DOE/CS/34696-1

(DE82007094)

Distribution Categories UC-59a, UC-59c,
UC-61, and UC-63

DEVELOPMENT OF A GAS BACKUP HEATER
FOR SOLAR DOMESTIC HOT WATER SYSTEMS

FINAL REPORT

APRIL 1978 - APRIL 1980

Dennis J. Morrison
Howard E. Grunes
Francis de Winter
Peter R. Armstrong

JUNE 1980

Contract No. DE-AC02-78CS34696-AC02

Prepared for
THE U.S. DEPARTMENT OF ENERGY

ALTAS CORPORATION
500 CHESTNUT STREET
SANTA CRUZ, CA 95060

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

ABSTRACT

A comprehensive program was undertaken to develop a unique gas fired backup for solar domestic hot water systems. Detailed computer design tools were written. A series of heat transfer experiments were performed to characterize the performance of individual components. A full scale engineering prototype, including the solar preheat tank and solar heat exchanger, was designed, fabricated and subjected to limited testing. Firing efficiency for the backup system was found to be 81.4 percent at a firing rate of 50,000 Btu/hr. Long term standby losses should be negligible.

TABLE OF CONTENTS

	<u>Page</u>
SECTION 1 INTRODUCTION	1.1
SECTION 2 COMPUTER MODELING	2.1
2.1 Heat Pipe Model	2.1
2.2 System Simulation Model	2.18
SECTION 3 EXPERIMENTAL WORK	3.1
3.1 Connecting Tube Experiment	3.1
3.2 Testing of Initial Heat Pipe Design	3.13
3.3 Performance Improvement Testing	3.19
3.4 Burner/Combustion Chamber Development	3.28
3.5 Demand Tank Stratification Test	3.28
SECTION 4 FINAL DESIGN, FABRICATION AND TESTING	4.1
4.1 Design and Fabrication	4.1
4.2 Testing	4.17
SECTION 5 COMMERCIALIZATION AND FUTURE WORK	5.1
SECTION 6 REFERENCES CITED	6.1
APPENDIX A Derivation of the Heat Pipe Computer Model	A.1
APPENDIX B Physical Properties of Water (Curve Fits)	B.1
APPENDIX C Listing of Heat Pipe Computer Model	C.1
APPENDIX D Derivation of Traced Coil Computer Model	D.1
APPENDIX E Listing of System Simulation TRNSYS Subroutines ..	E.1
APPENDIX F Nomenclature	F.1

Section 1

INTRODUCTION

This is a final report on the activities performed under contract to the Department of Energy toward the development of a highly efficient gas backup heater for solar domestic hot water systems. This contract resulted from a Program Research and Development Announcement entitled "Thermal Energy Storage Subsystems for Solar Heating and Cooling Applications" (PRDA EG-77-D-29-0004) which was issued by the DOE (then ERDA) in early 1977. An award was made in August of that year and a contract for a two year period of performance was received in July 1978.

The primary objective of the program was to design, fabricate, and test an engineering prototype of the proposed heater design. The prototype was to be safe, reliable, and efficient; with 80 percent or higher firing efficiency, and negligible standby losses. In addition, it was to resemble, as close as possible, a production model. Having met these objectives, within budget and on schedule, the program drew to a successful conclusion on April 30, 1980.

The essence of the proposed heater is the elimination of the central flue found in conventional gas water heaters, and the unique positioning and coupling of the backup and solar water tanks. A schematic of the hardware, including the solar tank and heat exchanger, is shown in Figure 1.1, and a picture of the final prototype is included as Figure 4.2. The backup heater has the following features:

1. Auxiliary energy is input to the backup tank via a gravity return heat pipe (no wick). The heat pipe evaporator is an array of horizontal finned tubes which are manifolded and connected to a jacketed backup tank. The condenser is then an integral part of the tank itself.
2. Parasitic standby losses should be below 500,000 Btu/year for a typical application, which is less than 1 percent of the 502 therms average annual gas consumption of present day heaters [38].
3. The heater uses an electronic pilot to further minimize gas usage.

FIGURE 1.1 SCHEMATIC OF THE ALTAS SOLAR AUGMENTED WATER HEATER

4. The gas burner has a constant firing rate with thermostatic on-off control.
5. The backup tank is positioned directly above the solar preheat tank, where it becomes an integral part of the solar storage.
6. The passageway between the two tanks incorporates a plastic insert, specially designed to act as a "thermal diode", which will readily convect solar heat upwards whenever the preheat tank temperature exceeds the backup tank temperature, but will only conduct heat downwards poorly whenever the reverse is true. An inherent stratification is involved in the design.
7. The entire domestic hot water heating system, except for the collector and collector loop pump, is included in a single self contained unit. With this hardware, a solar installation can be made by simply connecting the collector and the pump.

The program was originally conceived as twelve distinct tasks, listed below:

Task a - Development of an accurate computer model.

Task b - Determination of optimum materials, configuration, and design parameters of the backup heater tank/heat exchanger.

Task c - Determination of the best fluid to use in the heat pipe.

Task d - Determination of optimum heat pipe design parameters.

Task e - Determination of the connecting "thermal diode" design parameters.

Task f - Insulation and backup tank support study.

Task g - Determination of optimum materials, configuration, and design parameters of the gas burner/heat exchanger.

Task h - Maintenance of a continuing dialogue with the AGA, the IGT, the PG&E Co., and other utility and gas heater manufacturer personnel.

Task i - Documentation and technology transfer program.

Task j - Engineering prototype design.

Task k - Engineering prototype fabrication.

Task l - Experimental component work and final prototype testing.

These tasks proceeded in three distinct phases: 1) a theoretical analysis concentrating on the development of computer models of both the heat pipe assembly and the complete system; 2) experimental work to verify the theoretical analysis and to examine the feasibility of the preliminary designs proposed for the individual components; and 3) design, fabrication, and testing of the final prototype.

The first phase resulted in the development of two computer design tools. The first was a model describing the thermal and fluid mechanical performance of the heat pipe assembly. It was used to arrive at a preliminary design of the heat pipe components, and to investigate the various options available for the final design. The second model was a modified version of the solar system simulation program TRNSYS [14]. This model was used to investigate the effect of tank coupling on the long term performance of the complete system, and to compare the performance of the proposed heater to that of a comparable "conventional" solar system.

The second phase of the program entailed a series of five experiments; one to investigate tank coupling hardware (connecting tube geometries), three to acquire performance data and operating experience with the heat pipe system, and one to examine stratification in the backup tank. These experiments represent the heart of the development effort on this program.

In the final phase of the program, the computer results, performance data and operating experience were synthesized into a final engineering prototype design, which was then fabricated and tested.

The prototype has yet to be tested as a complete system with solar input and a realistic hot water demand. However, as part of this program, laboratory tests have shown the backup system to be extremely reliable and efficient, with a firing efficiency of 81.4 percent at a firing rate of 50,000 Btu/hr. Still unknown is the impact of tank coupling on overall system performance. In its present state, the system simulation model has not identified any impact; but as discussed in Section 2.2, final conclusions cannot be drawn until additional capabilities have been added to the model and more simulations performed.

The remainder of this report consists of detailed descriptions of the three phases of the program; Computer Modeling (Section 2), Experimental Work (Section 3), and Final Design, Fabrication and Testing (Section 4). Commercialization and Future Work are described in Section 5. Appendices A through F contain computer listings, theoretical background for the computer models, and nomenclature.

Prior to the publication of the final report of this U.S. DOE funded effort, enough results have come in from the Gas Research Institute (GRI) funded program (see Section 5) to merit discussion in this report.

Significant design changes were made in the concept to reduce the cost without reducing the performance. This included using glassed steel tanks for both the upper and lower tanks. The traced coil solar heat exchanger was replaced by a drainback system using only water (GRI patent applied for) which divided the solar tank into a pressurized and unpressurized dual tank system. The jacketed condenser system for the gas backup device was replaced with a finned-tube condenser fitted into a side flange on the upper tank. A GRI patent application was made on the new method to transfer the gas heat to the backup tank. Packaging was improved and the equipment size was reduced without reducing the volume of storage.

Performance numbers were very good. Firing efficiencies of 81-82 percent were maintained, at firing rates of 39,000 BtuH (the earlier 50,000 BtuH were deemed excessive) at CO concentrations in the flue gases of less than 100 PPM, and flue gas temperatures of 250-280°F. Stand-alone firing efficiencies (no solar heat input) were between 76-77 percent for a standard draw cycle of about 92 gallons per day, using a number of showers and other consumption components spread over a 24 hour period. This illustrates the low standby losses of the heater. With a 38 ft² collector system (two single glazed Sunworks panels with a selective coating) and the 92 gallons per day draw, a monthly "solar fraction" of 78 percent was obtained experimentally, with the 41,000 Btu per day load corresponding to this draw (thermostat setting was 120°F) taking only 11,400 Btu of gas, for a 79 percent efficiency used to supply the "gas fraction". As a comparison, a standard gas heater unit with a 700 BtuH pilot flame will consume 16,800 Btu per day even when coupled to a system with a solar fraction of 1.0.

Defining exact manufacturing costs will require significantly more work than was involved in the DOE and GRI funded efforts. At present it is

estimated at Altas that the manufacturing cost of a complete package, excluding collectors and the plumbing leading to them, but including the pump, controller, electronic pilot, solar tank with drainback system and backup tank, will be about \$400.00 in mass production. Discussions with several interested manufacturers are going on.

Section 2

COMPUTER MODELING

Two computer design tools were developed as part of this project; a model for predicting the steady state performance of the heat pipe/backup tank sub-assembly, and an extensively modified version of the solar system simulation program TRNSYS [14] for predicting the transient performance of the complete system. The heat pipe program is a detailed model of the heat transfer and fluid mechanical characteristics of each of the heat pipe components. It was used to investigate the potential of different heat pipe geometries and materials and also to generate performance data for use in the transient system simulations. The TRNSYS simulation model was used to optimize various system design parameters, and to predict the long term performance of the combined solar and gas backup systems.

