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MASTER

THERMACORE, INC.

LEOLA, PENNSYLVANIA

COST-EFFECTIVE SOLAR COLLECTORS

USING

HEAT PIPES

INTERIM PROGRESS REPORT

#1

September 1977 - March 1978

Donald M. Ernst
Principal Investigator

There is no objection from the patent
point of view to the publication or
dissemination of the document(s)
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BROOKHAVEN PATENT GROUP

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Foreward

This report is submitted to the United States Department of Energy, Office of Conservation and Solar Applications, Research and Development Branch for Heating and Cooling, Dr. Frederick H. Morse, Chief, and covers the work conducted by Thermacore, Inc. under Contract DOE EG-77-C-04-4099. The work was performed under the direction of Dr. Stephen L. Sargent, Program Manager from the DOE R&D Branch, Heating and Cooling Office, and Dr. Charles Bankston, the contract monitor from the Los Alamos Scientific Laboratories.

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Abstract

The objective of the program is the demonstration of high performance, cost effective non-concentrating solar collectors using heat pipes. The end products will be directly applicable for efficient use with absorption and Rankine cycle chillers. Evacuated tubular solar collectors were selected as the only economical non-concentrating approach capable of efficient operation of chillers. The General Electric TC family of collectors was chosen because of their superior performance and compatibility with heat pipe integration.

The system was designed and specified. This work included the integration of the heat pipe with the evacuated tubular solar collector and the pumped loop heat removal mechanism. To date, two heat pipe fluid-envelope combinations look attractive: water-aluminum bearing steel and ethanol-low carbon steel. The jury is still out on the ability for the water-aluminum bearing steel to survive freezing cycles and for ethanol-low carbon steel to withstand predicted 400 C stagnation temperatures. Full scale cost analysis was not completed for either case.

Two 4' x 4' panels, each with ten tubular collectors fitted with heat pipes, were erected at Thermacore to test various aspects of the heat pipe and its integration into the collector-pumped loop system.

1. INTRODUCTION

This interim report documents the work performed by Thermacore, Inc. and its subcontractor and consultants during the time period September 1977 to March 1978 under DOE Contract EG-77-C-04-4099. The object of the program is to investigate the cost effectiveness of heat pipe solar collectors.

The program to date has analyzed pertinent collectors systems and established heat pipe design goals for optimum system performance. From this analysis, separate components were designed and a search for an improved heat pipe fluid begun. The component design was subjected to careful scrutiny from the systems integration and producibility viewpoints. Component fabrication and testing of heat pipes, collector-to-heat pipe and heat pipe-to-pumped fluid loop interfaces were carried out in laboratory testing and actual outdoor collector testing.

Fabrication testing and evaluation will continue, the results of which will be used to finalize a cost effective design and heat pipe processing techniques. Concurrently a full scale GE solar module will be retrofitted to accept heat pipes for design verification and testing.

To carry out the above activities, five interrelated tasks were defined as shown in Figure 1.1 Program Schedule and 1.2 Task Flow Diagram. To date, work has progressed into Task IV.

It is assumed that the reader is familiar with heat pipe

Figure 1.1 Program Schedule

Figure 1.2 Task Flow Diagram

operation. A review, including solar applications is given in Appendix 1.

1.1 Background

Heat pipes are compatible with evacuated tubular collectors in terms of both power and physical size. DOE has recognized this in its prior funding program¹. The heat pipe and evacuated collector are each inherently sealed entities. To minimize life-cycle costs, these components should be separately replaceable so that the failure of one does not require the replacement of the other. To minimize manufacturing costs, the collector design should place minimum constraints on heat pipe performance versus size. Thermacore concluded that the General Electric TC collector family meets these conditions and also provides superior thermal performance. Thermacore, therefore, enlisted GE's subcontract support in developing a fully integrated collector/heat pipe system under the program, to assure that realistic and practical systems applications considerations are factored into the heat pipe design.

The General Electric Model TC-100 evacuated tubular collector is shown in Figure 1.3. This collector is an important advancement over today's "flat plate" approach in terms of its daily collection efficiency and its cost per collected BTU. Efficient performance has been demonstrated at temperatures necessary for absorption and Rankine cycle chillers. The TC-100 is a factory-built, self-contained modular collector which incorporates mature, low cost

Figure 1.3 Evacuated Tubular Collector Panel

Figure 1.4 Evacuated Shroud

lamp bulb materials and proven commercial production techniques.

The collector's high performance is achieved by an evacuated glass shroud in a "thermos bottle" configuration. Figure 1.4 shows the evacuated space between the outer and inner tubes which minimizes front and back thermal losses from the selectively coated absorber surface. This unique "solar trap" concept has several advantages, for example:

1. The concept allows simple insertion of heat extraction components such as fin-looped tubes or fin-heat pipes, which enhances vacuum permanence by omitting glass-to-metal seals.
2. The working fluid circuit integrity is not affected by inadvertent breakage of glass shrouds.
3. Thin fins and small diameter tubes or heat pipes result in low collector thermal inertia. As a result, the solar energy needed to bring the system up to its operating temperatures is kept to a minimum and shroud thermal shock reduced.

The present TC-100 utilizes a thin fin attached to a "U" tube, forming a single element. Ten elements are then joined in series to form a serpentine. The system is effective, but several advantages can be realized by using a heat pipe which will result in either a direct or an indirect improvement of the cost-to-performance ratio for an installed system. The key advantages include:

1. The thermal diode effect of a heat pipe essentially removes a broken shroud from the heat exchanger loop, thereby reducing the need for constant maintenance.
2. It becomes virtually impossible to thermally shock the shrouds, thereby eliminating the need for sophisticated controls.

3. The system pumping parasitics may be reduced by not pumping across a large ΔP serpentine.
4. Mass of the element and manifold is reduced, thereby reducing the thermal inertia of the system.
5. Manifolding can be reduced to single pipe systems, thereby reducing installation costs.
6. Water can be used for primary loop service since single pipe manifolds can be built for draining.
7. Secondary exchange loops can be eliminated if water is used in the primary manifolds.

1.2 Summary

Four tubular collector systems were reviewed:

1. TC-100 (0.85 x) non-concentrating tubular collector;
2. TC-101, a modified TC-100 low concentrating (1.085 x) tubular collector;
3. a 1.5 x low concentrating tubular collector; and
4. a TC-300 low concentrating (2.9 x) tubular collector.

Each represents a viable collector system for solar heating and cooling applications. The TC-101, a modified TC-100 collector, was selected as the program's baseline collector. A preliminary product specification was prepared for the corresponding heat pipe heat removal assembly. A summary of these key requirements is seen in Table 1.1.

