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MASTER

THERMACORE, INC.

LEOLA, PENNSYLVANIA

COST-EFFECTIVE SOLAR COLLECTORS

USING

HEAT PIPES

INTERIM PROGRESS REPORT

#2

APRIL 1978 - SEPTEMBER 1978

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DEPARTMENT OF ENERGY

CONTRACT

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There is no objection from the patent point of view to the publication or dissemination of the document(s) referred to in this letter.

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 heat pipe collector system design was re-evaluated as a "new" system, as compared to previous evaluation where the heat pipe was an interchangeable component in the standard TC-101 system. Collector hardware components were finalized, including production costs. Heat pipe fluid-vessel testing continued indoors and outdoors in the solar panel.

A prototype production processing station has been designed which shows that the labor content of processing individual heat pipes can be reduced to 15 seconds with a total cost of \$1.50.

1. INTRODUCTION

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

During the reporting period, work concentrated on the re-evaluation of system implications of a heat pipe collector, detail design of the manifold, its cover, collector frame and reflector assembly, extended fluid-vessel evaluation, additional cost details, the design of a prototype production processing station, and establishing the test program by which the heat pipe collector system will be evaluated.

The Program Schedule and Task Flow Diagram are seen in Figures 1.1 and 1.2.

Fluid-vessel evaluation and testing will continue including analysis. Design verification of the heat pipe collector will be carried out in parallel with the testing of a standard TC-101 Serpentine Collector for direct comparison of the daily collection efficiency.

Figure 1.1

PROGRAM SCHEDULE

Figure 1.2

TASK FLOW DIAGRAM

1.1 Background

Prior work analyzed pertinent collector systems and established heat pipe design goals for optimum system performance. From this analysis, separate components were designed and fabricated. The search for an improved heat pipe fluid was initiated. The component design was subjected to careful scrutiny from the system integration and producibility viewpoints. Heat pipes, collector-to-heat pipe and heat pipe-to-pumped-fluid loop interfaces were tested in the laboratory and outdoor collector facility. Initial cost estimates were done for several heat pipe fluid-vessel combinations.

1.2 Summary

The heat pipe collector system re-evaluation was based on emphasizing the advantages of the heat pipe in its own "system" rather than considering the heat pipe as an interchangeable component in the standard TC-101 system.

The heat pipe design has eight collectors in a series connected bank which is then paralleled with other banks in the system. The standard TC-101 has all collectors in parallel. The "collector" used for comparison of the two systems is the necessary material hardware, and associated labor to connect eight collectors in series or parallel divided by eight. This "collector" does not include the actual mounting of the collector which will vary depending on the application.

Using this "collector" definition, the re-evaluation showed a 1-3% calculated loss in daily collection efficiency,

and an estimated cost saving of 11-14% for the heat pipe collector. These values translate into a 9-13% reduction in cost per unit energy delivered per heat pipe collector as compared to the standard TC-101.

The potential for additional cost reduction per energy unit may be realized by utilizing east-west orientation of the collector. This orientation may allow for the use of potable water as the collector fluid in a drainable system with a net increase in daily collection efficiency and a net decrease in system costs, through the elimination of intermediate heat exchangers. Such a drainable system is not possible without the heat pipes.

Heat pipe cost estimates of \$1.50 each used to generate the cost per unit of delivered energy were based on the use of production processing equipment. This unit has forty individual stations and can produce a heat pipe every 15 seconds. A laboratory prototype processor has been designed which is capable of reducing critical steps in processing a heat pipe from 30 minutes to 15 seconds.

An additional 23 heat pipes were fabricated and tested. The total now stands at 75 heat pipes. An additional 50,000 hours of testing were accumulated. The total accumulated test hours were 67,000. To date, no fluid-vessel combination holds a clear cut edge as being the "system".

There are four combinations still under investigation. They are copper-water, aluminum bearing steel-water, steel-ethanol and steel-trimethyl borate. The heat pipes which have

been tested in the TC-100 collector panels at Thermacore are in the process of being analyzed.

A test plan was written from which the design of the heat pipe collector can be verified. The specific test objectives are to compare performance of the heat pipe collector and standard TC-101 collector, validate analytical predictions, and obtain operational experience with the heat pipe collector.

2. HARDWARE DESIGN AND COST

A preliminary design was defined for a solar collector utilizing a 0.312" O.D. heat pipe as a means of absorbing solar energy in a collector instead of the .25" O.D. serpentine tube design of the existing TC-100 and TC-101 solar collector units. The basic heat pipe collector consists of eight heat pipes and vacuum tube shrouds coupled to TC-101 type parabolic reflectors and has three major subassemblies; the manifold assembly and cover, heat pipe assembly, and frame and reflector assembly.

2.1 Manifold Assembly and Cover

The manifold assembly as seen in Figure 2.1 serves as the heat removal component for the heat pipe condenser as well as a manifold for coolant circulation. In the development phase, heat pipes are inserted into the manifold through a modified compression fitting which seals the heat pipe/manifold interface and allows quick individual removal of heat pipes. When system reliability has been established, the heat pipes will be brazed into the manifold, thus eliminating the compression fitting at a cost savings.

Construction of the manifold body is accomplished by die forming fluid channels and fitting sockets from sheet metal into identical halves. These halves are joined together along with the compression fittings by brazing. To minimize heat loss, excess material is removed from the manifold. The

FOLLOWING		MANIFOLD ASSY.	
DATE		HEAT PIPE COLLECTOR	
+ .03		FIRST MADE FOR	
QTY	PART NO	NAME	DESCRIPTION
1	1	ASSY.	
2	2	MANIFOLD HALF	4415 HOT ROLLED LOW CARBON STEEL PICKLED & OILED 12.9 GRADE
2	3	TUBE	1.583" OD 16.6 A 122-02 1/2 WALL COPPER TUBING AL 2-M-280 TYPE DHR CONDITION REGIONAL
2	4	NUT	273AG040PI
ALT	4	NUT	273AG045PI
8	5	FITTING	.50 OD. TO 31.61 BRASS REDUCER COMPRESSION FITTING
AR	6	BRAZE PASTE	FUSION INC. STL-1000-GSC
AR	7	PAINT	BLACK HIGH TEMP. ENAMEL