Section 2 describes the development and application of these two models. Section 2.1.1 introduces the theory behind the heat pipe model, 2.1.2 describes the code itself, and 2.1.3 the results. Sections 2.2.1, 2 and 3 deal with the same topics for the system simulation model.

2.1 HEAT PIPE MODEL

2.1.1 Heat Pipe Theory

The primary objective in modeling the heat pipe assembly is the ability to predict, given the backup tank temperature and the conditions of combustion (i.e., the flame temperature and the combustion products mass flow rate), the rate of energy transfer into the tank. In addition, intelligent design and optimization require knowledge of 1) the amount of fuel burned (and hence the overall efficiency); 2) the temperature at key locations; 3) the resistance to heat transfer associated with each component; 4) the pressure inside the heat pipe; 5) the pressure drop in each section; 6) the sonic, entrainment, and boiling limitations to heat transfer; and 7) the quantity of fluid entrained in the entire assembly. The heat pipe model will supply this information for heat pipes conceptually identical to the one shown in Figure 2.1; a tubular evaporator partially filled with heat transfer fluid; an adiabatic connecting section; and a condenser employing film condensation on the inner surface and free convection on the outer

FIGURE 2.1 HEAT PIPE HEATER CONFIGURATION

surface. The mathematical development of this model is summarized below, and a complete derivation including all the relevant equations is included as Appendix A.

The energy transferred from the combustion site to the backup tank via the heat pipe can be estimated using a modification of the heat transfer effectiveness (ϵ) and number of transfer units (NTU) method developed by Kays and London [7]. As shown in Appendix A, the overall heat transfer, Q , can be expressed as follows:

$$Q = \epsilon_{\text{tot}} (\dot{m}_p)_{\text{prod}} (T_{\text{flame}} - T_t) \quad (2.1)$$

where

ϵ_{tot} = overall heat pipe effectiveness

$(\dot{m}_p)_{\text{prod}}$ = capacity rate of combustion products entering the heat pipe evaporator, Btu/hr $^{\circ}\text{F}$

T_{flame} = temperature of combustion products, $^{\circ}\text{F}$

T_t = temperature of water inside backup tank, $^{\circ}\text{F}$

In simplified terms, the overall heat pipe effectiveness is a measure of how close the temperature of combustion gases entering the flue approaches the temperature of the water in the tank. An effectiveness of 1 represents a perfect heat exchanger; the maximum possible heat transfer rate under the given conditions.

In Equation 2.1, the tank temperature is the only "known", and hence, it is the function of the heat pipe model to calculate the temperature and capacity rate of the combustion products, and the overall heat pipe effectiveness, given the desired "heater rating", the percent theoretical air available, and the heat pipe construction. To do so, it must estimate the heat transfer resistance at each point in the heat pipe assembly, including: the convection heat transfer coefficient on the outer surface of the evaporator tubes; the resistance of the evaporator tube wall; the boiling heat transfer coefficient on the inside surface of the evaporator; the vapor pressure drop between the evaporator and the condenser; the film condensation heat transfer coefficient on the inner wall of the condenser (actually the outer wall of the tank); the resistance of the condenser (the tank) wall; and the free convection heat transfer coefficient on the outer wall of the condenser (actually the inner tank wall). In addition, once the overall

heat transfer rate is known, the model also checks the sonic, entrainment, and boiling limitations to heat transfer; the operating pressure; the temperature at key locations; and the quantity of fluid entrained in the entire assembly. These are checked to insure that the operating conditions are reasonable, safe, and within the bounds of the data used to predict them.

Empirical correlations for predicting the various heat transfer phenomena, and the limitations to heat transfer can be found in the literature and are discussed in detail in Appendix A. However, it should be noted that in some cases the data are limited or incomplete, and their application may lead to erroneous results. These circumstances, where present, are discussed in more detail in Section 2.1.3.

2.1.2 Heat Pipe Computer Code

The heat pipe model actually consists of seven distinct routines written in FORTRAN Extended Version 4 for use on Lawrence Berkeley Laboratory's CDC 7600. The function of each routine is described below, and complete listings are included as Appendix C.

DRIVER is the executive routine for handling input and output and is used primarily for debugging purposes. It is designed to run a single heat pipe operating point and print detailed output.

DATAPT is the executive routine designed for production runs (parametric studies) and generating "performance maps". It creates a matrix of operating points as input, and once the heat pipe heat transfer is calculated for each point, it places the results in an output file. This file is formatted so that it can be used directly by the system simulation program (see Section 2.2).

HPIPE performs the actual heat pipe heat transfer calculations. Operating points are completely specified by the heater rating, the percent theoretical air used in the burner, and the backup tank temperature. Using these operating points as inputs, and a list of parameters describing the heat pipe geometry and materials, HPIPE iterates to a steady state solution for overall heat transfer. The iterative process involves making an initial guess of overall heat transfer, calculating the heat transfer coefficients and fluid properties at each point in the assembly, calculating a new overall heat transfer rate, and repeating the process until the solution converges.

HTC is the subroutine which calculates boiling, film and free convection heat transfer coefficients for use in HPIPE.

PDROP calculates the pressure drop in each section of the heat pipe assembly.

H2O evaluates the thermal and physical properties of the liquid and vapor phases of the heat pipe fluid as a function of temperature. Heat pipe performance is extremely sensitive to variations in the properties of the heat pipe fluid and, to a lesser degree, the water inside of the backup tank. The computer analysis examined only water as the heat pipe working fluid. Hence, only a single routine, capable of furnishing the properties of water and steam as a function of temperature, has been required.

H2O consists of curve fits for the viscosity, density, thermal conductivity, specific heat, coefficient of thermal expansion, and surface tension coefficient for water, and the viscosity, density, and thermal conductivity of steam. These curve fits, along with the applicable temperature ranges, are shown in Appendix B.

CPPROD calculates the specific heat of the combustion products as a function of the percent theoretical air and the flame temperature. For the purpose of many engineering applications, the specific heat of pure air can be used without encountering significant error. However, for the range of conditions present in this application more accurate data are required. CPROD uses the Gibbs-Dalton law to determine the specific heat from the composition of the product mixture and the specific heat of the individual components.

2.1.3 Heat Pipe Results

The heat pipe model was first used to develop a preliminary design for the heat pipe/backup tank assembly. This exercise was extremely useful in designing the experimental heat pipe apparatus, making initial material purchases (e.g., selecting finned tubing), and in general, determining the feasibility of using a heat pipe for this application. As data from the heat pipe experiments became available, the accuracy of the model was checked against actual performance data, and limitations on its use as a design tool were determined. The model was found to be very effective for estimating the relative effects of varying important design variables, and investigating a variety of design alternatives over a range of operating conditions which could not possibly have been examined experimentally.

The results obtained with the computer model are described below. Section 2.1.3.1 describes comparisons to the experimental data, 2.1.3.2 through 2.1.3.4 examine design consideration for each of the heat pipe components, and Section 2.1.3.5 describes the performance limitation considerations.

2.1.3.1 Comparison to Experimental Results

The most significant comparison between theoretical and experimental results is how accurately the model predicts the overall heat pipe heat transfer, Q . Such a comparison is shown in Figure 2.2, where Q and the corresponding efficiency are plotted against the firing rate, or more appropriately, the "heater rating", which is the theoretical energy available in the fuel burned. Both the firing rate and the heater efficiency are based upon the higher heating value of methane. These results are for a copper/water heat pipe consisting of an evaporator with six, 8 inch-long helical finned tubes (O.D.=.729 in, ID=.591 in, fin diameter = 1.517 in and 9.2 fins per inch); a foot long, 1 inch diameter connecting tube; and a 15 inch diameter, 18 inch tall, 20 gallon backup tank with 16, 7/8 inch nominal "D-tubes" brazed vertically to the outer wall to create the condensing "channels". This is the experimental apparatus described in Section 3.1, and it is identical in design to the engineering prototype except that the prototype condenser is a double-walled jacketed tank.

In Figure 2.2, it can be seen that the model predicts a firing efficiency which is lower than that of the real system by 6 to 15 percent, depending upon the firing rate and what is assumed to be the percent theoretical air used in the burner (a point to be discussed in more detail below). The major portion of this error can be attributed to the correlation used for predicting the convective heat transfer coefficient on the outer surface of the finned tube evaporator. With an atmospheric burner (no fan), the thermal resistance of this boundary layer is much larger, typically an order of magnitude, than that at any other point in the assembly. Unfortunately, the correlations and experimental data used to predict the convection heat transfer coefficient represent the largest uncertainty in the calculations.

As discussed in Appendix A, there are considerable experimental heat transfer data available for gases flowing across finned tubes. However, most of the data are for Reynolds numbers above 1,000 and all of it certainly for Reynolds numbers above 100. Unfortunately, computer

FIGURE 2.2 HEAT PIPE HEAT TRANSFER VS. FIRING RATE

runs have shown that Reynolds numbers for this atmospheric burner/finned tube evaporator combination (based upon minimum free flow area) are typically below five hundred and in some cases below one hundred. The Reynolds numbers corresponding to the data shown in Figure 2.2 range from 49.2 at a 20,000 Btu/hr firing rate, to 208 at 70,000. As a result, the computer model must extrapolate the generalized correlation for predicting the convection heat transfer coefficient (which is based upon the experimental data) to lower Reynolds numbers.

Obviously, considerable error can result from extrapolating heat transfer data to Reynolds numbers outside of the intended range. However, from the limited data available for low Reynolds numbers, it was apparent that the correlation used in the model will always underpredict the convective heat transfer coefficient, and hence, yield a conservative estimate of performance. The results illustrated in Figure 2.2 show this to be the case.