The TC-100 collector assembly consists of 10 evacuated collectors in a straight V-trough with associated U-tube serpentine, mechanical interface and pumped loop hardware. The TC-101 consists of 8 evacuated collectors in a modified Winston cusp with

TABLE 1.1. SUMMARY PRODUCT REQUIREMENTS FOR A
TC-101 HEAT PIPE HEAT REMOVAL ASSEMBLY

AREA	REQUIREMENTS
General	<ul style="list-style-type: none"> o Factory built unit o Mechanical or brazed connections o Transfer heat from collector absorber-to-fluid loop
Envelope	<ul style="list-style-type: none"> o Evaporator - 44.6" long, up to 1.5" in dia. o Condenser - 3" length, 5" width, 1" depth
Interface	<ul style="list-style-type: none"> o Mechanical - clamping, or sonic welding, or brazing o Hydraulic - clamped or brazed joint or weld immersion
Performance	<ul style="list-style-type: none"> o Ambient temp. -45° C to 52° C o Operating temp. 38° C to 163° C o Start-up temp. 16° C o Heat Flux (watts/heat pipe) <ul style="list-style-type: none"> - maximum 103 - normal range 8-94 - distribution uniform longitudinally
Effectiveness	<ul style="list-style-type: none"> o Heat pipe-to-collector - 24° C @ 38° C loop fluid Fluid bulk ΔT 7° C @ 150° C loop fluid o Stability - less than a 10% increase in ΔT over required life
Pressure Drop (Collector inlet-to-outlet)	<ul style="list-style-type: none"> o 5 - 7.5 psi
Life	<ul style="list-style-type: none"> o 15 years
Durability	<ul style="list-style-type: none"> o Survive no-flow stagnation temperatures of up to 410° C o Survive all temperature cycles

the same U-tube serpentine, mechanical interface and associated pumped loop hardware.

The only difference between the TC-100 and TC-101 is that the heat removal per individual tubular collector can be up to 28% higher for the TC-101 at the maximum insolation rate. This difference affects only the heat pipe. Thus the system design of the TC-100 and TC-101 are equivalent, and, as the TC-101 will replace the TC-100 and be called the TC-100, this report makes use of TC-100 and TC-101 interchangeably in discussion of the system.

Using the requirements established in Table 1.1 as Design Goals analyses were performed on interfaces, components and the system. The current heat pipe-manifold and heat pipe-absorber interface concepts are seen in Figure 1.5. The prime condenser-to-manifold concept selected was heat pipes immersed into the manifold. Bench testing produced heat pipe evaporator -to-circulating fluid bulk ΔT 's within the limits of the design goals.

A clamped or welded fin approach was selected as the collector absorber-to-heat pipe evaporator interface design. This approach is currently being used in the TC-100 collector, and represents a proven technique to transfer the absorbed solar energy from the tubular shroud to a longitudinal tube located inside the shroud. A 10°C increase in absorber-to-fluid bulk ΔT due to the added thermal resistance of the heat pipe was shown to represent less than a 3% decrease in annual percent solar energy collection and will probably be offset by the reduced losses due to the thermal

Figure 1.5 Interface Concepts

Figure 1.6 Outdoor Test Facility at Thermacore

diode effect of the heat pipe .

To date, 52 heat pipes, which are a combination of 9 envelope materials and 8 heat pipe fluids, have been built and tested. Accumulated testing totals 17,000 hours and include 20 heat pipes installed in an outdoor test facility at Thermacore as seen in Figure 1.6. Ten heat pipes are currently being fabricated. Additional heat pipes will be fabricated and installed as the program continues.

Heat pipe testing revealed two fluid-envelope combinations which look attractive: water-aluminum bearing steel and ethanol-low carbon steel. The ability of the water-aluminum bearing steel to withstand freeze-thaw cycles is being investigated, as is ethanol-steel heat pipes' ability to withstand the theoretical stagnation temperature of 410 C.

Bench testing of 5/16" OD x 48" long heat pipes in the 38 - 165 C range at 90 - 110 watts has been carried out with no apparent operational limitations. As 38 C is approached with a water heat pipe, its operation approaches sonic conditions, and the condenser thermally decouples from the evaporator and operation continues with the condenser temperatures following the heat sink temperature and the evaporator maintaining a temperature near 38 C.

Freeze cycle tests have produced encouraging results in water heat pipes. Natural outdoor freezing of heat pipes not in a tubular collector appear to behave differently (less increase in pipe diameter) than quick freezing of heat pipes in a commercial

ice plant. Accordingly it is felt that outdoor natural freezing of heat pipes within the collectors will produce less total expansion and may allow for the use of water as the working fluid.

2. HEAT PIPE COLLECTOR DESIGN REQUIREMENTS

Four non-to-low concentration tubular collectors were evaluated:

1. TC-100 non-concentrating collector (0.86 x);
2. TC-101, a low concentrating (1.085 x) modified TC-100 collector;
3. a 1.5 x collector similar to the collector system currently under investigation at ANL; and
4. a TC-300 low concentrating (2.9 x) collector currently under development at GE.

The objective of the evaluation was to select a collector that has an attractive \$/BTU rating as well as representing a near-term (8 mo. to 1½ years) commercially available solar product suitable for solar heating and cooling applications. During the proposal phase of the current program, the GE Model TC-100 vacuum tube collector was used as the baseline. This collector is currently in production and represents one of the most cost effective collectors presently available.

2.1 Collector Selection

As with any product, improvements are continually being investigated. To this end, GE initiated a product improvement program that had as its aim the reduction of the \$/BTU value for the TC-100 tubular collector. As a result of this improvement program, a modified TC-100 (TC-101) tubular collector is being developed. This collector represents a value engineered collector

Figure 2.1 Annual Energy Collection Comparison

which results in a 15% reduction in the \$/BTU rating. This improvement was primarily accomplished by incorporating a more efficient reflector which has a geometric concentration ratio of 1.085. GE is currently in the process of introducing this improved collector into its family of solar products.

Figure 2.1 shows why the TC-101 has a better \$/BTU rating within the same 4' x 4' aperture. The TC-100 collects 2060 kW/yr with 10 shrouds in a "V" reflector. The TC-101 collects 2410 kW/yr with 8 shrouds in a modified Winston cusp. This is a 17% increase in energy collected for a 20% decrease in shroud cost. Factoring in the added cost of the cusp over the "V" trough results in the 15% reduction in the \$/BTU rating.

Of the 4 collector systems evaluated, the TC-101 tubular collector was selected as the program baseline because of its attractive \$/BTU rating and the fact that it has the best chance of becoming a high volume commercial product.

2.2 Heat Pipe Specifications

The objective of this task was to prepare a product specification for a heat pipe heat removal assembly. A secondary objective was to define typical heat pipe design requirements for a family of low concentrating tubular collectors.

In designing a heat pipe heat removal assembly, collector and system constraints must be considered. Each of the 4 candidate collector systems was evaluated to determine typical operating conditions and design constraints. A summary of the heat pipe design

conditions/constraints is presented in Table 2.1. A companion curve, Figure 2.2, is provided to show typical heat pipe heat transport requirements for each of the collectors vs absorber temperature. This table and curve can be used to provide insight into heat pipe performance and size requirements. Additionally, they can be used to evaluate the growth potential of current heat pipe designs.