- NOTES:
1. ALL INSIDE BEND RADII TO BE .06 UNLESS OTHERWISE NOTED.
 2. REMOVE BURRS & BREAK ANY SHARP EDGES.
 3. BRAZE PER G.E. SPEC P9G3. RINSE JOINT THOROUGHLY INSIDE AND OUT WITH HOT WATER TO REMOVE RESIDUAL FLUX.
 4. EACH BRAZE JOINT TO BE TESTED FOR LEAKAGE. LEAK RATE NOT TO EXCEED 1×10^{-4} SCC/SEC. HELIUM (OR EQUIV STANDARD OF MEASURE).
 5. FINISH: PAINT ITEMS 2 AFTER BRAZING WITH BLACK HIGH TEMP. ENAMEL.
 6. FLARE TUBE END WITH 45° FLARE. THERE SHALL BE NO VISIBLE CRACKS OR PITS AFTER FLARING.

Figure 2.1

MANIFOLD ASSEMBLY

REVISIONS		PRINTS TO	
SP		919E158	
HILA PA	LOCATION	COUNT ON SHEET	BY NO. 1

manifold body is 415 low carbon, hot-rolled steel, pickled and oiled. This steel was chosen as it is economical, soft and readily formed and is supplied with the mill oxide removed, making it suitable for good paint adhesion.

Insulation around the manifold consists of a 0.25" fiberglass blanket covered with a 1.00" layer of polyurethane foam. The fiberglass acts as a thermal buffer to protect the foam from direct contact with the high temperature of the manifold. The orientation of the manifold in the collector assembly is seen in Figure 2.2.

Since the heat pipe manifold has a much lower pressure drop than the TC-101 type serpentine tubing, a conventional header assembly is not necessary. Accordingly, heat pipe collectors are grouped in series, thereby reducing the cost and heat loss associated with the conventional header. The collector fluid leads are 0.25" O.D. copper tube with 45° flare fittings and serve as collector ties.

The manifold cover as seen in Figure 2.2 is similar to that used on TC-101 units. The cover is slightly larger to accommodate the heat pipe manifold. Attachment hardware, as on other maintainable parts of the collector, is accessible from the top. The manifold cover bridges the reflector tray to the stationary frame. Accordingly, it is sized to transmit the reflector wind loading to the collector mount.

2.2 Heat Pipe Assembly

The heat pipes have copper fins similar to those on the

Figure 2.2
COLLECTOR ASSEMBLY

serpentine tubing of conventional TC-101 collectors as seen in Figure 2.3. However, the heat pipe is located at the bottom of the shroud, where the highest heat flux is seen. Accordingly, to bring the heat pipe from its position in the shroud to the correct position in the manifold, necessitates an offset in the heat pipe. This offset also facilitates fluid flow from the condenser to the evaporator section in the heat pipe.

2.3 Frame and Reflector Assembly

The frame design of the heat pipe collector has been substantially simplified over that of the standard TC-101. The frame has been reduced to Z-section beams which run at right angles to the troughs at the top and bottom of the unit. Wind loading on the reflector is transferred directly to the Z-section at the lower end of the collector and through the manifold cover to the Z-section at the upper end. The assembly drawing which demonstrates this frame concept is seen in Figure 2.2. Note that reflector length is reduced to include only the active length of each shroud. The domed end of each shroud is held in place by a retainer wire. When the retainer is removed, shroud removal is unimpeded.

2.4 Hardware Cost

A meaningful way to present heat pipe collector cost is by comparing it with the present cost of the TC-101 U-tube

Figure 2.3

HEAT PIPE-FIN ASSEMBLY

serpentine collector. However, care must be taken to assure that the comparison only highlights those differences due to introduction of the heat pipe as the heat extraction component. For example, cost savings resulting from the frame redesign is not credited to the heat pipe collector since such a frame redesign could also be incorporated into the serpentine collector, even though the design change was prompted by heat pipe considerations.

Table 2.1 shows the selling price for the heat pipe and standard TC-101 U-tube serpentine collectors. Prices are FOB the manufacturing plant at the current production rate of 10,000 collectors per year.

Table 2.2 shows the cost breakdown for the heat extraction components of the heat pipe collector, and frame/bulkhead redesign used in Table 2.1.

From Table 2.1 it is seen that the heat pipe collector offers a cost reduction of 11% over the standard TC-101 collector. As discussed in section 2.1, an additional cost saving can be accrued by brazing the heat pipes into the manifold which at the selling price level amounts to \$5.99. The resulting heat pipe collector selling price is \$307. Accordingly, the heat pipe collector offers a cost reduction of 14% over standard TC-101 collector.

Table 2.1 also lists the hardware cost of the collector frame/bulkhead assembly for the heat pipe collector. Although the frame is approximately 10% larger than the standard TC-101 frame, its costs are approximately 10% less. However, as noted

TABLE 2.1

COLLECTOR SELLING PRICE

	<u>Standard TC Collector</u>	<u>Heat Pipe Collector</u>
Collector Frame & Reflector Less Installation & Mounting Hardware Plus Header and/or Interconnecting Ties Including Their Installation & Insulation Plus Necessary Heat Removal Components & Shrouds.	\$358 ⁱ	\$318 ⁱⁱ

Note: Prices based on current production rate of 10,000 Collectors per year

- i) Includes packaging, handling, warehousing, warranty, G&A and profit on header kit, which if purchased locally at time of installation may reduce total by 5%.
- ii) Based on heat pipe selling cost to collector manufacture cost of \$1.50.