Another significant comparison which can be made between theoretical and experimental results is the operating temperature of the vapor inside of the heat pipe. The vapor temperatures corresponding to the heat transfer results shown in Figure 2.2 are plotted in Figure 2.3. Here, the computer results agree quite well with those from the experimental setup. This was expected, since for a given overall heat transfer rate, the vapor temperature is a function only of the conditions present at the condenser; the free convection heat transfer coefficient on the inner tank wall, the temperature of the water in the tank, and to a much lesser degree, the film condensation heat transfer coefficient on the inside of the condenser channels. It should be noted that the tank temperature referred to here is the average temperature of the water in the tank. The computer model assumes a fully mixed tank, but in reality, the tank becomes stratified during firing. Stratification is even more pronounced in the experimental apparatus because of the continuous throughput used to achieve steady state and to make heat transfer measurements (see Section 3.5). Hence, for all of the computer runs, the tank temperature was simply assumed to be the average of the temperature at the top and bottom of the tank.

The effect of combustion conditions on heat pipe performance is illustrated in Figure 2.4, where the overall heat transfer and corresponding efficiency are plotted against the percent theoretical air used in the burner. The experimental apparatus is the same used to generate the other results presented thus far, with the firing rate held constant at 42,000 Btu/hr. Here, the percent theoretical air is defined as the ratio of the actual amount of air used, and that

FIGURE 2.3 HEAT PIPE VAPOR TEMPERATURE VS. OVERALL HEAT TRANSFER

FIGURE 2.4 HEAT PIPE EFFICIENCY VS. PERCENT THEORETICAL AIR

required by stoichiometry; 100 percent theoretical air is then the stoichiometric mixture. In theory, for any given flow rate of gas, and hence, firing rate (based on the higher heating value of the gas), the temperature and capacity rate of the combustion products are dependent only upon the percent theoretical air used in the burner.

The most interesting effect to be observed in Figure 2.4 is how little efficiency is effected by varying the percent theoretical air. As is shown in Appendix A (Section A.3), the flame temperature, and hence, the overall heat pipe temperature difference, decrease as the percent theoretical air is increased. However, the mass flow rate of combustion products, and in turn, the convection heat transfer coefficient on the outer wall of the evaporator, increase as the percent theoretical air is increased. These two effects appear to balance out resulting in a relatively constant heat transfer rate over a range of fuel to air mixtures.

The lower firing efficiency predicted by the computer model when compared to the experimental data is again apparent in Figure 2.4, as it was in the results shown in Figure 2.2, and for the same reason; a low value predicted for the convection heat transfer coefficient on the outer wall of the evaporator. Figure 2.4 also illustrates some combustion effects that the model does not take into account. It is believed that the efficiency begins to drop at the lower air mixtures because of incomplete combustion. The model assumes that complete combustion occurs, and that the products reach the theoretical flame temperature (again see Section A.3). This is probably a reasonable assumption for the combustion of methane, however, it may lead to significant error for atmospheric burners as the percent theoretical air approaches 100. In most of the heat pipe experimental work, and for the engineering prototype, the percent theoretical was set at approximately 150 percent.

2.1.3.2 Condenser

As discussed in the previous section, the thermal resistance at the evaporator end of the heat pipe is much greater (typically an order of magnitude) than that at any other point in the assembly, and hence, is the limiting factor with regard to overall heat transfer. Therefore, the computer optimization strategy for the adiabatic and condenser sections has been to reduce size (i.e., materials) and complexity until either the overall heat transfer is significantly affected, or until one of the performance limitations is exceeded.

Results show that the limiting factor in condenser design is not the overall heat transfer, but instead the associated operating pressure. This was alluded to above in the discussion of heat pipe vapor temperature as a function of firing rate. In Figure 2.5, the heat pipe pressure and overall heat transfer are plotted as a function of the condenser area. These results are for the same copper/water heat pipe described in Section 2.1.3.1, except that an additional evaporator tube has been added. The area of the jacketed tank wall (condensing area) was varied from just under one square foot (roughly 13 percent of the total tank wall), to just under four square feet (roughly 60 percent).

Figure 2.5 shows that reducing the wall area by a factor of 4 results in a 3.5 percent drop in overall heat transfer, but a tenfold increase in operating pressure. In light of the intended application and restriction on pressurized vessels, it was decided that this pressure should be at or below atmospheric. With regard to heat transfer it is not necessary to jacket the entire backup tank; however, for the first engineering prototype it was decided to do so in order to guarantee a reasonable pressure for any conceivable operating conditions.

Another important condenser condition examined was the effect of the backup tank temperature on heat pipe performance. Overall heat transfer and operating pressure are shown as a function of tank temperature in Figure 2.6 for the heat pipe described above. As expected, increasing the tank temperature has the same effect as reducing the condenser heat transfer area.

2.1.3.3 Adiabatic Section

The only critical design consideration for the adiabatic connecting section is that it not be so small that excessive vapor velocities cause entrainment of the returning condensate. Hence, the optimization procedure is to size the evaporator and condenser, and then decrease the size of the connecting tube until entrainment becomes a factor. Such a process is illustrated in Figure 2.7 where the critical gas velocity (defined in Section A.2.2) is shown for various adiabatic tube sizes. The onset of entrainment has been shown to occur for a critical gas velocity of 2.46×10^{-4} . For the heat pipe configuration described above, Figure 2.7 shows that entrainment will begin to occur as the tube diameter approaches 1/2 inch.

FIGURE 2.5 HEAT PIPE HEAT TRANSFER AND OPERATING PRESSURE AS A FUNCTION OF CONDENSER AREA

FIGURE 2.6 HEAT PIPE HEAT TRANSFER AND OPERATING PRESSURE VS. TANK TEMPERATURE

FIGURE 2.7 ENTRAINMENT LIMIT VS. VAPOR TUBE DIAMETER

2.1.3.4 Evaporator

Although the model underestimates the convective heat transfer coefficient on the outer wall of the evaporator, it is still extremely useful for examining the effects of varying evaporator tube and array geometries. For example, Figure 2.8 illustrates the effect of varying evaporator area and geometry (based upon the extrapolated values for the evaporator convection heat transfer coefficient). Here the number of finned tubes (those described above) was varied from three to thirteen in both single and double row arrays. In this case the condenser is the half-round vertical "D-tube" configuration covering roughly 34 percent of the 20 gallon tank.

As expected, increasing the evaporator area increases the overall heat pipe heat transfer, but not nearly to the degree expected. For example, for the single row case, increasing the area from five square feet to just over twenty results in only a 20 percent increase in overall heat transfer. This is because increasing the evaporator area also increases the minimum free flow area, which in turn decreases the Reynolds number and the resulting convection heat transfer coefficient. This is illustrated in Figure 2.8 where heat transfer, Q , is on the left ordinate and the heat transfer coefficient, h_e , is on the right ordinate. This phenomenon is also responsible for the dramatic increase in heat transfer for the same evaporator area when the array is changed from one row to two.

2.1.3.5 Performance Limitations

It is interesting to note where the heat pipe operating points described thus far lie with regard to the performance limitations (see Section A.2 for a complete description of these). Operating pressure has already been discussed, and it is apparent that it is an important design consideration. However, of the other three; sonic, boiling, and entrainment, only the entrainment appears to be an important factor in the design process for this application.

The sonic limit occurs when the vapor velocity at the evaporator exit is increased to a critical or choked flow state, and conditions at the condenser are no longer transmitted upstream. However, none of the operating points simulated thus far have begun to approach the sonic limit, which for these evaporator configurations is on the order of hundreds of thousands of Btu per hour.

FIGURE 2.8 HEAT PIPE HEAT TRANSFER AND EVAPORATOR CONVECTIVE HEAT TRANSFER COEFFICIENT AS A FUNCTION OF EVAPORATOR AREA

The boiling limit is the point at which the peak nucleate boiling heat flux in the evaporator tubes is exceeded and the heat transfer rate decreases as the boiling mechanism makes the transition from the nucleate to the film regime (see Section A.1.3). Under the conditions simulated thus far, the boiling limitation, like the sonic limitation, has not been a factor in the design process. An example of the maximum nucleate boiling heat transfer rate as a function of heat pipe firing rate is shown in Figure 2.9 for the heat pipe configuration described above. Here it is clear that the peak nucleate boiling heat flux is always considerably above any reasonable firing rate for this application.

As discussed in Section 3, entrainment has been a problem in the experimental systems. It is apparent that at certain points in the flow passages, such as bends and component interfaces, entrainment is occurring which the model is incapable of predicting, and for which no applicable data exist. Entrainment in the straight sections of the assembly can be estimated, as was done for the adiabatic section (see Figure 2.7), but the problem can really only be overcome experimentally. As described in Section 3, this was done for the current prototype design and resulted in the incorporation of a separate return tube to segregate the vapor from the returning condensate.

2.2 SYSTEM SIMULATION MODEL

Computer modeling at the system level was used on this project to estimate the performance characteristics of the stacked tank arrangement, and to assist in selecting the optimum backup tank size and gas firing rate. In order to address these considerations, the computer program TRNSYS [14] was modified to accommodate the unconventional components found in the heat pipe heater.

2.2.1 System Simulation Theory

TRNSYS is a transient simulation computer program developed at the University of Wisconsin Solar Energy Laboratory for simulating solar energy systems. It consists of executive and I/O (input/output) routines, and a collection of subroutines which model the individual components present in a solar system such as the collector, pumps, valves, storage and load. The ducts, pipes, and wires which connect the components of a real system are represented by the flow of information between subroutines. The algebraic and differential equations present in each subroutine are solved simultaneously with weather data and load profile forcing functions.