Using the TC-101 collector, typical requirements were established for performance, interface, envelope, physical characteristic, maintainability, environmental, transportability and qualification tests. The Useful Heat Flux curve for a TC-101 collector is presented in Figure 2.3. Table 2.2 summarizes the allowable ΔT 's from the heat pipe evaporator to the collector fluid bulk for a 2% performance degradation as compared to the TC-101. From Figure 2.3, one can see how heat flux or heat pipe heat transport requirement varies with absorber temperature, incident solar energy and representative absorber coating emissivity values.

From Table 2.2 it can be seen that the lower the incident solar energy and the higher the emissivity, the more critical a low evaporator-to-collector fluid bulk ΔT becomes. Likewise, the lower the nominal operating fluid temperature, the less critical this ΔT becomes. However, since the temperature-time curve is skewed to the lower temperatures, the actual integrated daily performance will be greater as the heat pipe evaporator-to-collector fluid ΔT is shown to be always less than the 2% degradation curve.

TABLE 2.1. HEAT PIPE DESIGN REQUIREMENTS
FOR CANDIDATE COLLECTOR SYSTEMS

HEAT PIPE DESIGN CONDITIONS/ CONSTRAINTS	TYPICAL DESIGN PARAMETERS FOR COLLECTOR SYSTEMS			
	TC-100	TC-101	1.5 X	2.9 X
Operating Temp. °C Normal Range	38 - 150	38 - 163	38 - 204	177 - 215
Operating Heat Load per tube - watts				
Maximum	85	109	150	260
Normal Range	7.3 - 75.5	7.3 - 93	7.3 - 137	7.3 - 235
Evaporator Envelope Length - inches	44.6	44.6	44.6	44.6
Diameter - inches	1.5	1.5	1.5	1.5
<u>Loop Interface Constraints</u>				
Inlet-outlet of Heat Pipe manifold - ΔP - psi	5-6	5-6	5-6	4-7
Heat Pipe to fluid bulk ΔT °C	38°C - 22 150 C - 6	38° C - 24 150 C - 7	38° C - 25 204 C - 8	177° C- 9 204 C-12
Condenser Interface Envelope - LxWxH	3.5 x 4.5 x 1"	3.5 x 5 x 1"	4.5 x 5.5 x 1"	10 x 7 x 1.5"
Start-up Sink Temp. °C	16	16	16	16
<u>Orientation</u> Incline Slope range-degrees	Variable 0-90	Variable 0-90	Variable 0-90	Horizontal 0-5
Ambient Temperature °C	-45 to 52	-45 to 52	-45 to 52	-45 to 52
Heat Source	I.D. of Glass Tube	I.D. of Glass Tube	I.D. of Glass Tube	I.D. of Glass Tube
<u>Environment</u> Corrosion Resist- ance	Aqueous Solution	Aqueous Solution	Aqueous Solution	Aqueous Solution
Stagnation/oxida- tion resistance	382°C	410°C	425°C	287°C
Life - years	15	15	15	15

Figure 2.2 Power vs Absorber Temperature

Figure 2.3 Useful Heat Flux Curve

TABLE 2.2. ALLOWABLE ΔT 'S FROM HEAT PIPE EVAPORATOR
TO COLLECTOR FLUID BULK

Incident Flux Watts/cm	Emissivity Normal @ 100° C	Normal Oper- ating Temp °C	Allowable ΔT - C @ 2% Degradation
.10 .10	.04 .04	38 150	24 7
.10 .10	.06 .06	38 150	18 5
.06 .06	.04 .04	38 150	14 5
.06 .06	.06 .06	38 150	10 3

3. HARDWARE DESIGN

The necessary hardware and its relevant designs can be categorized as follows:

1. Collector absorber-to-heat pipe evaporator
2. Heat pipe
3. Heat pipe condenser-to-collector fluid loop
4. Manifold
5. Collector

The hardware design which evolved as a result of the system analysis was arrived at via a trade-off on heat transport capability of a large heat pipe versus the ability to get the heat out of the condenser to a pumped loop with the overall smallest ΔT at the lowest cost with the smallest additional non-collecting area added to the system. (See Figure 3.1)

3.1 Collector Absorber-to-Heat Pipe Evaporator

The hardware design for this interface will be similar to the design currently used on the GE model TC-100. Copper or aluminum fins that are welded or clamped to the heat pipe and subsequently preformed to the inside curvature of the tubular shroud such that a maximum air gap of 0.5 mm (.020 inch) is maintained, and for the most part actual contact is made. As the area is large, the heat flux is low; thus a 0.5 mm air gap would result in less than a 3° C ΔT at maximum insolation. Likewise, the reflector provides a non-linear distribution of heat to the absorber; thus by placing the

heat pipe along the bottom of the glass shroud, the thermal drop through the copper fin is less than 5° C. (See Figure 3.1).

The TC-100 fins are currently copper which are ultrasonic welded to the copper U-tube and have been used successfully in 6" and 24" lengths. Alternate methods of attaching the fins to copper and steel pipes are being investigated. The use of aluminum fins will dictate mechanical clampings and is under investigation. Copper fins which have been laser welded to copper and steel pipes are being obtained from Hamilton Standard for test and evaluation.

Outdoor and bench testing at Thermacore is being carried out on copper and steel pipes which are continuously and intermittently brazed to copper fins. To date, no differences or problems have been encountered.

3.2 Heat Pipe

A heat pipe design was selected which was felt to be capable of handling the heat transport requirements for some of the commonly accepted heat pipe fluids such as water, methanol and ethanol. This design consists of a 5/16" OD tubular vessel with a .02" to .035" wall. The variation in wall thickness accomodates the internal pressures to be encountered for different fluids while keeping the material content to a minimum.

The evaporator length is 42.5", which is 2.1 inches shorter than the absorber and fin length. The 3" condenser was chosen as a minimum length in order to keep the film ΔT , in an immersion

Figure 3.1 Heat Pipe Design

heat pipe-to-manifold design, to an acceptable value, as discussed in 3.3. The heat transport zone is 2.5" long, of which 1" is used to accomodate mechanical connection to the hydraulic manifold. The remaining 1.5" of heat transport zone is for a dog leg bend in the heat pipe in order to provide more space for foam insulation around the manifold.

The condenser end of the heat pipe will be swaged to a 0.225" diameter, which when pinched off and welded will have a maximum dimension of less than 5/16", thus the condenser can be inserted into a 5/16" clearance hole in the manifold.

3.3 Heat Pipe Condenser-to-Collector Fluid Loop

A thermal analysis was performed to determine if it was practical to remove the heat from the condenser portion of the heat pipe with a small enough area to be practical with respect to physical size. If the above proved feasible, the analysis sought to determine the appropriate size of the condenser, manifold film delta temperatures, necessary flow rates and manifold fluid flow path sizes as well as pressure drops.