TABLE 2.2

COST OF HEAT EXTRACTION COMPONENTS
OF HEAT PIPE COLLECTOR

Heat Pipes 8 @ \$1.50 + 20% Mark-Up	\$14.42
Attachment Fittings	6.73
Steel Manifold	3.70
Flare Nuts (2)	.24
Pigtails (2) - Collector Ties	.25
Fins (16)	10.70
Fin Attachment	4.41
	<hr/>
	\$40.45

FRAME-BULKHEAD COST

Preliminary Cost Estimate

Clip Hardware	\$.50
Side Members	1.36
Bulkhead Cover	1.32
Bulkhead Bottom	1.10
Insulation	.34
Header Panel	.60
Misc. Screws, Rivets	1.70
Clips	2.40
Frame/Tray Assembly	9.60
	<hr/>
	\$18.92

Note: Costs are through G&A at an assumed production rate of 10,000 collectors per year. Heat pipe cost of \$1.50 is selling price to collector manufacture at a rate of 100,000 per year.

above, these cost savings could be utilized by both collectors. Accordingly, the figures in Table 2.1 reflect this fact.

Header kits are not required for the heat pipe collectors since their low pressure drop allows series hook-up. The definition of the "collector" used in this cost analysis is the necessary material, hardware and associated labor to connect eight collectors in series or parallel divided by eight. This "collector" does not include the actual mounting of the collector which will vary depending on the application. The heat pipe selling cost is set at \$1.50 and is detailed in Appendix A.

The result of this costing exercise is that heat pipes have the potential to reduce collector costs if their low coolant pressure drop is properly exploited.

3. HEAT PIPE COLLECTOR SYSTEM IMPLICATIONS

The major mechanical components of a collector system are seen in Figure 3.1. T_i and T_o is the inlet and outlet temperatures of the collector field respectively. T_{av} is the average fluid temperature in the collector field. T_{abs} is the average absorber temperature, and T_s is the thermal storage tank temperature. Field piping is shown schematically and is not meant to indicate either series or parallel collector plumbing. This diagram contains the major features which come into play when the heat pipe collector and its system implication, are analyzed. The important application temperature of Figure 3.1 is the thermal storage tank temperature, T_s . Systems must be compared on the efficiency with which they maintain this temperature. On the other hand, T_{abs} is the temperature at which collectors lose heat to the environment. For a given T_s , T_{av} is a function of the fluid flow rate and solar insolation. Since heat pipe collectors and serpentine collectors operate at different flow rates, their average fluid temperatures are different for the same T_s . In addition, the two collector types have different thermal resistances between absorber and fluid. The heat pipe has a slightly higher absorber temperature for a given average fluid temperature resulting in a higher heat loss for the same T_{av} . However, the heat pipe collectors are connected in series which reduces field piping and the associated heat loss. These several effects are discussed in detail in the following sections.

Figure 3.1

SYSTEM SCHEMATIC

3.1 Heat Pipe Fluid ΔT

The measured temperature difference between the heat pipe evaporator and collector working fluid at a heat pipe heat flux of 94 watts (321 BTU/HR) for several collector fluids and temperatures was reported in the first Interim Report. Additional data for 60% water - 40% Prestone II are given in Figure 3.2. These experimentally determined ΔT 's, or thermal resistances, are a sum of six resistances: The heat pipe evaporator wall, the internal evaporation coefficient, the heat pipe vapor ΔT , the condensing coefficient, the condenser wall and the condenser wall to bulk fluid film coefficient. The first three resistances are measurable as a group and are on the order of 1-3 F. The second three resistances, or temperature differences are also measurable as a group and comprise the remaining ΔT 's of Figure 3.2. Some measure of the magnitudes of the internal and external condenser coefficients is evident by the relative flatness of the curves of Figure 3.2 and indicate that the internal resistance of the condenser is dominant, since the film resistance outside the condenser wall is roughly inversely proportional to coolant flow rate.

It is important to characterize the ΔT between heat pipe and collector fluid since this thermal resistance increases the absorber temperature for a given collector fluid temperature. All heat extraction components involve some thermal resistance. Accordingly, the thermal resistance of the heat pipe should be evaluated in light of its effect

Figure 3.2

TEMPERATURE DIFFERENCE BETWEEN HEAT PIPE EVAPORATOR AND COLLECTOR FLUID

on the total thermal resistance. Other heat extraction components which have significant thermal resistances are, the heat transfer fin, the air gap between the shroud and fin, and the junction of fin to heat pipe. When the fins are joined directly to a tube carrying collector fluid (the heat extraction technique presently used for GE production collectors), these latter resistances are still present. Figure 3.3 shows the total thermal resistance between absorber and pumped fluid for the heat pipe and standard TC-101 collectors.

The effect of the increased thermal resistance on collector performance can be directly evaluated. From test data of efficiency versus absorber temperature, the absorber temperature is increased by the thermal resistance ΔT and the new efficiency is read from the performance curve. A second method of estimating performance uses a computer model of the GE collector. This model includes the effect of the total thermal resistance, from which the efficiency is calculated for various resistances.

Both techniques were used in assessing the effect of the heat pipe thermal resistance on collector performance. Results from the latter technique are shown in Figure 3.4. Here, actual efficiency is expressed as a fraction of ideal efficiency, no thermal resistance, over a range of actual resistances and fluid temperatures.

The serpentine resistance is .45 F-hr-ft/BTU while the heat pipe resistance ranges from .59 to .72 F-hr-ft/BTU at a flow rate of .7 gpm in the temperature range of interest. The

Figure 3.3

TOTAL ABSORBER WORKING FLUID THERMAL RESISTANCE
VS
FLOW RATE

Figure 3.4

EFFECT OF FLUID-ABSORBER THERMAL RESISTANCE

theoretical effect on efficiency of this increased resistance can be obtained from Figure 3.4.

The effect of the heat pipe's ΔT on efficiency was also assessed using actual test data of efficiency versus temperature. These results are presented in Section 3.4 where a net system efficiency comparison is made.

3.2 Heat Pipe Collector ΔP

The pressure drop across a heat pipe collector depends on the manifold design. The present design includes annuli, where collector coolant flows over the exterior condenser surface, 90° bends and straight pipe runs which carry coolant to successive condensers. Flow in short annuli and the associated transitions to open pipe flow is difficult to model. The technique used to model this flow is at best 50% accurate in calculating the pressure drop. The results are shown in Figure 3.5. Also shown in this figure, is the pressure drop across an eight shroud serpentine collector as a function of flow rate, and preliminary test data for a heat pipe collector. The most striking result is that the pressure drop across the heat pipe collector is less than that of the serpentine collector by an order of magnitude.