FIGURE 2.9 PEAK NUCLEATE BOILING HEAT FLUX VS. FIRING RATE

Because it is a modular program, TRNSYS is ideally suited to a study involving new hardware because any new component models can be integrated without affecting the executive, I/O or other component models. Such is the case here. In order to simulate the heat pipe heater system, new routines for the traced tank heat exchanger, the connecting thermal diode, and the heat pipe assembly were required. These were written independently, according to the TRNSYS format, and incorporated without changes in any of the other TRNSYS routines.

2.2.1.1 Heat Pipe/Backup Tank Model

The heat pipe assembly and backup tank were combined in a single TRNSYS routine. The parameters used by the routine to describe the backup system are the tank height and diameter, the set temperature of the gas backup heater, the overall heat transfer coefficient between the tank and the environment, and the percent theoretical air used in the burner. The time dependent inputs are the mass flow rate and temperature of the water entering from the solar tank (when there is a demand), the net energy transfer through the connecting diode, and the temperature of the air in the vicinity of the tank (for calculating losses). At each time step during the simulation the routine determines the current backup tank temperature, the energy input from the heat pipe, the energy delivered to the load, and the losses to the environment.

Transient heat pipe performance is determined from a "performance map" generated using the detailed heat pipe model described in Section 2.1. For long term simulations it is impractical to solve the complete set of heat pipe equations at each time step. Instead, the equations are solved in advance for a range of anticipated operating conditions. The results are placed in a two dimensional array, or "performance map", which gives the overall heat transfer for any combination of operating points. Here, the operating points (the coordinates of the map) are the tank temperature and the percent theoretical air, to which the flame temperature is directly related. The performance map is generated by the heat pipe routine DATAPT (see Section 2.1.2). The TRNSYS heat pipe routine simply reads the performance map and, for any operating point of interest, performs a two dimensional linear interpolation to arrive at the appropriate heat transfer.

The backup tank is assumed fully mixed and its change in temperature from one time step to the next can be calculated using a simple energy balance as follows:

$$M_t c_p \frac{dT_t}{dt} = \dot{Q}_{load} + \dot{Q}_{diode} + \dot{Q}_{pipe} + \dot{Q}_{env} \quad (2.2)$$

where

M_t = mass of water in tank, lbm

c_p = specific heat of water, Btu/lbm °F

The term $\dot{Q}_{load}$ is the rate of enthalpic energy transfer from the tank due to hot water being drawn from the top and replaced by water from the solar tank. $\dot{Q}_{diode}$ is the net rate of energy transfer between the two tanks via the connecting thermal diode. This quantity is calculated in a separate routine described below. The term $\dot{Q}_{pipe}$ is the rate of energy input through the heat pipe taken from the performance map. Finally, $\dot{Q}_{env}$ is the standby loss term.

Incorporating the heat pipe/backup tank model into TRNSYS presented a problem in selecting the appropriate simulation time step. In terms of accuracy and cost, the optimum TRNSYS time step is on the order of 15 minutes. Although this is sufficient for the solar portion of the program, it is much too large for the heat pipe/backup tank routine because the burner may fire for much shorter periods. The problem was resolved by subdividing the TRNSYS time step within the heat pipe/backup tank routine (a simulation within a simulation). TRNSYS can then be run at its optimum time step while the heat pipe simulation time step is adjusted independently.

A listing of the heat pipe/backup tank routine (labeled TYPE 31 in keeping with TRNSYS nomenclature) is included in Appendix E along with copies of the other routines described in this section.

2.2.1.2 Connecting Tube Model

Before experimental work was begun, a model of free convection in the tank connecting tube was formulated. This model assumed steady, laminar, annular counterflow. The model first solved for the diameter of the imaginary inner annulus wall which minimizes friction loss, and then solved for the flow rate at which the friction loss head is just equal to the difference between the static head of the liquid falling through the tube and the static head of the liquid rising through the tube. The heads differ not because the liquid column heights differ but because the temperatures of the two streams, and hence their densities, differ. A TRNSYS subroutine using this model was written to solve for the rate of heat transfer between the tanks during simulation.

Experimental results showed that heat transfer was poorer than predicted by the annular flow model. Further investigation with a flow visualization experiment revealed that steady laminar flow is not obtained in a plain cylindrical channel because a smooth interface between the counterflowing streams is quickly destroyed by shear stresses. Consequently, the unbaffled connecting pipe was abandoned and a novel "split tube" arrangement devised. As discussed in Section 3.1, experimental results have shown this to be an extremely efficient configuration, especially if the separate upward and downward flow passages are "offset".

The development of a TRNSYS subroutine for the split tube arrangement began with the derivation of an approximate mathematical relationship for its performance. Figure 2.10 shows a split tube with offset which produces an upflow of hot water and a downflow of cold water which are kept separate. If it is assumed that the bottom tank is heating up at a constant rate, and the process has settled down so that the top tank is following with a constant temperature lag, an expression for the temperature lag can be derived rather simply. Assuming the top tank and the top node of the bottom tank are well mixed, so that the temperatures can be represented by T_t and T_b , then the increase in heat content of the top tank is equal to the net transport of heat through the connecting tube:

$$\dot{M}_t c_p \frac{dT_t}{d\tau} = \dot{m}_d c_p (T_b - T_t) \quad (2.3)$$

where

$$\dot{m}_d = \text{mass flow rate through diode, lbm/hr}$$

Equating the buoyancy terms to the pressure drop yields:

$$\begin{aligned} h\rho\beta (T_b - T_t) &= \frac{2\rho v^2}{2g} \left[1.5 + f \frac{L}{D} \right] \\ &= \frac{\dot{m}_d^2}{g\rho A_d^2} \left[1.5 + f \frac{L}{D} \right] \end{aligned} \quad (2.4)$$

where

f = average friction factor

v = velocity of water in diode, ft/sec

β = coefficient of thermal expansion, $1/^\circ\text{R}$

ρ = average density of water, lbm/ft³

FIGURE 2.10 SPLIT TUBE WITH OFFSET

The factor 1.5 approximates the entrance and exit losses. Combining Equations (2.3) and (2.4), we get:

$$T_b - T_t = \sqrt{\frac{3 \left(1.5 + \frac{L_f}{D}\right) M_t^2 \left(\frac{dT_t}{d\tau}\right)^2}{\rho^2 g \beta h A_d^2}} \quad (2.5)$$

From Equation 2.5, and the experimental results described in Section 3.1, it was apparent that the overall heat transferred by the connecting tube when the bottom tank is hotter than the top (Q_{diode} in Equation 2.1) can be expressed quite simply as follows:

$$Q_{diode} = K(T_b - T_t)^{1.5} \quad (2.6)$$

where K is a constant dependent upon the particular tube geometry.

The final TRNSYS connecting tube model is based upon Equation 2.6. Optimum tube design was investigated experimentally and K determined for several of the more promising alternatives (again, see Section 3.1). With this data it was possible to accurately simulate connecting tube performance, and in turn, investigate any advantages to be gained from a stacked tank arrangement.

2.2.1.3 Traced Tank Model

The steady state heat transfer rate across the traced tank heat exchanger depicted in Figure 2.11 is a function of the tank's vertical temperature profile and the temperature of the collector fluid as it enters the traced coil. A closed form solution for this heat transfer rate cannot be derived. Therefore it was necessary to solve for the heat transfer rate numerically. Because it is computationally inefficient to solve for the heat transfer rate in this manner over and over again during simulation, an empirical "performance map" was computed in advance as was the case for the heat pipe routine described above. This involved solving the heat transfer rate for an array of

FIGURE 2.11 PHANTOM VIEW OF TRACED TANK

tank conditions and collector fluid temperatures. A table of effective heat transfer conductances was generated and all that was required during the simulation were table lookup and interpolation operations.

The traced tank behaves approximately like a cross-flow heat exchanger with fins on both sides. Referring to the cross-section, Figure 2.12, each tube comprises two curved fins; the section of tank wall between tubes also comprises two fins which are straight in the vertical direction. Thus each tube/tank bond is the "base" of four fins. The "ends" of the tube fins are essentially equidistant from the bond; the ends of the tank fins are also essentially equidistant from the bond. While the ends of these fins are not physically salient they are adiabatic which is all that is important in terms of the thermal model.

Employing this crossflow heat exchanger analogy, and assuming the solar tank is fully mixed, it was possible to simultaneously solve for the average temperature of the inside surface of the coil, the corresponding average temperature of the tank's inside surface, the corresponding free and forced convection heat transfer coefficients, and in turn, the rate of heat transfer per unit wall area. A detailed description of the derivation of this model is included in Appendix D.

2.2.2 System Simulation Computer Code

Four new TRNSYS subroutines were written, listings of which are included as Appendix E. In addition, a series of eight routines were used to generate the traced coil performance map. All are described below, with the TRNSYS routines labeled by "TYPE" according to the TRNSYS nomenclature.

TYPE 31 is the heat pipe backup tank model. The user has the option of reading in a performance map or specifying a constant firing efficiency. In many cases, the latter can result in saving computer time without significantly affecting the results. The tank model assumes a fully mixed tank.

TYPE 32 is the traced coil performance map reader. It is used in conjunction with the standard TRNSYS TYPE 34 tank model.

TYPE 33 is a special routine for processing load profiles and modeling the mixing value used to insure that the temperature of the water leaving the demand tank does not exceed the set point.

TYPE 35 is the connecting tube heat transfer routine.

FIGURE 2.12 CROSS SECTION OF TRACED COIL

DESIGN is a general purpose, user oriented code used here to solve the traced coil equations by the Newton-Raphson method. The general purpose part of the program consists of main program DESIGN and subroutines SIMBQS, GRAD and LINEQF. DESIGN was developed by Suh [32], and was modified slightly on this project to facilitate generating a performance map.

YCOMP is a user supplied routine defining the set of traced coil equations to be solved in DESIGN.