Two heat transfer configurations were evaluated: surface contact by a welded or mechanical clamp interface as seen in Figure 3.2 and immersion where the heat pipe is immersed directly into the manifold flow path. A third configuration is currently undergoing evaluation. It consists of an immersion well contained within the manifold such that contact is by a .002" gap filled with a thermally conductive compound. Both immersion concepts are

Figure 3.2 Mechanical Interface Concepts

seen in Figure 3.3

The first interface considered was surface contact. In this configuration the heat pipe condenser would be mechanically welded or clamped to the manifold heat collection area. One problem encountered with this type of interface was that the whole heat pipe condenser was not offered as heat transfer area to the manifold heat collector. Thus conduction of heat must take place circumferentially around the heat pipe condenser for that area not directly in contact with the manifold.

Figure 3.4 represents calculated values for the fluid film ΔT vs flow rate for surface contact and immersion interfaces. A 5/16" OD heat pipe with a 3" condenser was used for these calculations. It can be seen that immersion provides the lowest ΔT for the two interfaces and water is the better fluid for the pumped loop.

Figure 3.5 shows the pressure drop through a typical heat pipe manifold for several tube ID's in the range of interest. From a system standpoint, a ΔP of approximately 5-6 psi is desired, which translates to a manifold inside diameter of $\approx .37"$.

Additional analyses were conducted to evaluate ΔT vs. contact length, water vs 40% Prestone II and water, as well as inlet-to-outlet ΔT for a heat pipe collector.

As a result of these analyses the following conclusions were reached:

1. A much more dramatic decrease in film ΔT can be accomplished by decreasing the manifold diameter than by increasing contact or immersion length.

Figure 3.3 Immersion Concepts

Figure 3.4 Calculated Film ΔT 's

Figure 3.5 Manifold Pressure Drop

2. The immersion type heat transfer configuration results in the lowest ΔT and should be used as the prime interface approach.
3. Increased collector concentration or heat flux will result in an increased film ΔT for the same length contact or immersion.
4. The manifold pressure drop can be tailored to be more compatible with collector system flow distribution requirements.
5. A 40% Prestone II/water fluid results in a factor of two increase in film ΔT as compared to a water fluid system.
6. The final manifold design should strive for the lowest flow rate and smallest diameter flow path as possible. Typically, the manifold flow should be around .8 gpm, the inlet-outlet ΔT around 25° F and the manifold ΔP around 5-6 psi.

As part of the heat pipe condenser-to-pumped loop interface design, a materials compatibility and fluid life analysis was done for the more common ferrous and nonferrous metals and water and Prestone II. The result of the analysis was favorable and is Appendix 2.

3.4 Manifold

The initial immersion manifold concept is seen in Figure 3.6 which was used in bench testing of the interface design. However, real world costs prevent the use of such a design in production.

Several manifold manufacturers were contacted to determine their potential assistance in the design and assembly of a heat pipe collector manifold. From this investigation it was determined that the collector manufacturer would design and build the manifold.

In order to evaluate the impact of heat pipes in a typical collector system, a preliminary system integration analysis was

Figure 3.6 Initial Manifold Concept

performed. Specifically, the analysis investigated collector loop pumping requirements, flow distribution, collector mounting and performance. As a result of this analysis, several important design changes were identified:

1. For residential installation (18 collectors) the manifold flow rate should be less than 50 cc/sec (.8 gpm) with a 5-7 psi pressure drop.
2. For large industrial installations (300 collectors) the manifold flow rate should be less than 32 cc/sec (.5 gpm) with a 5-7 psi pressure drop.
3. The heat pipe manifold compartment must be able to pivot up to allow for convenient shroud removal and replacement.

A joint manufacturing and engineering study produced an attractive production manifold design. This design consists of two die stamp sheets of steel brazed together to form the manifold. This manifold design incorporates low cost materials and production techniques. The present design will use compression fittings to join the heat pipe to the manifold.

3.5 Heat Pipe Collector Design

A joint Manufacturing and Engineering study generated the prototype production oriented heat pipe collector design, as seen in Figure 3.7. The design consists of the following basic parts:

1. Steel collector
2. Reflector Tray Assembly
3. Eight evacuated tubular absorber shrouds
4. Rigid foam insulation compartment
5. Eight heat pipes

Figure 3.7 Heat Pipe Collector Design

6. One die-stamped low carbon steel manifold
7. Collector pigtails (2) with hydraulic fittings

Basically, the manifold design consists of two plates of low carbon steel with the fluid flow path die-stamped into each plate. The two plates will be brazed together to form the manifold. By present thinking, the heat pipes will be connected to the manifold with standard compression fittings. This feature will permit easy installation and removal of the heat pipes.

An analysis was performed to evaluate the potential thermal expansion problem. In the present TC-100 "U" tubes or transition loops are used between each shroud to compensate for any thermal growth. This approach was considered for the heat pipe manifold but was eliminated due to costs and complexity. A possible approach would be to anchor the manifold in the middle and accept the thermal growth.

An analysis was conducted to determine the expected thermal growth and ascertain the resultant impact on collector operation.

A summary of this analysis is as follows:

1. The maximum thermal expansion for a 48" manifold at 315° C (600°F) stagnation temperature is 3 mm (.125").
2. The effects on fin to glass air gap if the fin is considered to be dead soft, is an average increase in the average air gap of .5 mm (.02"). This would cause an average collector performance degradation of about 1% at 38° C (100° F) inlet and 2.5% at 150° C (300° F) inlet.
3. Since most of the TC-101 operation will be at inlet temperatures around 200° F and present thermal expansion designs are costly, it is recommended that the loss in performance due to thermal expansion be accepted.

In order to facilitate tubular shroud removal and replacement a novel feature has been built into the heat pipe collector design. The reflector tray assembly and foam insulation/manifold compartment are attached to the collector frame such that they can be raised up enough to permit the removal/installation of the tubular shrouds, as seen in Figure 3.8.

Figure 3.8 Shroud Removal/Installation

4. SYSTEM INTEGRATION

4.1 System Integration

A variety of factors must be considered when designing a collector for solar system applications. To draw attention to these factors and identify possible system dictated design changes, a system integration evaluation was conducted on an initial heat pipe collector design.

During the evaluation the following areas were examined:

1. Reverse mounting of collectors
2. Pressure drop across the collector
3. Header sizing for proper flow distribution
4. Parasitic losses (pipe friction)
5. Operating temperature and pressure
6. Pump requirements
7. Primary loop fluid inventory
8. Excess energy heat dump
9. Primary to secondary loop heat exchanger (HX-1)

As a result of the evaluation the following conclusions were reached:

1. In order to provide a reasonable immersion length of 3 inches for the heat pipe condenser the length of the existing TC-100 internal manifold compartment was increased by approximately 4 inches. In addition, the heat pipe collector was mounted with the heat pipe condenser up. This requirement meant that the heat pipe collector was reverse mounted as compared to an existing TC-101. The overall result was that the effective gross length of the collector was increased by the increased length of the manifold compartment plus the length of the interconnecting collector row header which normally is below grade within the supporting framework.

The combination of the increased collector length plus the need to reverse mount the collector required that the collector row spacing was 27% greater than that of the standard TC-101 system to limit shading effects. In essence the packing factor for a heat pipe collector was less than that of a standard TC-101 collector.