The small ΔP across a heat pipe collector manifold implies that neither the optimum flow rate nor the plumbing arrangement of the heat pipe need match that of the serpentine collector. In fact, if this is attempted, flow restrictors are required in each collector to artificially increase the

Figure 3.5

PRESSURE DROP PER COLLECTOR - 60% WATER, 40% PRESTONE

pressure drop. (Collectors installed in parallel, such as the standard TC-101, require ΔP 's which are large compared to the header pipe ΔP which feeds them). Rather than restricting the heat pipe collectors to a serpentine plumbing system, the small manifold ΔP of the heat pipe collectors was exploited by installing them in series and operating at a flow rate on the order of .7 gpm. Under these conditions, eight collectors can be connected in series before the total pressure drop equals that through a single serpentine collector at its design flow rate of .25 gpm, (5.6 psi). Series installation eliminates costly header piping and the associated heat loss from such piping. At .7 gpm, the heat pipe thermal resistance has begun to level out. While not necessarily optimum, this flow rate is a reasonable choice for initial calculations. Arguments given below show that the optimum flow will depend on the specifics of a given installation.

3.3 Implications of Series Hook-up On System Installation and Performance

Two rows of TC-101 collectors as they are presently installed in the field are illustrated in Figure 3.6. The collectors are installed in parallel. The flow rate for each row is $8 \times .25$ gpm or 2 gpm. Figure 3.7 shows the same number of heat pipe collectors installed in series. The flow rate for each row is .7 gpm which is 35% of the

Figure 3.6

EXISTING PARALLEL HOOK-UP

Figure 3.2

SERIES HOOK-UP

flow rate of the serpentine system. The elimination of the headers reduces material cost, installation cost, and the energy lost by field piping. A further advantage of series connected collectors is that drain down potential is increased. It is easier to design a drain down system if the collector fluid is contained only in the manifolds rather than in the header pipes and U-tube serpentine. The fluid volume is substantially reduced with the heat pipe collector, which reduces the thermal inertia of the collector and the time required to heat the system to operating temperature.

The reduction in field piping heat loss depends on the heat lost from the field headers, the serpentine box and the proposed manifold. To date, these heat losses have not been experimentally measured. Accordingly, standard heat transfer equations were used in a consistent fashion to assess the benefit gained by eliminating the headers. The results of this analysis are seen in Figure 3.8. The upper curve shows the heat lost from the headers required for one serpentine collector plus the heat lost from the serpentine box of one collector. The calculation assumed two and one half inches of insulation surrounded all piping runs. The serpentine box heat loss is approximately 25% of the total loss. The lower curve shows the heat lost from the manifold of one heat pipe collector, again with 2-2½ inches of insulation. Details of these calculations appear in Reference 3. The difference between the two curves is the energy savings per collector which occurs when headers and serpentine box are replaced by

Figure 3.8

MANIFOLD AND HEADER PIPE LOSSES VS INLET TEMPERATURE

the heat pipe manifold. It is emphasized that these calculations are approximate. The serpentine box is especially difficult to analyze since convective air currents may play some role in the heat loss.

The final system consideration involves the flow rate through the collector field. The design flow rate of the standard U-tube serpentine collector is .25 gpm. Since all collectors are installed in parallel, the total flow rate is .25 gpm times the number of collectors. For a heat pipe collector system, a design flow rate must be chosen. For eight heat pipe collectors placed in series with a flow rate of 2 gpm through each collector, the net ΔP across the collector field is approximately 16 psi. This is found by extrapolating the lower curve of Figure 3.5 to 2 gpm, obtaining 2 psi pressure drop per collector, and multiplying by eight, the number of collectors in series. This is three times the ΔP of the serpentine system. Therefore, the pump parasitic loss is increased since the pumping power is proportional to the pressure drop times the flow rate. On the other hand, the temperature rise across the collector field will closely match that of the serpentine system since this ΔT is given by $\Delta T = Qn/mC_p$, where Q is solar insolation, n is efficiency, m is flow rate, and C_p is specific heat. As the flow rate is decreased, the ΔP across the collector field is decreased and the pump power requirement drops. At the same time, the temperature rise across the field increases. At .7 gpm, the ΔP across the field matches that across the serpentine system. The net flow

rate is, however, $1/3$ less. Thus, the pumping power is less by $1/3$ and the temperature rise is greater by a factor of 3. The former effect decreases the pump parasitic loss. The latter effect increases the average collector temperature and negatively affects efficiency as discussed below.

As mentioned earlier, systems must be compared on their ability to maintain a given thermal storage tank temperature, T_s . As the flow rate is decreased, $T_o - T_i$ increases and $T_{av} - T_i$ increases. The higher the average temperature, the less efficient the collector. Thus, to assess the effect of flow rate on efficiency, the relationship between T_i and T_s must be known. However, this relationship depends on the heat exchanger design. In general, the larger $T_o - T_i$, the smaller the heat exchanger area required. This effect decreases as the heat exchanger area increases. Since the heat exchanger cost is small compared to the total system cost, the increased $T_o - T_i$ at decreased flow is a small perturbation on the heat exchanger performance and cost.

To make this analysis traceable, it was assumed that the heat exchanger was ideal. This implies $T_i = T_s$, thus $T_{av} - T_s$ is inversely proportional to the flow rate. The effect of flow rate on system performance can be predicted as long as the efficiency versus average temperature of the collector (performance curve) is known. Such calculations are presented in the next section.