SOLVEF solves the traced coil inner problem (as discussed in Appendix D) using Newton's method.

BLOCKDATA and DESPEC define the basic traced coil parameters, from which SPCALC performs initial calculations of physical dimensions (e.g., surface area, tube spacing, etc.) for use in DESIGN.

2.2.3 System Simulation Results

2.2.3.1 Connecting Tube Performance

The modified TRNSYS program was first used to investigate the performance characteristics of the stacked tank arrangement. The primary objective was to determine what, if any, improvement in the fraction of the total load supplied by solar would result from the close coupling of the backup and solar preheat storage. As discussed in Section 1, with the proposed design, whenever the top of the solar tank becomes hotter than the demand tank, the connecting tube (thermal diode) will convect the "excess" heat up to the demand tank, but will prevent the reverse from occurring when the solar tank is cooler. It was believed that this would improve the solar fraction, relative to the same system with uncoupled tanks, for two reasons. First, during periods of high collection, it would increase the effective solar storage, and in turn, the amount of solar energy collected (assuming that a portion of the time the temperature at the top of the solar tank exceeded the demand tank temperature). Second, water preheated in the solar tank to temperatures exceeding the demand tank set temperature would be directly available to meet the load, and upon exiting the demand tank, could be mixed down to the set temperature with a tempering valve, thereby reducing the volume of gas heated water used.

The investigation of the coupled tank concept involved both computer

and experimental work, as alluded to in Section 2.2.1.2. While the computer effort was intended to investigate the effect of coupling on system performance, the experimental effort was designed to determine how closely the two tanks could be coupled, and what type of connecting tube hardware would be required. In addition, the experimental work led to the correlation upon which the computer model was based.

Two sets of simulation were run, one for a standard uncoupled tank system, and another for the same system with the tanks stacked. The stacked tank system incorporated the split connecting tube with "offset" that resulted from the experimental work described in Section 3.1.1. With this arrangement, the demand tank will track the solar tank within a few degrees whenever the latter is hotter.

Initially, both systems were modeled with 65 square feet of single glazed collector with a flat black surface, an 80 gallon preheat tank, a 25 gallon demand tank, a 50,000 Btu/hr firing rate, and a 130°F thermostat set temperature (also the set temperature of the tempering valve). In all of the simulations, collector tilt was set equal to the latitude. Both tanks had loss coefficients of 0.073 Btu/hr ft² °F. The traced coil consisted of five turns of 0.394 inch diameter copper tubing soldered to the bottom 20 inches of the preheat tank wall. Both systems also included a relief valve in the collector loop set at 212°F. Albuquerque was the first location simulated, and the ground water temperature there was assumed to be 72°F year round.

In order to develop a daily hot water demand profile, an appropriate application had to be selected. It was assumed that the heater was to be used by a family of 3 to 5 members living in a three bedroom, 1 1/2 bath house with laundry equipment. There is a significant amount of data available concerning the volume of water typically used in a residential setting for functions such as dishwashing, showering and food preparation; the temperature at which it will satisfy the majority of users; and the time of day it is called for [33,34,35,36,37]. Based on these data, and the hypothetical family described, the daily load profile shown in Figure 2.13 was constructed. It consisted of a total load of 100 gallons per day, including 15 for a dishwasher, 30 for showers or baths, 10 for sinks, and 45 for a clothes washer. (See Section 4.1.2 for a further discussion of loads and "sizing".)

When coupled and uncoupled systems were simulated for a one year period under the conditions described above, no significant difference was found between the performance results of the two. Both systems met the entire load called for (18.6 million Btu), and used the identical

FIGURE 2.13 DAILY LOAD PROFILE

amount of gas to do it (0.93 million Btu). Both had a solar fraction of 95 percent and average collection efficiency of 47 percent.

The fact that the coupling of the tank did not appear to affect the solar fraction was quite surprising, especially since during the course of the year the solar tank was hot enough for the split connecting tube to convect 2.8 million Btu up to the top tank.

A number of system parameters were changed in an attempt to determine under what conditions tank coupling would impact system performance. First, it was thought that the collector was oversized; the effect of coupling was being masked simply because the solar fraction was so close to 100 percent. The initial choice of 65 square feet of collector was high for the assumed load in Albuquerque, primarily because the year round ground water temperature was so high (72°F).

The year long simulations were rerun with 35 square feet of collector, but again the solar fractions were the same for both systems: 75 percent. However, under these conditions the solar tank appeared to be oversized relative to the collector. This was evidenced by the fact that in the coupled tank system, the solar tank was seldom hot enough to allow the connecting tube to operate, since only 0.7 million Btu were convected up to the demand tank. A set of simulations were then run with the same 35 square feet of collector, but with 55 gallons of storage instead of 80. As expected, the solar fraction dropped slightly for both systems, and the energy convected by the connecting tube increased for the stacked tank system. However, the two systems still yielded almost identical performance.

The next system parameter investigated was the collector loss coefficient. The larger the collector loss coefficient, the more sensitive collection efficiency is to operating temperature, and in turn, storage temperature. Consequently, the effect of tank coupling should be more pronounced as the collector loss coefficient increases. To investigate this effect, simulations were run with the collector loss coefficient doubled. Again, this resulted in the expected drop in solar fraction for both systems, but no significant difference was observed between the two.

Location was the next factor tested, under the assumption that weather variability could affect the impact of tank coupling. Because it differs dramatically from Albuquerque, Madison, Wisconsin was used to drive a set of simulations for the system with 35 square feet of high loss collector, and a 55 gallon solar preheat tank. Still no impact on performance could be attributed to tank coupling.

The final parameter investigated was the effect of load distribution. The original load profile was clustered around meal times, and had a 10 a.m. laundry peak. In terms of thermal coupling, it was thought that it could be more advantageous to have the largest draw at the end of the period of maximum collection instead of at the beginning. This was facilitated by moving the 10 a.m., 30 gallon laundry peak, illustrated in Figure 2.3, to 4 p.m. Simulations were driven with this new load, for the 35 square foot system in Albuquerque, but again, no advantage was observed for the coupled tank system.

At this point, it was apparent that any improvement in solar fraction associated with the coupling of the demand and solar tanks could not be detected using the current models and operating assumptions. However, because of simplifying assumptions in the initial modeling exercise, it was felt that further work would be required before concluding that tank coupling has no impact on solar fraction. First, as discussed in Appendices A and D, the two tanks are assumed to be fully mixed, so that the connecting split tube will not function until the entire solar tank is heated to the demand tank set temperature. Obviously, this is a very conservative assumption, since there will be many instances where the solar tank will stratify and only the top portion will exceed the demand tank temperature before the thermal diode begins to function.

The second restrictive assumption is that the hot water demand shows no day-to-day variations. In residential applications consumption varies in a random fashion, based on occasional vacations or weekend trips, on dishwasher and washing machine loads, on guests, and on baths or showers. There is only one paper of any significance in which this has been considered, a paper by Buckles, Klein and Duffie on "Analysis of Solar Water Heating Systems" in the Atlanta (1979) ISES Proceedings, pp. 959-963, in which weekly consumption was kept constant and day-to-day variations were based on educated guesses. A comparison was made between a system for which the daily consumption was always equal, and one for which the total weekly draw was the same but the day-to-day draw varied by a factor of 15 from the smallest to the largest. They concluded that "the effect of daily load distribution was larger than that of any of the design variables (except collector area) studied."

The effect of these daily consumption variations is quite real. At the Electric Power Research Institute (EPRI), Mr. Gary Purcell (who is in charge of the EPRI SHACOB work) installed a solar DHW system with electrical backup in one of the EPRI office buildings. The heaviest electrical consumption is on weekends, when no hot water is used.

Under these conditions, since there is no thermal conductance between the main tank and the backup tank, all the backup tank heat losses have to be made up by electric power even though the main tank is full of very hot water.

Clearly, modeling work beyond the scope of this program is required in order to incorporate stratification into the tanks (which is complicated by the unique heat transfer mechanisms of the heat pipe and traced coil), and to investigate the effects of more realistic demand profiles including day-to-day, and possibly annual variations. At this writing, a follow-on program including these tasks, is already in progress.

2.2.3.2 Optimum Tank Size and Firing Rate

The TRNSYS system simulation model was also used to examine the transient response of the backup portion of the system to peak demand conditions. This information was useful in selecting the demand tank volume and firing rate. As discussed in detail in Section 4.1.2, these two design parameters are selected in combination to yield the desired system capacity, commonly measured in terms of the "Hot Water Supply Rating" or "one hour draw". The accepted sizing approach for conventional systems is to determine the maximum hot water use in any one hour, and to select an appropriate system capacity to match this demand [36,37]. Obviously, the proposed heater is not a conventional system, but for the purpose of sizing the backup portion of the system, a worst case approach was taken (i.e., no solar preheat available), and demand tank size and firing rate selected accordingly.

Two worst case demand scenarios were constructed; a 20 minute early morning peak, and a 30 minute laundry peak. These were used to drive short-term simulations of the backup portion of the system for a range of tank volumes, firing rates, and thermostat set temperatures. A sample output from one of these simulations is illustrated in Figure 2.14. Here, the backup tank volume is 25 gallons, the thermostat set temperature 130°F, and the firing rate 50,000 Btu/hr. Shown plotted against time are the load profile, demand tank temperature, and the temperature "delivered" to the load.

FIGURE 2.14 TRANSIENT RESPONSE OF BACKUP SYSTEM TO WORST CASE LOADS

Section 3

EXPERIMENTAL WORK

There were two phases of experimental work in this program: experiments designed to characterize the performance of individual components (as part of tasks b, c, d, e, and g), and performance tests of the final prototype (task l). The component experiments constituted the heart of the program and were designed to verify the computer design tools (as discussed in Section 2), to supply heat transfer data where none were found in the literature, and, through what was, in effect, a trial and error process, to arrive at safe, reliable, and efficient component designs.