2. Pressure drop across the TC-101 heat pipe collector may be less than that of the TC-100.
3. In order to achieve the 1/10 collector pressure drop thru the header pipe to assure proper flow distribution, the header pipe diameter will be much larger than the 3/4" of the TC-100 system.
4. Due to the high flow rate larger diameter field piping will be required to minimize field piping pressure drops.
5. Operating temperature and pressure were above that of the TC-100 system but below that of the TC-101 collector. Primary loop components used for the TC-101 collector were found to be more than adequate for use with the TC-101 heat pipe.
6. Pump head requirements for the heat pipe were lower than the equivalent TC-100 system. Due to the heat pipes high flow rate the water horsepower and consequent power requirements of the pump motor were greater than those of a TC-100 system.
7. Required fluid in the heat pipe primary loop was shown to be greater than that of the TC-100 system if larger pipe diameters were used. The use of a larger pipe diameter was desirable to achieve the same pressure drop thru the piping as occurs in the TC-100 system.
8. Excess energy to be dissipated thru the heat dump was the same for the TC-100 and TC-101 heat pipe. Higher flow rates of the heat pipe system may require resizing of the heat transfer area.
9. Heat exchanger ΔT of the heat pipe system was calculated as being much lower than that of the TC-100. The amount of heat to be transferred by HX-1 was the same in both systems. High flow rates of the heat pipe collector necessitated selection of a larger HX than would be used for an equivalent TC-100 system.

The above conclusions were based on an evaluation of an early

heat pipe design. As a result of this evaluation the heat pipe manifold is presently being redesigned to provide a more optimum ΔP and flow rate. After the redesign effort the system integration evaluation will be repeated to quantify header pipe size, loop flow rate, loop piping pressure drop and fluid inventory.

4.2 System Performance

A study was conducted to determine the influence of the heat pipe on the performance of the TC-101 collector. All year energy collection studies were made using a single family home in Fort Worth as a load model for heating, cooling, and domestic water pre-heating. The influence of the heat pipe was simulated by varying an overall heat transfer conductance between the collector fin and the fluid in the collector, i.e. the heat pipe evaporator to bulk fluid ΔT . An increased ΔT at this point in the system implies that the collector absorber must run at a higher temperature for a given collector fluid temperature. In general, losses are directly proportional to the absorber temperature. The question is how does this increased ΔT affect the annual percent solar energy utilization for a typical heating and cooling application. The results of the study are presented in Figure 4.1 in terms of the percent solar energy utilization compared to a TC-101 standard collector.

From this curve one can evaluate how increased ΔT affects the annual performance of a typical solar application. At present the data implies that a heat pipe collector will experience less than a 10°C (18°F) increase in ΔT as compared to a standard TC-101.

Figure 4.1 Heat Pipe Influence on Energy Collected Annually

This increased ΔT corresponds to an effective conductance of around 18 which will result in 3% reduction in percent solar energy utilization.

5. TEST RESULTS

As of this report, 52 heat pipes have been fabricated with a total accumulation of 17,000 hours of bench testing. An additional 370 heat pipe days have been logged on the outdoor test facility, along with 2100 hours of laboratory reflux test samples and 127 freeze-thaw cycles.

These tests have been performed on a combination of 8 fluid and 9 vessel materials, as seen in Table 5.1, and represent 19 different combinations.

5.1 Fluid Search

The environmental limits of -40 to +410 C for a heat pipe which has a 38 to 165 C operating range imposes several interesting problems. Accordingly, a search was undertaken in an attempt to define a workable heat pipe fluid. Several approaches with different goals were defined as follows:

1. Passivation or protection of a steel vessel wall for use with water and a freeze resistance wick.
2. The addition of a non, or slightly volatile solute to lower the freezing point of water.
3. The use of a "new" pure fluid.
4. The use of inhibitor to reduce corrosion.

To date 5 fluids have been tested with 1010 steel in refluxing glass capsules as seen in Table 5.2. The column headed "Pressure Torr" indicates the pressure of evolved gas over the test sample at

TABLE 5.1. VESSEL-FLUID COMBINATIONS

VESSEL-FLUID	WATER	METHANOL	ETHANOL	WATER & LITHIUM CHLORIDE	WATER & LITHIUM CHLORIDE PH 12	WATER & SODIUM HYDROXIDE	WATER & PHOSPHORIC OXIDE	TRIMETHYL BORATE
	H ₂ O	CH ₃ OH	CH ₄ OH	H ₂ O + LiCl	H ₂ O + LiCl + NaOH	H ₂ O + NaOH	H ₂ O + P ₂ O ₅	(CH ₃ O) ₃ B
COPPER								
CDA - 101	X							
CDA - 122	X							
STEEL								
1010	X	X	X	X	X	X	X	X
1010*	X			X			X	
1010 A1**	X							
1010 A1***	X							
1018	X		X					
NP 254	X			Y	Y			
18 SR	X			Y	Y			
* Passified fuming nitric acid ** Alonized steel *** Oxidized alonized steel X - tested to date Y - to be tested								

TABLE 5.2. REFLUX FLUID TEST WITH STEEL

FLUID	TEMP °C	TEST HOURS	TOTAL HOURS	PRESSURE TORR	VISUAL REACTION
(CH ₃ O) ₃ B	65	160	160	0	No
	88	36	196	0	No
	95	216	412	0	No
	95	192	604	0	No
	130	100	704	0	No
H ₂ O + LiCl + NaOH	88	36	36	2	No
	95	216	252	0	No
	95	192	444	0	No
	130	100	544	0	No
H ₂ O + LiCl	88	36	36	2	Yes
	95	216	252	25	Yes
	95	192	444	4	Yes
H ₂ O + NaOH	88	24	24	0	Yes
	95	216	240	6	Yes
	95	192	432	1	Yes
H ₂ O + P ₂ O ₅	88	1	1	High	Yes

room temperature after it has been expanded 10:1 into a gas sampling system.

From Table 5.2 two fluids look promising: trimethyl borate and alkaline lithium chloride. The initial gas evolution of the alkaline lithium chloride solution is believed to have resulted from an inadvertent leak in the sampling system.

Trimethyl borate has a -27 C freezing point and a 68 C boiling point. At 25 C, its viscosity is 0.35 cp and its surface tension is 25 dy/cm, both about 33% that of water. It is reported to be thermally stable to 700 C.

Actual testing of the fluid in a steel heat pipe is semi-encouraging. Its performance is stable with no degradation after 600 hours at 165 C. However, the heat pipe had a relative larger ΔT in it from the beginning and which was believed to result from hydrolyzed trimethyl borate. Trimethyl borate hydrolyzes into boric acid and methanol. The presence of a small quantity of methanol in the heat pipe is analogous to water in an ethanol heat pipe and a possible cause of the larger ΔT .

The alkaline lithium chloride is 5.88 molar solution (250 gm LiCl/liter H₂O) with enough NaOH added to get a ph of 12. Its freezing point is -47 C. A steel heat pipe is being prepared to be tested with the solution.