The question of the optimum flow rate for the heat pipe collector remains open. At high flow rates the temperature

differences between absorber and collector fluid and collector fluid and storage tank are minimized. Both are beneficial effects. On the other hand, pump parasitics are increased at high flow rates. As the flow rate decreases, both ΔT 's are increased which has a negative effect on efficiency, but the field ΔP , pipe run ΔP 's, and pumping power decrease, which is a beneficial effect on efficiency. Accordingly, the flow rate must be chosen with a given installation in mind. In the analysis of net efficiency which follows a heat pipe collector flow rate of .7 gpm was used. This flow rate is well past the knee of the absorber - fluid ΔT curve of Figure 3.2. It results in 1/3 the pump parasitic loss of the U-tube serpentine system and three times the ΔT through the collector field.

3.4 Net Efficiency Comparison

Table 3.1 shows the effect of the field ΔT on the efficiency comparison of the serpentine and heat pipe collectors. The total flow rate through the heat pipe system is 1/3 that of the serpentine system as discussed above. The efficiency difference is for a solar insolation rate of 300 BTU/hr-ft². Figure 3.9 shows the experimental performance curve on which the calculations of Table 3.1 are based. In Table 3.2, the effect of the heat pipe-fluid ΔT is added to results of Table 3.1. Thus, Table 3.1 includes both the negative effects of the heat pipe: greater thermal resistance and greater fluid ΔT resulting from reduced field flow rate. Table 3.3 shows

TABLE 3.1

EFFECT OF FIELD ΔT ON COLLECTOR PERFORMANCE

TES TANK TEMP = INLET TEMP	SERPENTINE COLLECTOR			HEAT PIPE COLLECTOR			$\eta_s - \eta_H$
	ΔT ACROSS FIELD	T_{AVG}	η_s	ΔT ACROSS FIELD	T_{AVG}	η_H	
100	22.4	111.2	58.0%	64.0	132.0	57.6%	0.4
175	21.0	185.5	55.2%	59.2	204.6	54.2%	1.0
280	18.0	289.0	48.2%	52.8	306.4	46.9%	1.3

INSULATION = 300 BTU/HR-FT²

HEAT PIPE FIELD FLOW RATE = 1/3 SERPENTINE FIELD FLOW RATE

TABLE 3.2

EFFECT OF CONDENSER-COOLANT ΔT ON COLLECTOR PERFORMANCE

(TABLE 3.2 INCLUDES EFFECTS SHOWN IN TABLE 3.1)

TES TANK TEMP = T_{IN} TEMP	SERPENTINE COLLECTOR			HEAT PIPE COLLECTOR			$\eta_s - \eta_H$
	ΔT CONDENSER - COOLANT	T_{AVG}	η_s	ΔT FLUID - CONDENSER	T_{AVG}	H	
100	0	111.2	58.0%	25.9	157.9	56.6%	1.4
175	0	185.5	55.2%	19.8	224.4	53.0%	2.2
280	0	289.0	48.2%	14.8	321.2	45.2%	3.0

INSULATION = 300 BTU/HR-FT²

Figure 3.9

PERFORMANCE CURVE

TABLE 3.3

EFFECT OF PIPING LOSSES ON COLLECTOR PERFORMANCE

TEST TANK TEMP = T _{IN} TEMP	SERPENTINE	HEAT PIPE	S - H PIPING LOSSES ONLY
	PIPE LOSSES IN BTU/HR	PIPE LOSSES IN BTU/HR	
100	21.30	13.64	- .17%
175	75.00	47.50	- .60%
280	149.00	95.00	-1.20%

INSULATION = 300 BTU/HR-FT²

(DATA TAKEN FROM FIGURE 3.8)

the advantage which results from the reduced field piping required with the heat pipe collector. These results are based on Figure 3.8. To within the accuracy with which piping losses and system performance can be predicted, Tables 3.2 and 3.3 indicate that the system performance of heat pipe and serpentine collectors are equivalent.

4. VERIFICATION TEST PROGRAM

The results of Section 3 indicate that two counter-balancing effects must be considered when a heat pipe replaces the U-tube serpentine as the heat extraction component. On one hand, the heat pipe increases the absorber temperature for a given fluid inlet temperature. On the other hand, the low pressure drop of the heat pipe manifold allows series collector installation and thus reduces field piping, thermal losses and pump parasitics. These effects roughly balance when considered in the context of the accuracy with which performance and piping losses can be predicted. A test program is now required which will isolate the several effects analyzed in Section 3 so that a reliable system efficiency comparison can be made between the heat pipe and serpentine designs. The specific test objectives are to:

1. Compare performance of heat pipe collector and serpentine collector
2. Validate analytical predictions (ΔT 's, ΔP 's, η)
3. Obtain operational experience with heat pipe collector

4.1 Test Units

The proposed test will require both a standard TC-101 and a heat pipe TC-101. The heat pipe TC-101 will be a retrofitted production TC-101. The manifold will consist of copper tubing, tees and required fittings. The flow passages

will match those of the proposed die-stamped manifold. Required hardware for both units has been received.

4.2 Test Loop and Instrumentation

Figure 4.1 displays the test loop and associated instrumentation which will be used to compare the heat pipe and serpentine collectors. Useful energy gain, Q , will be calculated with

$$Q = m C_p \Delta T$$

where m is the flow rate, C_p is the specific heat, and ΔT is taken across the collector alone. For the serpentine unit the ΔT will be taken across the collector and associated header piping. Thus, the efficiency comparison will account for the reduced header piping heat loss associated with the heat pipe design. Both collectors will be installed in series to insure an accurate measurement of the flow rate. The flow rate will be varied from .25 to 1.0 gpm. At the higher flow rates, the serpentine collector may have to be bypassed because of its relatively large pressure drop. Measurements will also be made of the temperature difference between fin and coolant for both collectors, for measurement of the total thermal resistance of each system.

Figure 4.1

TEST LOOP AND INSTRUMENTATION

5. TEST RESULTS

During the reporting period an additional 23 heat pipes were fabricated and tested. The total now stands at 75 heat pipes. An additional 50,000 hours of testing were accumulated. The total test hours are 67,000. Table 5.1 shows the heat pipes of significance. To date, no fluid-vessel combination holds a clear cut edge as being the "system."

There are four combinations still under investigation. They are copper-water, aluminum bearing steel-water, steel-ethanol and steel-trimethyl borate.