Section 3 describes the five component experiments performed under this program. Section 3.1 describes the testing of the various tank connecting tube designs. Extensive testing of the initial or "first cut" heat pipe design is discussed in Section 3.2. Sections 3.3 and 3.4 deal with the experimental development efforts to improve the efficiency and reliability of the heat pipe and burner/fire box respectively. An experiment to investigate stratification in the demand tank is described in Section 3.5. Testing of the final prototype is described in Section 4.

3.1 CONNECTING TUBE EXPERIMENT

The connection between the top (demand) and bottom (solar preheat) tanks should be a thermal diode, which transfers heat well in the upward direction, but not in the downward direction. If this is achieved, there will not be a significant use of gas (by downward heat flow) to heat the lower tank, which should be heated by solar energy. In addition, if there is enough solar energy so that the solar tank temperature exceeds the temperature of the backup tank, a properly designed thermal diode can transfer enough solar heat to the upper tank so that the tanks become closely coupled.

In a solar heating system the temperature of storage will typically rise at the rate of about 10°F/hr under good conditions. With a stacked tank arrangement the upper tank will lag behind the lower tank in temperature. The larger the temperature lag, the hotter the preheat tank, and in turn, the lower the collection efficiency. Hence, it is desirable to minimize the temperature lag.

It is simple to inhibit downward flow of heat. Water is a poor conductor and it is quite easy to obtain stratification even in connectors (between the tanks) of large cross sectional area. It is not so easy to promote the upward flow of heat. Hence, the primary objective of this experiment was to characterize the thermal and fluid mechanical performance of the connecting tube so that it could be optimized with regard to the resulting temperature lag.

The arrangement used to characterize the connecting tube is shown in Figure 3.1. Two well insulated "piggy back" steel tanks were used. The lower tank had an 80 gallon capacity and the upper tank a 20 gallon capacity. Solar input to the lower tank was simulated with a 4 inch wide electric band heater which wrapped tightly around the bottom of the tank. Voltage was supplied to the heater through a variable transformer so that input power to the tank could be varied from 0 to 4000 Watts. The tanks were constructed so that various connecting tubes could be installed with diameters up to 3 1/2 inches.

A diagram showing thermocouple construction and placement is shown in Figure 3.2. The lower tank was instrumented with three levels of thermocouples, each level averaging four temperatures. The upper tank had two levels of thermocouples. Thermocouple outputs were wired through an electronic 32°F reference junction and a signal scanner, and were plotted out with a strip chart recorder. This gave a hard copy record of millivolt thermocouple output vs. time which could be converted to temperature with a resolution of < 0.1°F.

When measuring heat transfer from the bottom tank to the top tank, both tanks began at the same uniform temperature. Heat was then supplied to the bottom tank through the band heater and temperatures in both tanks monitored and recorded. As the tanks heated up, steady state temperature differences developed between the tank levels. Heat transfer between the tanks was determined from the rate of temperature rise of the top tank as follows:

$$Q = Mc_p \frac{dT}{d\tau} \quad (3.1)$$

where

Q = heat transfer rate [Btu/hr]

M = mass of top tank [lb]

c_p = average specific heat of top tank [Btu/lb°F]

$\frac{dT}{d\tau}$ = rate of temperature rise [°F/hr]

Figure 3.1 Connecting Tube (Thermal Diode) Experimental Apparatus

FIGURE 3.2a THERMOCOUPLE PLACEMENT

FIGURE 3.2b THERMOCOUPLE CONSTRUCTION

3.1.1 Results

Three types of diode geometries were tested; single tube, split tube, and split tube with offset. These are illustrated in Figure 3.3. A typical thermal diode test is shown in Figure 3.4. A series of runs was made for each geometry and these are summarized in Table 3.1. Temperature lags between the upper and lower tanks at specific rates of temperature rise are given, along with the associated degree of stratification of the upper tank (i.e. the temperature difference between the two thermocouple levels). Note that the temperature lag shown here is the difference between the temperature at the top level of the bottom tank and the bottom level of the top tank.

These results show that a split tube considerably reduces the temperature lag between the tanks (increases the flow rate) for a given rate of temperature rise (heat transfer rate) compared to a single tube. This is caused by the presence of two isolated flow paths; a downward flowing cold path and an upward flowing hot path, which impose less restriction than the case where two streams are "flowing through" each other. Adding offset to the split tube further decreases the temperature lag. Offset in the tubes prevents fluid exiting from one tube from mixing with fluid near the entrance of the other tube. Offset in the tube also increases stratification in the upper tank. With a single tube and a split tube without offset, the upper tank remains fairly well mixed. Stratification enhances the performance of the thermal diode since it increases the buoyancy driving force which circulates fluid between the two tanks.

Figures 3.5 and 3.6 illustrate the difference in the transient response between a single straight tube, and a split tube with offset (the lines labeled with two temperatures represent the average of the two thermocouple locations). Here, the advantage of a split tube with offset is quite apparent.

In Figures 3.5 and 3.6 the temperature in neither tank is shown rising above 80°F, well below the temperature anticipated in an actual system. However, this does not affect the temperature lag results. As illustrated in Figure 3.4, once a steady state lag is initiated, it remains constant beyond the temperature which would be encountered in an actual system. Runs were made as high as 140°F with no significant changes in the slope of the temperature curves. In all runs, temperatures were taken to about 100°F, and once the preliminary transient was passed no significant changes were observed.

SINGLE
TUBE

SPLIT
TUBE

SPLIT TUBE
WITH OFFSET

FIGURE 3.3 CONNECTING TUBE GEOMETRY

FIGURE 3.4 TYPICAL THERMAL DIODE TEST RESULT

FIGURE 3.5 TANK TEMPERATURES DURING SOLAR HEATING WITHOUT CONVECTION ASSEMBLY

FIGURE 3.6 TANK TEMPERATURES DURING SOLAR HEATING WITH CONVECTION ASSEMBLY

The length and diameter of the tubes also affect heat transfer. With a single tube, increasing the length reduces heat transfer because of increased flow resistance. With a split tube, increasing tube length decreases temperature lag because the increased flow resistance has less effect than the increase in buoyancy driving force. Diameter increases improve performance for either split or single tube diodes. The results also show that, for a given diode construction, temperature lag increases as the rate of temperature rise increases.

As discussed in Section 2.2.2, in order to perform computer system simulations, a mathematical description of connecting tube heat transfer was required. Based upon the form of the analytical derivation (Section 2.2.1.2) and plots of the experimental results listed in Table 3.1, the following simple relationship was found to satisfactorily model connecting tube heat transfer:

$$Q = K\Delta T^{1.5} \quad (3.2)$$

where

Q = heat transfer between tanks, Btu/hr
 K = a constant characteristic of geometry
 ΔT = temperature difference between tanks, $^{\circ}\text{F}$

The constant K was determined graphically from the plots of experimental results for the geometries of interest. The values for several representative tubes are shown in Table 3.2.

TABLE 3.2

K for Representative Tube Geometries

<u>Tube Dimensions</u>	<u>K [Btu/hr $^{\circ}\text{F}^{1.5}$]</u>
3" ID x 8" Long x 2" Offset	325
2" ID x 10" Long x 2" Offset	105
1" ID x 16" Long x 6" Offset	29

TABLE 3.1

Thermal Diode Test Results

	Diode Geometry	Rate of Temp. Rise °F/HR	Temp. Lag °F	Stratification of top tank °F
1.	3 1/2" ID x 4" long ⁺ Single Tube	11.4	10.2	<.5
2.	3 1/2" ID x 8" long Single Tube	12.5	13.4	<.5
3.	3 1/2" ID x 8" long Single Tube	21.2	15.6	<.5
4.	3 1/2" ID x 8" long Single Tube	9.6	10.5	<.5
5.	3 1/2" ID x 8" long Single Tube	12.3	12.2	<.5
6.	3 1/2" ID x 9" long Split Tube-No Offset	9.7	4.7	<.5
7.	3" ID x 9" long Split Tube-No Offset	12.3	5.1	<.5
8.	3" ID x 28" long Split Tube 12" Offset	9.8	2.0	1.3
9.	3" ID x 16" long Split Tube 6" Offset	9.9	2.6	1.3
10.	3" ID x 8" long Split Tube 2" Offset	11.9	3.3	.8
11.	3" ID x 8" long Split Tube 2" Offset	9.4	2.8	0.9
12.	3" ID x 8" long Split Tube 2" Offset	4.35	1.9	0.5
13.	2" ID x 28" long Split Tube 12" Offset	9.4	4.2	3
14.	2" ID x 16" long Split Tube 6" Offset	13.3	6.05	3
15.	2" ID x 16" long Split tube 6" Offset	9.5	4.7	2.5

TABLE 3.1 (continued)

	Diode Geometry	Rate of Temp. Rise °F/HR	Temp. Lag °F	Stratification of top tank °F
16.	2" ID x 10" long Split tube 6" Offset	5.8	3.4	2
17.	2" ID x 10" long Split tube 3" Offset	13.8	6.9	2
18.	2" ID x 10" long Split tube 3" Offset	9	5.4	1.5
19.	2" ID x 10" long Split tube 3" Offset	5.3	3.7	1.3
20.	1" ID x 28" long Split Tube 12" Offset	11.5	17.1	12*
21.	1" ID x 28" long Split Tube 12" Offset	7.5	14.4	10.5**
22.	1" ID x 28" long Split Tube 12" Offset	4.4	8.7	6.1
23.	1" ID x 16" long Split Tube 6" Offset	12.3	16.9	6.7***
24.	1" ID x 16" long Split Tube 6" Offset	8.2	12.7	5.3
25.	1" ID x 16" long Split Tube 6" Offset	5.5	9.3	4.3

* Note: 1.5° Stratification in lower tank

** Note: 1° Stratification in lower tank

*** Note: 1.6° Stratification in lower tank

+ The "long" dimension represents the overall tube length including offsets

3.2 TESTING OF INITIAL HEAT PIPE DESIGN

The primary thermal performance criteria for the heat pipe heater called for a firing rate of 50,000 to 70,000 Btu/hr (based on the higher heating value of the fuel burned), a firing efficiency of greater than 80 percent, and a subatmospheric operating pressure. With the aid of the computer model described in Section 2.1, an initial or "first cut" heat pipe design was developed, and a fully instrumented bench prototype constructed. With this prototype it was possible to observe the operating characteristics of the proposed design, to identify and resolve any unexpected problems, and to verify the accuracy of the computer model. In addition, the task of assembling the prototype resulted in valuable fabrication experience.