The addition of sodium hydroxide increases the concentration of the hydroxyl ion which is believed to form a recombination process with any hydrogen generated by the attack of the solution on the

steel walls. It is a known fact that the decomposition product of a water-lithium chloride heat pipe without the addition of sodium hydroxide is hydrogen. This was determined when a gas filled heat pipe was opened to atmosphere and the gas ignited as it expelled from the heat pipe.

Several approaches are yet to be tested and include the use of boiler scale to protect the steel wall from the reaction of water, several silicon based compounds and the testing of a very dilute solution of a dissolved solute such as lithium chloride.

5.2 Life Tests

Significant life test results are seen in Table 5.3. The processing and life test setups are seen in Figure 5.1. The copper-water and 1010 steel-water heat pipes are being tested for comparison norms. Table 5.4 details the operating conditions of the bench tests. The increase in $\Delta T/Q$ for the NP 254 heat pipe occurred in the first several hours and is believed to be gas evolved from the seam weld along the length of the heat pipe, as it was fabricated from sheet stock.

NP 254 and 18 SR are aluminum bearing alloys. Their chemical analyses are seen in Table 5.5. These heat pipes were air fired for one hour at 800 C to diffuse aluminum to the surface where it oxidized forming a protective coating of aluminum oxide, which is compatible with water. Tests with un-oxidized heat pipes show very rapid gas generation until a protective layer of oxide is formed.

In an effort to find other low cost aluminum bearing alloys for

TABLE 5.3. LIFE TEST RESULTS

<u>BENCH TEST</u>		
CDA - 101	H ₂ O	3600 hours
1010 St	Ethanol	1500 hours
1010 St	H ₂ O	4100 hours
NP 254	H ₂ O	1000 hours
<u>OUTDOOR TEST</u>		
CDA - 122	H ₂ O	10 x 30 days
1010 St	Ethanol	5 x 7 days
1018 St	Ethanol	5 x 7 days

TABLE 5.4. OPERATING CONDITIONS

HEAT PIPE WALL	FLUID	TEMP °C	POWER WATTS	ΔT °C	$\Delta T/Q$ - C Wall INITIAL 5/2/78	Hours
CDA-101	H ₂ O	140	65	4	.06 .1	3600
1010 St	Ethanol	156	82	9	.11 .13	1500
NP 254	H ₂ O	198	80	8	.1 .17	1000
18 SR	H ₂ O	158	75	7	.11 .11	1000
1010 St	H ₂ O	275	79	11	.14 .42	4100

TABLE 5.5. CHEMICAL ANALYSIS

NP 254 - ARMCO EXPERIMENTAL ALLOY

18 SR - ARMCO SPECIALITY PRODUCTS

	C	Mn	P	S	Si	Cr	Ti	Al	Ni	Fe
NP 254	0.05	0.27	0.007	0.007	0.75	1.93	0.48	1.82		Rem
18 SR	0.05	0.50	0.020	0.010	1.0	18.0	0.40	2.0	0.5	Rem

Figure 5.1 Life Test

testing, several steel heat pipes were Alonized, i.e. aluminum was diffused with the inside surface of the heat pipe. Oxidized and non-oxidized alonized heat pipes generated hydrogen at an exceptionally high rate. The result is believed to be similar to water's reaction with pure aluminum. Alonized steel has a 50% surface layer of alloyed aluminum with a total of 20% aluminum within the .015 inch deep diffusion zone.

5.3 Performance Testing

Performance testing of heat pipes has several categories which are all interrelated. Most heat pipe performance testing focuses on obtaining the maximum thermal capacity of the heat pipe at any given temperature. This program interest is slightly different in that once a maximum power has been identified as in the design requirement, the evaporator to condenser ΔT and overall thermal conductance at various temperature and tilt angles are the performance of interest.

Testing electrically heated heat pipes with and without the copper absorber conduction fins produced interesting results. Figure 5.2 shows the evaporator to condenser ΔT for water-copper heat pipes when the heater was wrapped directly around the heat pipe. Figure 5.3 shows the evaporator to condenser ΔT for copper-water heat pipes when the heater was wrapped around the copper conduction fins. The difference is attributed to the fact that the unfinned heat pipes had localized hot spots along the evaporator which came from the electrical heater being in direct contact with the heat pipe.

The performance of the two types of heat pipes below 45 C is

Figure 5.2 Heat Pipe ΔT vs Maximum Evaporator Temperature

• $Q = 90$ WATTS
 T_{MAX} - °C - PUMPED WATER HEAT SINK WITH PIPE IMMERSED
 • INPUT THROUGH COPPER FINS

Figure 5.3 Heat Pipe ΔT vs Maximum Evaporator Temperature

identical and the increasing ΔT with decreasing temperature is associated with a pressure/temperature drop in the vapor as the heat pipe nears the sonic vapor velocity operating condition.

The data in Figures 5.2 and 5.3 were taken at various operating angles between 15 and 45 degrees. To date, testing has been bounded by these angles as they are considered to be the normal extremes for properly installed systems in all but the most northern part of the U.S. and covers over 95% of expected usage areas. Testing from 1 to 90 degrees will be carried out during the remaining portion of the program.

An unfinned heat pipe was tested using various sized water cooled clamped heat sinks as a thermal load. Its performance is seen in Figure 5.4 along with the thermal transport requirement of the heat pipe as defined by the system analysis and the finned and unfinned ΔT 's as shown in Figures 5.2 and 5.3. Also plotted in Figure 5.4 is the sonic power limit as described by Busse² and the vertical flooding power as described by Sakhuja.³ How the vertical flooding limit is affected by a tilt is not known at the present time. It is interesting to note that the heat pipe operated along its performance curve in a very stable manner and attempts to drive the heat pipe into a non-operating mode or past a limit were unsuccessful.

Since the actual performance which is of interest to the collection system is the overall ΔT from the heat pipe evaporator to fluid bulk, a test was set up as seen in Figure 5.5 to measure the overall thermal conductance of the heat pipe evaporator to fluid

Figure 5.4 Heat Pipe Performance

Figure 5.5 Performance Test

for a three inch long condenser immersed into the pumped fluid. The heat transfer coefficient or thermal conductance which was measured is the sum of six conductances: the evaporator wall, the internal evaporation coefficient, the ΔT in the heat pipe vapor, the internal condensation coefficient, the condenser wall and the outside fluid to condenser film coefficient. By placement of thermocouples on the outside of the evaporator, the outside of the insulated heat transport zone, which is the same temperature as the vapor at that point and within the pumped fluid a measure of the sum of the first three and last three conductances could be obtained.

Figures 5.6, 5.7 and 5.8 show the measured values of the heat pipe vapor to bulk fluid heat transfer coefficient for water, Prestone II and 40% Prestone II in water. Figure 5.9 shows the overall evaporator to fluid bulk conductance for water and 40% Prestone II at 32 and 50 cc/sec as compared to the desired conductance value established in the analysis for an allowable 2% degradation. 50 cc/sec (.8 gpm) corresponds to the maximum flow rate for an 18 collector panel residential system and 32 cc/sec (.5 gpm) corresponds to the maximum flow rate for a 300 collector panel commercial system.