5.1 Copper-water

As seen in Table 5.1, the copper-water system is the most desirable on the basis of known life times and low ΔT . However, the two extremes of the environmental temperature range -40 C and 410 C still represent problems which appear not to have a simultaneous solution.

To date, a copper-water heat pipe has 144 freeze-thaw cycles. This heat pipe has a fiberglass wick in it to relieve the freezing stresses. However, the use of fiberglass is limited to an operational limit of 165 C above which the samples tested to date have dissolved in water. Accordingly, a stagnation temperature control is required to use the copper-water system. The added cost of the control mechanism prices the copper-water heat pipe out of consideration.

Attempts to use copper mesh in place of the fiberglass

TABLE 5.1

LIFE TEST DATA

PIPE	MATERIAL	FLUID	TEMP-°C	HOURS	T - °C			
					0-10	10-20	20-30	30
S-3	1010 ST	H ₂ O	175-225	8500				X
CU-1	CDA-101	H ₂ O	150	7900	X			
NP454-1	NP454	H ₂ O	175-200	5400		X		
NP454-3	NP454	H ₂ O	150	3000		X		
18SR-2	18SR	H ₂ O	200-225	2600	X			
SR-3	18SR	H ₂ O	175	3000	X			
18SR-4	18SR	H ₂ O	225	2600	X			
ATS-13	1010 ST	(CH ₃ O) ₃ B	75	5000		X		
S-24	1010 ST"	(CH ₃ O) ₃ B	175 25-300	300 65 days	X X			
S-25	1010 ST	(CH ₃ O) ₃ B	175	1200	X			

wick have not been successful. The copper mesh appears to reinforce the ice as it is formed rather than relieving the freezing stress. Copper mesh formed into a cylinder with a diameter equal to 20-25% of the inside diameter of the heat pipe may provide the wick movement which is required. However, the diameter is small and the cylinder of copper mesh is quite strong which may still act as a reinforcing rod rather than a stress reliever.

Several other possibilities exist which to date have not been tested. They include a copper feltmetal slab, bias woven copper braid, and copper wool. The feltmetal slab may perform well, however, its cost rules out its use. The bias woven copper braid likewise is relatively expensive, but may provide the flexibility required. Because of the small diameter of the copper heat pipe, the copper wool would be hard to use as it would fill the entire vapor space and may impede proper heat pipe operation.

An additional possible freeze survival method for copper-water heat pipes is to place the collector in an east-west orientation which places the heat pipe horizontal with enough positive tilt to make the heat pipe a thermal diode. The tilt is not enough to cause the liquid to bridge the inner diameter of the heat pipe in the evaporator which in turn reduces the freezing stress and rupture possibility.

In addition to the freezing-rupture problem associated with the copper-water system, copper will not survive the 3000 psi vapor pressure of water at stagnation temperatures.

However, if the fluid change is minimized, which benefits the horizontal freeze protection scheme, the internal pressure is reduced. This reduction in pressure occurs when the density of the water vapor is sufficient to fill the heat pipe with no liquid remaining at a given temperature. Then an increase in temperature will cause an increase in pressure according to the equation of state rather than the equilibrium vapor pressure curve.

The 10 copper-water heat pipes which were tested in the TC-100 collector panel at Thermacore for 130 days are in the process of being evaluated. Each heat pipe has been retested to observe any change from its initial test data. Seven of the heat pipes showed no change. The remaining three were observed to have varying degrees of differences from their initial test data. These discrepancies have been traced to leaks in the evaporator end closure. Preliminary estimate of the cause of the leaks is attributed to the use of an oxidizing flame to weld the evaporator closed as compared to a reducing flame for the heat pipes which remained sealed.

The volume of a $\frac{1}{4}$ " I.D. x 48" long heat pipe is 15.2 cc. Water has a critical density of 0.315 gm/cc at 374 C and a pressure of 221.29 Bar or 3210 psi. Critical fluid change is 4.788 grams of water. However, heat pipes have been fabricated and tested with fluid changes as low as 1.5 grams. Accordingly, the critical temperature of these heat pipes will be less than 374 C with a substantial reduction in pressure at 410 C.

If one assumes that the liquid and vapor charge are to be equal at 165 C, then the minimum fluid charge permissible is .11 gm of water which corresponds to the critical volume at 197 C. At 197 C, water has a vapor pressure of 211 psi. Super heating the water to 410 C will raise the pressure by a factor of 1.45 to 306 psi, which is well within the capability of copper at 400 C.

To date, no test data has been obtained for fluid charges below 1.5 grams in a 15 cc heat pipe. A 10% fill develops a pressure of 2250 psi at 410 C. Tests are warranted to explore the range from 1.5 grams to .11 grams charge to determine the lower fluid charge limit.

5.2 Steel-Ethanol

During the past six months the various steel-ethanol heat pipes on life test and performance test in the solar collector were removed for analysis. This action was precipitated by an increase in operational ΔT in two life test heat pipes and less than optimum performance of the 10 steel-ethanol heat pipes in the solar collector. Initially, the increase in operational ΔT and low performance was attributed to thermal decomposition of the ethanol into ethylene and water. Accordingly, one of the heat pipes from the solar panel and one life test heat pipe were submitted to General Electric for analysis.

To date, only the result from the solar panel heat pipe is available. The analysis showed that the fluid in the

heat pipe was 60% ethanol and 40% water and that there was no ethylene present over the liquid. The vapor analysis did indicate that there was air present in the heat pipe. However, the background scan performed prior to heat pipe puncture did not indicate the presence of ethanol, which implied that the pinch-weld did not leak.

X-ray analysis of the condenser pinch-weld showed evidence of cracking which was confirmed by a mass spectrometer leak check following the analysis. The leak was observed to be 10^{-5} STD cc of helium per second which is small enough to only allow the air to be pumped out of the crack during the background scan which would explain why ethanol was not seen in the scan.