The first prototype bench set up is illustrated in Figure 3.7. It consisted of an evaporator enclosed in an insulated steel combustion chamber, a single "adiabatic" connecting tube, and a well insulated, 20 gallon tank with condenser flow passages soldered to the outer wall. The entire system was made of copper and used water as the working fluid. The evaporator was constructed of six, 8 inch finned tubes (5/8 inch ID, 1.5 inch fin diameter, and 7 fins per inch) manifolded at one end with a 1 inch pipe. The tank was 18 inches in diameter, 18 inches tall, and had 16, 3/4 inch half round or "D-tubes" soft soldered to the wall and manifold at the top and bottom with 1 inch pipe*.

The top condenser header had an extended nipple with a valve and pressure gauge, so that any desired vacuum could be drawn on the heat pipe flow passages, and the operating pressure monitored. The first burner consisted of four, 10,000 Btu/hr "tips" evenly distributed below the evaporator. In addition to the pressure gauge, the prototype was instrumented with a bellows type gas meter, and thermocouples on the fin tips and inside of the tank.

*This demand tank/condenser arrangement was the most convenient way to simulate the condensing passages which would be available with a jacketed tank.

INITIAL BENCH PROTOTYPE HEAT PIPE WATER HEATER TEST SET UP

FIGURE 3.7

3.2.1 Results

Initial attempts to run the prototype were unsuccessful due to severe overheating at the evaporator, as evidenced by the fin tip thermocouples. It was obvious that a stable heat pipe cycle was not functioning. After a long trial and error process, it was found that the system would function if the firing rate were low enough and/or the operating pressure high enough.

In order to investigate the relationship between overheating and operating pressure, a series of tests were run in which the pressure was varied while holding the firing rate constant. Pressure was controlled simply by adjusting the water level in the tank. The results are plotted in Figure 3.8. They show that for any given firing rate there is a critical operating pressure below which the evaporator overheats, and above which the evaporator fin temperature remains only slightly above the saturation temperature of the boiling heat pipe fluid (i.e., the saturation temperature corresponding to the operating pressure).

At this point it was apparent that something was occurring at the critical pressure to cause parts of the evaporator to run dry. It was also apparent that any one of several phenomena could be responsible. The first to be examined was the volume of working fluid with which the heat pipe was charged. If too much of the charge was being held up in the condensing film and returning condensate, then portions of the evaporator could easily run dry and overheat. To check this, the prototype was run several times with different volumes of working fluid. No significant improvement resulted.

Flame distribution was examined next. With combustion occurring at four point sources (i.e., four burner tips), it was thought that hot spots were developing along the evaporator, possibly causing a jump from nucleate to film boiling inside of the tubes, with a corresponding jump in temperature. A new burner was designed and built in an attempt to achieve more even flame distribution. It consisted of six, 1/2 inch diameter, 8 inch long pipes manifolded and laid out in the same arrangement as the evaporator itself. Each pipe had two rows of 0.063 inch ports spaced at 1/4 inch intervals. With this arrangement, a more even flame distribution was achieved. Unfortunately, it did not alleviate the overheating problem.

Another possible cause of evaporator overheating investigated was dryout due not to an inadequate charge, but to entrainment of the

FIGURE 3.8 EVAPORATOR OVERHEATING AS A FUNCTION OF OPERATING PRESSURE

returning condensate. Under certain conditions, the velocity of the vapor flowing over the returning condensate could produce a shear force at the liquid-vapor interface large enough to pull droplets of liquid into the vapor stream. This was consistent with the results shown in Figure 3.8. For a given firing rate, as the operating pressure was lowered, the vapor density would decrease, and the vapor velocity would increase. At some point, apparently corresponding to the critical pressure identified in Figure 3.8, the velocity would reach a point where significant entrainment would occur, and in turn, the evaporator would dry out and overheat.

Correlations are available for predicting the conditions under which entrainment will occur with a gas flowing over a body of liquid, and in fact, an empirical correlation developed by Steen [16] was incorporated into the heat pipe model (see Section A.2.2). This correlation did not predict the onset of entrainment for the flow conditions present in this application. However, the correlation applies to straight runs of duct, and does not account for the bends and other restrictions present in the heat pipe.

In order to investigate the entrainment theory, an entirely new evaporator and connecting tube were constructed. The size of the finned tubes was increased from 5/8 to 7/8 inch ID and the header tube was increased from 1 to 1 1/2 inches. These changes resulted in a significant improvement as illustrated in Figure 3.9, where critical pressure is plotted as a function of firing rate for both the original and the enlarged evaporator. (The shaded area represents the range of operating points experienced during the many runs with the original evaporator.) The data show that the enlarged flow passages, and hence lower vapor velocities, resulted in roughly twice the firing rate for any given critical pressure. However, even with this system, overheating occurred at firing rates greater than 30,000 Btu/hr.

These results were extremely encouraging, but the existing hardware was still far from satisfying the design criteria. Further changes were necessary if a heat pipe capable of firing at 50,000 to 70,000 Btu/hr and subatmospheric pressure was to evolve.

Confident now that the major cause of evaporator overheating was entrainment, and that very large flow passages would be required to overcome it, it was decided to attempt to take advantage of this phenomena instead of trying to design around it. A new arrangement was conceived which completely segregated the vapor and returning condensate through the use of a separate condensate return tube. This

FIGURE 3.9 CRITICAL PRESSURE VS. FIRING RATE FOR ORIGINAL AND IMPROVED EVAPORATOR

arrangement is illustrated in Figure 3.10. The vapor tube was connected to the top condenser header, and the condensate tube to the bottom header, thereby creating a complete loop. Counterflow between the vapor and condensate was replaced by parallel flow. It was predicted that this would actually enhance the return flow of condensate, and lead to a slight, although probably insignificant, improvement in condensation heat transfer.

Initial tests of the "two pipe" system were extremely successful. It was tested at firing rates of up to 70,000 Btu/hr with no overheating and an operating pressure of only 10 psia. During subsequent testing, some overheating was encountered, but this was eliminated by adjusting the fluid charge and the tilt of the evaporator tubes. The latter was particularly significant. The most reliable evaporator performance was achieved by tilting the evaporator roughly $1\frac{1}{2}$ degrees towards the condensate return header. This evidently counteracted the shear effects of the vapor flow and insured an even distribution of fluid throughout the evaporator.

As shown in Figure 3.10, a glass section (sight tube) was inserted into the condensate return leg. During operation, a column of standing water was observed in this leg and gave a direct reading of the pressure drop through the system. For the prototype operating in the 50,000 to 70,000 Btu/hr range, this was typically 10 to 15 inches of water.

3.3 PERFORMANCE IMPROVEMENT TESTING

Two of the three primary design criteria for the heat pipe were met with the successful operation of the two pipe system; firing rate and operating pressure. The remainder of the experimental work concerning the heat pipe concentrated on the final criteria, firing efficiency, and on several other important performance considerations; cycling, startup transients, and combustion air supply.

The test set up used to perform efficiency measurements is shown in Figure 3.11. It was simply the two pipe prototype described in the previous section, with additional temperature and flow measurement instrumentation. Efficiency measurements were taken under steady state conditions, which were achieved by running a continuous flow of water through the tank. By measuring the quantity and temperature rise of the water passing through the tank in a known time period, and also the volume of gas burned, the firing efficiency was calculated simply as follows:

FIGURE 3.10 MODIFICATION OF BENCH PROTOTYPE HEATER TO TWO-PIPE SYSTEM

FIGURE 3.11 BENCH PROTOTYPE WATER HEATER EFFICIENCY TEST SETUP

$$\eta = \frac{M c_p (T_{out} - T_{in})}{VH}$$

where

M = mass of water collected on scale during time period [lbm]

c_p = specific heat of water [Btu/lbm^oF]

T_{out} = temperature of water leaving tank [^oF]

T_{in} = temperature of water entering tank [^oF]

V = volume of gas burned during time period [ft³]

H = higher heating value of gas [Btu/ft³]

Tests were typically conducted over 15 minute time periods, with three temperature readings per period, usually commencing an hour and a half after initial firing in order to insure steady state conditions. According to the supplier, Pacific Gas and Electric, the heating value of natural gas used was 1,080 Btu/ft³.

3.3.1 Results

Initial efficiency measurements were much lower than predicted by the computer model, typically 60 to 65 percent instead of 75 to 80 percent. This was not surprising since it was observed that the walls of the combustion chamber surrounding the burner and evaporator were extremely hot, indicating significant losses. In addition, it appeared that there were significant radiant losses from the bottom of the combustion chamber.

Several design changes were made in an attempt to eliminate these losses. In order to lower the flame intensity directly adjacent to the chamber walls, two new burners, one for firing up to 35,000 Btu/hr and another up to 70,000 Btu/hr, were designed with much smaller overall flame areas than the original. The smaller of the new burners was constructed of 4 (instead of 6 like its predecessor), 1/2 diameter, 8 inch long pipes, manifolded at one end with a 1 inch header. The larger burner employed a close packed matrix consisting of 8, 10,000 Btu/hr burner "tips". In addition to changes in the burner, the

combustion chamber walls were heavily insulated, with 1 inch of ceramic on the inside, and 3 inches of fiberglass on the outside, and a reflective panel was installed beneath the chamber to reduce the radiant losses.