The test set-up was also used to performance test the heat pipes below 38 C, which corresponds to the system's lower operating point. However, start-up may occur as low as 16 C.

Full power performance has been accomplished with 10 C water circulating over the heat pipes at a rate of 100 cc/sec (1.6 gpm). The heat pipes operate at 30 C with a 7 C evaporator to condenser ΔT and a 13 C ΔT external film heat transfer coefficient.

Figure 5.6 Heat Transfer Coefficient ($\Delta T^\circ\text{C}$) vs Flow Rate for 100% Water

Figure 5.7 Heat Transfer Coefficient (ΔT) vs Flow Rate for 100% Prestone II

Figure 5.8 Heat Transfer Coefficient
($\Delta T ^\circ\text{C}$) vs Flow Rate

Figure 5.9 Fluid Bulk Temperature

Full power start-up with the same 10 C, 100 cc/sec water flowing over the heat pipe has been successfully demonstrated numerous times. No attempt is made on start-up to ease the heat pipes into operation, thus any time the test is started up it goes through a full power start-up into a load that generally exceeds the design requirements.

5.4 Freeze Testing

Obviously ethanol, trimethyl borate and lithium chloride - pH 12, heat pipes do not need to be freeze tested. However, water heat pipes do. To date, three water heat pipes have undergone 127 freeze-thaw cycles as seen in Figure 5.10.

The copper-water heat pipe with a copper mesh wick failed after 18 cycles. Failure is attributed to the fact that the wick was restrained by the pipe wall and not allowed to move freely with the forces of freezing and thawing. Accordingly, a retest with a loose fitting wick will be carried out.

The fiberglass wicked heat pipe has undergone 47 cycles with virtually no change in heat pipe diameter and is very similar to Thermacore's initial tests. However, testing of the fiberglass wick in copper and steel at temperatures from 150 C to 410 C showed that the fiberglass is slowly attacked by the water, impairing heat pipe performance.

The unwicked copper-water heat pipes' ability to survive 68 cycles is attributed to a low but sufficient fluid charge.

The arrows in Figure 5.10 indicate where forced fast freezing replaced slow natural freezing. As can be seen in the copper mesh

Figure 5.10 Freezing Test

wicked heat pipe, a quantum jump occurred in the pipe diameter, and is part of the reason it failed soon thereafter.

This apparent difference in the way in which the rate of freezing affects the forces exerted on the heat pipe is interesting. It may be possible to take advantage of the fact that the heat pipes in a real installation will be insulated with the shroud, thereby slowing the rate of freezing. Accordingly, it may be possible to lower the freezing point of water to -1 to 5 C with a minute amount of solute and, as the water freezes out against the heat pipe wall, the concentration of the solute increases, thereby lowering its freezing point. As the inner cylinder becomes slushy it may be slowly extruded into the vapor space relieving the freezing stresses.

5.5 Stagnation Testing

Several bench heat pipes have been tested to 410 C and the expected result occurred. The wall of the heat pipe outgassed. The non-condensables collected in the condenser and impaired heat pipe performance. Thus a vacuum firing schedule has been instituted and is part of the overall processing-assembly procedure being developed.

Several heat pipes for stagnation testing at GE are in fabrication and will be tested at full solar stagnation. Several outgassed compatible fluid-vessel combinations will be set up and life tested at 410 C.

The use of fluids, such as ethanol, which are above their critical temperature at 410 C, require that the fluid charge be carefully controlled such that its mass is less than the internal volume

of the heat pipe times the critical density. This will help keep the pressure to a practical value.

In addition to outgassing at stagnation condition, care must be taken to insure against fluid breakdown and corrosion. If testing of nonfreezing fluids at stagnation does not produce a viable system, there are at least two heat pipe options open. The first would be to make each heat pipe a gas controlled heat pipe with an excess power dump. The second would employ one single heat pipe attached to the manifold to regulate stagnation.

Either individual or collective control would introduce an unwanted thermal loss in the system, which occurs as the temperature of the control heat pipe approaches the cutoff value. However, this could be kept to a minimum.

The individually controlled heat pipe would also have a lower operating temperature at which the entire condenser area would be blocked by the control gas. This would then force the heat removal to be via conduction along the heat pipe wall imposing an unwanted ΔT and loss in efficiency. This lower operating temperature and resultant ΔT occurs at the highest efficiency part of the collector's performance where the allowable ΔT 's are large for a small degradation and may still fall within the allowable curve of Figure 5.9.

In order to fully evaluate stagnation control, a collective gas controlled heat pipe design will be explored and evaluated.

5.6 Outdoor Testing

Thermacore installed two 4' x 4' TC-100 panels for testing of

heat pipes as seen in Figure 1.6. The lower panel has ten copper-water heat pipes, the upper, ten steel-ethanol heat pipes. The copper has a total of 300 heat pipe days and the steel has a total of heat pipe days of testing.

The copper has been stagnation tested to 500 F for three days, the steel and copper to 400 F for two days. To date, no significant changes have occurred.

Calorimetry at an insolation rate of approximately 900 W/m^2 has produced 1600 watts at 75 C use temperature.

5.7 Cost Analysis

To date a full cost analysis has not been completed as the design is not finalized. However, cost comparison of several pipe materials is seen in Table 5.6.

At the present time the cost of production, exclusive of profit, for a four foot long serpentine without the copper fins and manifold connections in quantities of 100,000 per year is \$2.30. Using the material prices of 4.1¢, 6¢ and 11¢, a preliminary cost analysis was done for a heat pipe. The cost factors are seen in Table 5.7. The costs were \$1.71, \$1.81 and \$2.10 for the three materials. These estimated costs are for 100,000 heat pipes per year after one year's production, which is similar to the \$2.30 cost for the serpentine.

From this initial cost analysis the heat pipe appears to be cost competitive with the serpentine heat removal system. A full cost analysis including manifolds and connection costs will be done after the heat pipe and manifold design are finalized.