The absence of ethylene over the ethanol may be due to the fact that it escaped through the crack, if in fact it was ever present, or that the working fluid never decomposed. In the latter case, the slow degradation of the heat pipe could have been due to water from the pumped loop being sucked into the heat pipe through the crack during the nighttime, non-operating condition.

The pinch-weld is being analyzed. It is suspected that the leak is due to external corrosion from the pumped fluid loop.

5.3 Aluminum Bearing Steel-Water

The aluminum bearing steel heat pipes are continuing to perform satisfactorily. However, several problems do exist.

Of major concern is the inability to find a manufacturer willing to make a pilot production run of 5/16" O.D. welded and drawn tubing. Without tubing of the correct size, it is impossible to consider these materials for use in the verification test to be performed in January, 1979.

In addition to the lack of tailor-made tubing, the use of water presents the same freezing problem with these steels as with copper. In fact, since wire mesh or steel wool is not available in the aluminum bearing steels, the anti-freeze mechanism would have to be fiberglass wick with stagnation control, east-west orientation with horizontal heat pipe operation, or the use of an additive to lower the freezing point. To date, no suitable additive has been identified which does not react with steel.

5.4 Steel-Trimethyl Borate

Testing of steel-trimethyl borate heat pipes has produced some interesting results. Initial testing of 5/16" O.D. heat pipes produced large ΔT 's which were reduced by going to 1/2" O.D. heat pipes. However, testing of several 1/2" O.D. heat pipes, including one in the solar panel under stagnation condition, has produced varying results. These results include heat pipes of identical construction and processing with large differences in the ΔT along the heat pipe at the same temperature and comparable power transfer.

In an effort to explain the performance of trimethyl borate, several items have turned up. Initially, poor

performance was attributed to trimethyl borate having a low latent heat which then requires a large fluid circulation and resultant vapor pressure losses within the heat pipe. The most likely result is sonic vapor velocity. These conditions have not been verified as the required physical properties of trimethyl borate have not been obtained. To date, efforts to obtain the necessary information through NTIS have not been successful.

Heat pipe operation at temperatures where sonic conditions should not occur also produced large ΔT 's in several heat pipes. The temperature profile along those heat pipes resembles that of a two fluid heat pipe such as an azeotrope. Accordingly, it was decided to determine if the supply of trimethyl borate was in fact pure.

Tests on the fluid by Thermacore's inorganic consultant, Dr. Alan G. MacDiarmid, showed that, indeed, the fluid was an azeotrope of 85% trimethyl borate and 15% methanol. Dr. MacDiarmid is currently purifying the fluid by treating it with lithium chloride, decanting the 99.8% pure fluid from over the settled precipitates and repeating. A spectrographic analysis is being prepared on the fluid to determine its composition. After verifying the purity, Thermacore is to be supplied with evacuated glass ampules of trimethyl borate for use in processing and testing a new series of heat pipes.

6. COST ANALYSIS

The selling price to the collector manufacturer of a heat pipe, FOB the manufacturing plant, has been quoted at \$1.50 each, without the fins. Appendix A discusses a rotary processing unit which reduces the processing time of the heat pipe to fifteen seconds. Table 6.1 shows the breakdown necessary for a \$1.50 heat pipe based on a material cost of \$.39 per heat pipe and four minutes of \$4.00 in labor. Figure 6.1 shows the relationship necessary between material cost and labor time to achieve the \$1.50 heat pipe for \$4, \$5 and \$6 per hour wage rates. Obviously, there is a trade-off between time, material and hourly wage rate. The curves of Figure 6.1 can be expressed as follows:

$$\frac{T \times R}{60} = \frac{1.06 - M}{2.5} \quad \text{Eq. 6.1}$$

Where T = Total labor time to make on heat pipe in minutes
R = Wage rate per hour
M = Material cost in dollars

From Equation 6.1, the limits on a \$1.50 heat pipe are \$1.06 for materials with no labor or \$1.06 for labor with no materials. The maximum labor content at \$4, \$5 and \$6 per hour wage rate at zero material content is 6.35, 5.05 and 4.25 minutes respectively.

From Figure 6.1, it is readily seen that materials like copper, which have a material cost of \$.55, would need to have a total labor content of three minutes or less in order

TABLE 6.1

COST ANALYSIS

Material	\$0.349	
Shipping 1% Material	0.004	
Fluid - Maximum	0.040	
Total - Material	\$0.393	- - - - - \$0.393
Labor - 4 Minutes of \$4.00/hr	\$0.267	
Overhead - 150%	0.400	
Total - Labor, Overhead	\$0.667	- - - - - \$0.667
TOTAL - Material, Labor, Overhead		\$1.060
Waste: 5%	\$0.053	
Total - - - - -		\$1.113
G&A: 10%	0.112	
Total - - - - -		1.225
Warranty:	0.060	
Total - - - - -		1.285
Warehousing: 5%	0.065	
Total - - - - -		1.350
Profit: 10%	0.135	
Total - - - - -		1.485
Shipping: 1%	0.015	
TOTAL		\$1.500

Figure 6.1

MATERIAL COST VS LABOR TIME FOR VARIOUS LABOR RATES

to meet the \$1.50 cost objective. On the other hand, a material such as 1010 steel which has a material content of \$.20, has a maximum labor content of 5.2 minutes. Accordingly, it behooves a heat pipe manufacturer to develop rotary processing equipment as described in Appendix A plus additional handling equipment to reduce the labor content to a minimum in order to meet the \$1.50 heat pipe price.

If 375,000 heat pipes are sold per year at \$1.50 each, this equals a net sales of \$562,500 per year. Using the premise that one person is required for every \$33,000 of net sales reveals that 17+ people are required to produce these heat pipes. The question thus becomes, is this consistent with the time required to produce the heat pipes. Table 6.2 details this analysis and shows that 17.3 people are indeed capable of producing 375,000 heat pipes per year at a selling price of \$1.50 each.

