as part of this task. Typical constructions consist of sloping beams or rafters which are located on 16 to 24-inch centers. The rafters are covered by plywood sheathing which, in turn, is covered by the roofing material (shingles, etc.). One installation technique is to install the collectors on top of the plywood sheathing. This permits the use of any collector size independent of the location and spacing of rafters but requires the collection manifold (e.g., water or air) to be located on top of the sheathing. Although the collector would provide some insulation, nevertheless, freezing of the water in the manifold could occur. Thus, the nonfreezing advantage of a heat pipe collector would be lost.

CONFIGURATION**	#1		#2		#3		#4	
Condenser Length (inches)	12	16	12	16	12	16	12	16
Air Velocity in Single Collector (FPS)	4.3	3.2	5.1	3.8	10.3	7.7	14.5	10.9
Possible Number of Collectors in Series***	3	5*	3	4*	1	2	1	1

*These collector configurations can be used in a series-parallel connection.

**Reference Table III-3.

***Based on maximum air velocity of 15 FPS (900 FPM).

TABLE III-6
MULTIPLE CONNECTIONS OF AIR-HEATING SOLAR COLLECTORS

a. Side View

b. Schematic of Multiple Collector Arrangement

FIGURE III-17
HEAT PIPE AIR-HEATING COLLECTORS WITH VERTICAL AIRFLOW

A second installation is shown in Figure III-18. The collectors are located on the outside of the rafters but their condenser section, together with the manifold, is recessed below the rafter line and protected from the environment. However, this approach limits the collector width to that of the spacing between rafters and requires a bend in the collectors.

The third approach under consideration eliminates the bend in the collectors (Figure III-19). In this approach, the collectors are recessed below the rafter line and the manifold is again protected from freezing but the rafters will shade part of the collector during times of oblique solar incidence. The effects of shading can be mitigated by using reflective coating on the sides of the rafters. This may also create an architecturally interesting appearance. Ultimately, the reflectors could be combined with a finely segmented heat pipe panel.

FIGURE III-18

HEAT PIPE COLLECTOR WITH
OFFSET HEAT TRANSPORT MANIFOLD

SECTION A-A

FIGURE III-19

HEAT PIPE COLLECTORS RECESSED BETWEEN ROOF RAFTERS

IV. REFERENCES

- (1) "Dornier Sea Water Desalination Still with Solar Heated Heat Pipes," Technical Brochure Dornier System, Friedrichshafer, Germany
- (2) "Method of Testing for Rating Solar Collectors Based on Thermal Performance." Center for Building Technology, National Bureau of Standards, Washington, D. C. 20236, November 1, 1974
- (3) Whillier A., "Design Factors Influencing Solar Collector Performance," Low Temperature Engineering Applications of Solar Energy, pp. 27-40
- (4) "Mechanical Engineers' Handbook," by Lionel S. Marks, 5th Edition, 1951

V. APPENDIX

A. Research Contributors

The following Dynatherm Corporation personnel contributed to the technical work:

Dr. Walter B. Bienert, Principal Investigator

Mr. Donald S. Trimmer

Mr. Gilbert A. Wadsworth

Mr. David A. Wolf

Mr. David Wilson of the architectural firm of Wilson, Magruder, Webb and Ratych participated as a consultant.

B. Related Reports and Papers

During this reporting period the following technical papers and other reports were generated:

- (1) "Heat Pipes Applied to Flat-Plate Solar Collectors," Walter B. Bienert. Paper presented at NSF/RANN Workshop on Solar Collectors held during November 1974.
- (2) "Quarterly Management Report," DTM-74-12, December 1974.