TABLE 5.6. COST OF TUBING

5/16" OD x 4' MULTIPLE LENGTHS

MATERIAL	WALL	VENDER	COST/FT	MATERIAL COST/LB
1010 St	.028	Republic Steel	.179¢	.174¢
CDA 122	.030	Reading Industries	.137¢	.64 ¢
18 SR	.022	Bundy Tube	.11 ¢	.77 ¢
NP 254	.022	Bundy Tube	.06 ¢	.42 ¢
1010 St	.022	Bundy Tube	.059¢	.174¢
1010 St	.028	Edgcomb Steel	.041¢	.174¢
1010 St	.022	Edgcomb Steel	.013¢	.174¢

(copper brazed laminated tubing)

TABLE 5.7. PRELIMINARY COST ESTIMATES

	1010 4.1¢	NP 254 6.0¢	18 SR 11.0¢
Material Cost	16.40	24.00	44.00
Shipping 1% of Material	.16	.24	.44
Fluid - Maximum 4¢/heat pipe	<u>4.00</u>	<u>4.00</u>	<u>4.00</u>
Total Material Cost	20.56	28.24	48.44
Labor: 6 minutes of \$4.00/hr	40.00	40.00	40.00
Overhead: 150%	<u>60.00</u>	<u>60.00</u>	<u>60.00</u>
Total Material, Labor, Overhead	120.56	128.24	148.44
Waste: 5%	<u>6.03</u>	<u>6.41</u>	<u>7.42</u>
Total	126.59	134.65	155.86
G & A: 10%	<u>12.66</u>	<u>13.47</u>	<u>15.59</u>
Total	139.25	148.12	171.45
Warranty: 5%	<u>6.96</u>	<u>7.41</u>	<u>8.57</u>
Total	146.21	155.53	180.02
Warehousing: 5%	<u>7.31</u>	<u>7.78</u>	<u>9.00</u>
Total	153.52	163.31	189.02
Profit: 10%	<u>15.35</u>	<u>16.33</u>	<u>18.90</u>
Total	168.87	179.64	207.92
Shipping: 10%	<u>1.69</u>	<u>1.80</u>	<u>2.08</u>
Total	170.56	181.44	210.00
\$ Value	\$1.71	\$1.81	\$2.10

6. CONCLUSIONS AND RECOMMENDATIONS

The program is approximately one-third completed. Results have been encouraging. Several heat pipe fluid-vessel combinations have been identified. Initial cost analysis look promising.

Water and ethanol heat pipes have been shown to operate within the power, temperature and overall thermal conductance range as defined by the system analysis. Full stagnation and water freeze resistance tests are currently being performed.

The system analysis has quantified that the presence of the heat pipe and its thermal resistance will result in less than a 3% loss in annual collected energy. The benefits the heat pipe brings to the system are harder to quantify and will be explored analytically and in the final systems verification tests.

The initial cost analysis is encouraging and will be expanded to include the manifold fitting and fins at the conclusion of the program. Included in the analysis will be inputs from the production process to be developed during the next six months.

APPENDIX 1

HEAT PIPES AND LOW COST WICKS

The heat pipe is a sealed heat transfer element. It makes use of vapor heat transfer to carry heat at low temperature loss from an input area where a fluid is evaporated to an output area where the vapor is condensed, giving up its heat of vaporization. As originally conceived,^{4,5} the heat pipe used capillary action in a porous wick to return the condensed liquid to be re-evaporated. When gravity is available and the heat source is below the heat sink, the condensate will return without a wick, as in a single pipe steam heating system for a building. In general, wicks are expensive to form and to insert into the heat pipe. Therefore, whenever possible in terrestrial, gravity-aided applications, wicks are minimized or eliminated. A general discussion of heat pipe operation and limits is given by Eastman⁶. See references 7-11 for mathematical treatment of the physical principles and tabulation of test results. Undegraded operating life in excess of 70,000 hours has been demonstrated at temperatures of interest for solar collectors¹² (230 F). Heat pipe life experience is summarized in references 11, 13 and 14.

Heat pipe performance is predictable, but complex mathematically. The calculation of a single data point takes one-half an engineering man-day. To avoid the excessive labor Thermacore has written computer programs for the purpose.

Heat pipes are compatible with solar collection requirements in terms of power, operating temperature and physical size. They have been considered as elements in at least four solar collector applications: power towers, cylindrical parabolic collectors¹⁵ and flat plates^{16,17} in addition to the tubular collectors under test in the program.

APPENDIX 2

MATERIALS COMPATIBILITY

In order to ascertain any possible material compatibility and fluid life problems a preliminary materials compatibility analysis was performed. The fluid and materials considered are as follows:

<u>Fluid</u>	<u>Heat Pipe</u>	<u>Fittings</u>	<u>Manifold Material</u>
Prestone II & Water	Copper	Brass	Copper
	Stainless Steel	Copper	Stainless Steel
	Low Carbon Steel	Stainless Steel	Low Carbon Steel
		Low Carbon Steel	

As a result of analysis the following conclusions/recommendations were formulated:

1. Based on in-house fluid life test data it may be concluded that the metal/alloy combinations evaluated are satisfactory and compatible for long term exposure at 290 F to a Prestone II-distilled water solution. While there is a 35 F difference between the completed loop experiment and the intended heat pipe application, it is believed that the Prestone II solution will also be compatible with the same metals and combinations at 325 F. However, in all probability the rate of depletion of the corrosion inhibitor system in Prestone II may be accelerated due to higher temperature effects. Stability of a Prestone II solution and its effects on the same combinations of metals and alloys are now being investigated in a corrosion loop operating at 325 F. About 65 hours of operation have been accumulated at this time. The first fluid sample and specimens of tubing, brazed joints, and fittings will be examined after 1100 hours have been accumulated.
2. The following metals/alloys and combinations of each are considered compatible from the standpoint of minimizing corrosive effects within the limitations detailed in the succeeding paragraphs 3 and 7:

CDA 122 Copper Tubing
Brass 45° Flare Fittings
SS or Brass Swagelok Fittings
Black Iron Piping
Carbon Steel

3. Threaded fittings of dissimilar materials such as copper-steel,

copper-stainless steel, stainless steel - carbon steel or black iron should be coated with a dielectric type sealant, e.g., Loctite Pipe Sealant with Teflon, to minimize galvanic effects.

4. Joining of copper tubing to itself or to brass, or to stainless steel can be accomplished by torch brazing with Fusion's STL-1000-650 mix following a process similar to the one described in GE process specification P9G6 (Torch Silver Brazing of Dissimilar Materials).
5. Oxidation and scaling effects on copper tubing exterior surfaces may become significant for extended exposure periods in air above about 680 F. Experiments performed have indicated that these effects can be minimized by application of an electroless or electrolytic nickel plating 0.2-0.4 mil thick onto the exterior surface of copper tubing.
6. With respect to the use of water in a copper heat pipe, certain perhaps widely accepted fundamentals may be worth repeating. Generally speaking, copper is not appreciably corroded in the absence of oxygen. Copper is less prone to corroding in hard water than in very soft water characterized by low permanent hardness in combination with high dissolved carbon dioxide and oxygen contents. The rate of corrosion with soft water may become excessive. The carbonic acid formed, even though a weak acid, prevents formation of the protective cuprous oxide and cupric hydroxide films ordinarily produced. These films significantly retard the rate of corrosion of copper. On this basis, the least aggressive medium may well be deaerated, distilled, demineralized water.
7. In the passivated state, stainless steels show a noble galvanic potential. If connected with an equal or greater area than the more active metals such as carbon steel or cast iron, accelerated corrosion will occur on the latter metals. However, stainless steel may be coupled with these metals if its area is significantly less than that of the active metal, e.g., a stainless steel fitting connected to a steel pipe. The compatibility of this combination with a Prestone II solution at 290 F was satisfactorily demonstrated in the recently completed corrosion loop test.

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