TABLE 6.2

NUMBER OF PEOPLE REQUIRED
TO MANUFACTURE 375,000 HEAT PIPES PER YEAR

Number of heat pipes shipped per year = 375,000

Total labor required per year at
4 minutes per heat pipe = 1,500,000 minutes

Normal work year per laborer is
242 days x 8 hours per day x 60
minutes per hour = 1,116,160 minutes/person

Total number of yearly minutes
divided by the yearly minutes per
laborer equals total direct charge
laborers = 12.9 people

Vacation, holidays, sick days = 15%
Divide 12.9 people by .85 = 15.2 people

Overhead people per laborer = 12%
Divide 15.2 people by .88 = 17.3 people

APPENDIX A

HEAT PIPE ROTARY PROCESSING UNIT

The attractiveness of heat pipes for use in many applications is often predicated on the ability to produce the product by automated or semi-automated volume manufacturing techniques which provide cost effectiveness in the market place. The intent of this design study is to show by means of a simplified manufacturing model, the practicality of manufacturing heat pipes by volume at competitive labor costs.

Discussion

The ability to produce cost effective heat pipes in volume has been an obstacle for many applications. For this reason, effort has been devoted to the design of a multiple station rotary processing unit in which repeatable processing can be attained with a minimum of direct labor effort. The design is based on 10 minutes of heat pipe processing and an adequate number of stations in which to achieve a practical rotary index time. The loading and unloading function is the only station where direct labor is required. A total of 40 stations with a 15 second rotary index time is practical and results in a machine capable of producing approximately 375,000 heat pipes per year for a single eight-hour shift with 85% efficiency and 5% scrap.

A simplified model utilizing five stations has been selected for evaluation of the various technologies required.

The five-station design is seen in Figure A-1 and A-2. A flow chart of this concept is shown in Figure A-3. The index time for this device would be 2 minutes per station giving the 10 minutes of required processing time for each heat pipe. Thus, all station functions and operations are similar to the ultimate 40 station unit. It is thus obvious that the simplified (5) station unit will provide adequate data to properly evaluate the concept.

The multiple station processing unit design has been developed based upon the following:

- a. Use of precleaned and vacuum-fired tubing as the heat pipe vessel;
- b. use of a pinched, welded, cut-off closure;
- c. one end of pipe closed after vacuum firing;
- d. utilization of a rotary vacuum seal concept similar to rotary vacuum tube exhaust machines;
- e. a processing time of 10 minutes per heat pipe, and
- f. an adequate number of stations to achieve a practical index time for the particular heat pipe being processed.

Since the basic operation can be described by the (5) functional stations, each performing an operation pertinent to the processing of the heat pipe, these stations and functions are described in detail.

Station #1 - Unload/Load

This station is the start/finish point of the processing operation. As the unit indexes from the #5 station to the #1 station, the completed heat pipe is removed, the exhaust

Figure A-3

FUNDAMENTAL FIVE STATION FLOW CHART

Figure A-1

ROTARY PROCESSING UNIT

STATION 1 LOAD & UNLOAD
 STATION 2 EVACUATE & HEAT (VACUUM PUMP 1)
 STATION 3 EVACUATE & COOL (VACUUM PUMP 2)
 STATION 4 EVACUATE, FILL & PINCH OFF } VACUUM
 STATION 5 EVACUATE & WELD } PUMP 3

Figure A-2

TITLE SECT A-A & DETAILS FROM C46-23	
NAME	STD. TOLERANCES
DRAWN RLC	DECIMAL .XX = ±.015 XXX = ±.005
DATE 10/23/78	FRACTION ± 1/64
APPROVED	ANGULAR ±.003 PER. INCH OF LENGTH
SCALE HALF MATL.	DWG. NO. C46-24
ASS'Y or BILL	
THERMACORE, INC. HEAT TRANSFER SPECIALISTS	

appendage removed and the next heat pipe inserted for start of processing. This involves removal of the clamp which provides power for resistance heating of the pipe, opening of the vacuum quick-disconnect, inserting of the next heat pipe into the vacuum fitting and attachment of the heating clamps.

The proper design and selection of components will permit a cycle time of fifteen seconds to be adequate for these operations. If necessary, the heating clamp attachment can be automated. Immediately upon indexing from station #1 to station #2, evacuation begins and heat-up is initiated.

Station #2 - Evacuation and Bake-Out

The use of a rotary vacuum seal permits evacuation of the heat pipes with a minimum of external control valves and minimum vacuum plenum length.

Heating is accomplished by resistance means with power introduced through slip-rings placed around the rotary unit. A back-out of 400 C is assumed and power input clamps would be located an adequate distance from the vacuum seal to provide the necessary thermal isolation.

The 2 minutes at temperature is adequate based on the use of previously out-gassed components. In the 40 station processing unit, additional heating stations are added until the index time times the number of stations equals the desired time at temperature.

Station #3 - Cool-Down

Thermal cool-down of the heat pipe is required to keep the working fluid vapor pressure low upon insertion. The method of cooling can be adjusted as required in order to achieve a reasonable time. Again, additional stations are utilized.

Station #4 - Fluid Insertion/End Closure

The repeatable insertion of an accurate quantity of working fluid is required if consistent quality heat pipes are to be produced by high volume processing techniques. A fixed volume, pneumatic controlled metering pump enables accurate deployment of a fixed amount of working fluid in each heat pipe. Thus, operator judgment is eliminated and "look alikes" should result. Adjustment can be made to accommodate varying heat pipe volumes and differ working fluids.

Following fluid insertion, the heat pipe end toward the vacuum connector is "pinched" together by means of pneumatically controlled clamping jaws. While this operation does not provide a vacuum tight seal at the "pinch" area, it does prepare the area for welding performed at station #5.

Station #5 - Closure Welding

Final vacuum closure of the heat pipe is accomplished by passing high current through the "pinched" area prepared at station #4. This is done by use of a stud-type spot welder

which applies high current and pressure to the weld area. The small section of heat pipe between the weld area and the vacuum fitting is clipped off when the completed heat pipe is removed at station #1.

Summary

A design of a prototype machine for the manufacture of heat pipes by high volume processing techniques is described. Practical processing technology along with innovative design provides manufacturing capability which makes heat pipes attractive for high volume, low cost applications.

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