These changes proved extremely effective in reducing losses from the combustion site and, in turn, improving firing efficiency. Data for seven runs over a range of firing rates are listed in Table 3.3 and plotted in Figure 3.12. Firing efficiencies ranged from 86 percent at a firing rate of 16,600 Btu/hr, to 78.6 percent at 69,700.

The effect of combustion air supply on firing efficiency was investigated next. The combustion chamber was fitted with an air supply plenum through which all of the primary and secondary air was drawn. The plenum was attached an air compressor line and a rotometer so that the total air supply could be controlled and measured. The flue gas temperature was also measured.

A series of efficiency tests was performed with different air flow rates, while holding the firing rate and condenser temperature constant. The results are listed in Table 3.4 and are plotted, along with the computer predictions, in Figure 2.4 (Section 2.1.3.1). Peak efficiency was achieved with an air flow rate of 11 CFM, which corresponds to approximately 150 percent theoretical air (see Section A.3 for definition), and a flue temperature of 245°F.

As discussed in Section 2.1.3.1, the most interesting aspect of these results is how little efficiency is affected by varying the combustion air supply. According to the computer results, this is due to two counteracting effects; flame temperature and mass flow rate. Increasing the percent theoretical air decreases the flame temperature, but at the same time increases the combustion products flow rate, and hence, the convection heat transfer coefficient on the outer surface of the finned tubes. The result is a relatively constant heat transfer rate over a range of fuel to air mixtures.

For comparison, a test was also performed with the plenum disconnected and the burner normally aspirated, as would be the case during actual operation. Under these conditions, the efficiency was slightly higher than the maximum achieved with the air plenum; 83.2 percent with a flue temperature of 247°F. This implied that the burner/combustion chamber configuration was indeed near optimum in terms of combustion air supply.

FIGURE 3.12 HEAT PIPE EFFICIENCY VS. FIRING RATE FOR FINAL BENCH SETUP

TABLE 3.3

Efficiency Test Results

Run #	Q_{in} Btu/hr	Q_{out} Btu/hr	η -	$\dot{m}$ lb/hr	T_{fin} °F	T_{flue} °F	T_{in} °F	T_{out} °F	T_{vapor} °F
1	16673	14372	86.2	503.8	139.6	135.2	61.2	89.7	123.6
2	24430	20667	84.6	501.1	157.8	155.6	60.7	102.0	143.0
3	32808	26706	81.4	500.6	173.0	163.8	60.3	113.5	158.8
4	45161	36897	81.7	506.6	194.9	235.0	61.2	134.0	180.5
5	57403	46669	81.3	512.6	214.2	244.4	61.5	152.6	198.4
6	69732	54809	78.6	748.8	215.1	369.0	60.1	133.3	200.4

Nomenclature

Q_{in}	=	firing rate
Q_{out}	=	energy transferred to tank
η	=	firing efficiency
$\dot{m}$	=	mass flow of heat pipe working fluid
T_{fin}	=	average fin tip temperature
T_{flue}	=	temperature of flue gases
T_{in}	=	temperature of water entering bottom of tank from water main
T_{out}	=	temperature of water leaving top of tank
T_{vapor}	=	temperature of heat pipe vapor

TABLE 3.4

Combustion Air Supply Test Data

Air CFM	Gas CFM	Percent Theoretical Air	Fin Temp	Flue Temp	Efficiency
10	.729	116	197	237	80.9
9	.726	130	195	224	79.9
8	.721	144	190	238	73.2
11	.729	158	197	245	82.3
12	.729	173	197	253	81.6
13	.730	187	197	263	80.7
Normal Aspiration	.730	-	198	247	83.2

At this point in the program, all testing had been performed under steady state conditons, so potential problems with cycling or startup transients had yet to be addressed. Hence, a test program was undertaken to examine transient performance characteristics. The heat pipe prototype was operated over a wide range of temperatures and through a variety of cycling sequences. As discussed below, several problems were diagnosed and corrected.

The first malfunction occurred when the heat pipe underwent a quick cycling sequence. When the burner was shut down after achieving steady state, the water in the evaporator flashed to vapor and took 15 to 20 seconds to recondense. Severe overheating would occur if the burner was fired again before a sufficient quantity of the vapor had recondensed and wetted the evaporator. This is obviously not a very realistic operating sequence, since the thermostat deadband would normally not allow the burner to cycle so quickly. However, in order to make the system as fail-safe as possible, it was decided to build a time delay into the ignition system. This was actually quite simple because the intermittent ignition system chosen came with a 30 second "reset time" built in.

Another condition conducive to malfunctioning was found to occur when a cold evaporator assembly was initially fired into a well mixed hot tank. The evaporator had a tendency to dryout and overheat, in most cases not recovering sufficiently to establish a stable heat pipe cycle. This was especially true at higher firing rates.

The cause of this problem was eventually traced to condenser "priming"; the time lag associated with the establishment of an effective condensation process and condensate return flow. Under the conditions of an initially cold evaporator and a well mixed hot tank, condenser pressure was low, condensation less effective, and vapor velocities also relatively low. This in turn lead to lower vapor-liquid shear, which in a parallel flow system, enhances condensate return.

The problem was entirely solved with two relatively minor design changes. The working fluid charge was increased to 350 cc to insure a sufficient water reservior throughout the startup transient, and the condenser itself was tilted 4 degrees towards the condensate return tube in order to accelerate the "collection" and return of condensate. In lieu of these modifications, it was found that a two stage ignition sequence could also be used to eliminate startup overheating. This

involves firing at a lower rate until the condensate return is established (on the order of 30 seconds at 20,000 Btu/hr), and then raising the firing rate to the desired level (50,000 to 70,000 Btu/hr). However, adjusting fluid level and condenser tilt were so effective, a two stage ignition was not deemed necessary [39].

3.4 BURNER/COMBUSTION CHAMBER DEVELOPMENT

The burner and combustion chamber that evolved from the experiments described in the previous section were quite adequate when evaluated strictly in terms of thermal performance. However, with 3 inches of fiberglass insulation, a 1 inch ceramic lining, and a reflective panel, it was hardly the most optimum arrangement in terms of fabrication and materials costs. Clearly a more streamlined design was required; one that required minimal insulation and was compact in size.

The development of a more practical configuration began with the design of yet another burner. It was apparent that the flame pattern itself would have to be shortened and more well defined, so as not to spread past the periphery of the evaporator. This would minimize the flame intensity at the combustion chamber walls, and also reduce the amount of wall area exposed by reducing the distance between the burner and heat exchanger required to avoid quenching the flame.

It was discovered that a more well defined flame could be achieved by using a dense pattern of small, independent ports. A new burner was constructed as illustrated in Figure 3.13. It consisted of 13, 7 1/2 inch long, 3/8 inch diameter tubes manifolded at one end with a 1 1/2 inch header, and fit with a 1 1/4 inch venturi mixer designed to supply 50,000 Btu/hr of gas at 5 inches water column gas pressure. The burner tubes had 13, 0.081 inch diameter ports drilled on 1/2 inch centers covering 6 inches of each tube. The result was a 6 inch square flame pattern consisting of 169 independent flames, each approximately 2 inches high. The combustion chamber was constructed of 18 gauge steel, and was designed to hold the burner in place beneath the evaporator, and a new reflecting panel in place beneath the burner.

When tested, the new burner and combustion chamber resulted in a firing efficiency of 81.5 percent with only 1 inch of ceramic fiber insulation on the outer wall, and 79 percent with no insulation at all.

3.5 DEMAND TANK STRATIFICATION TEST

A final component experiment was conducted to determine the degree of

FIGURE 3.13 NEW BURNER CONFIGURATION

stratification occurring in the demand tank due to heat input through the walls (as opposed to heat convected up from the preheat tank). It was felt that these data could be used to determine the placement of the thermostat, and also the optimum length of the offset section of the tank connecting tube.

The bench prototype tank was instrumented with a three dimensional array of thermocouples, placed at five distinct levels. At the top, bottom and middle levels, thermocouples were placed at three radial positions. At the two intermediate levels, thermocouples were only positioned midway between the axis and wall of the tank.

Starting with a fully mixed cold tank, the prototype was fired at 50,000 Btu/hr. The three dimensional temperature readings revealed that essentially no radial stratification developed. However, there was significant axial stratification, on the order of 90°F from the top of the tank to the bottom with the average "bulk" temperature occurring about one-third of the way from the bottom. The degree of stratification was roughly equal to the temperature difference between the wall and the water. This indicated that stratification could be reduced by reducing the wall temperature (increasing effective heat transfer area). This was deemed desirable since excessive stratification complicates control and does not make the most effective use of the tank capacity. These results also indicate that the upper leg of the split tube connecting diode should be relatively short, since the water at the top of the solar tank will be hotter than the water at the bottom of the demand tank more often than that at the top.

Section 4

FINAL DESIGN, FABRICATION AND TESTING

Section 1 described how the tasks identified for this project fell into three distinct phases: computer modeling; experimental work; and final design, fabrication, and testing. Completion of the first two phases resulted in a thorough understanding of the performance characteristics of each of the key heater components, and the design features that would be required to produce safe, efficient, and reliable hardware. What remained was to synthesize this performance data and first hand fabrication and operating experience into a complete engineering prototype.

The result of the final design effort is shown schematically in Figure 4.1, and the result of the final fabrication effort is shown in a series of pictures included as Figure 4.2. The remainder of this section will describe, on a component by component basis, the key features of the engineering prototype, and any of the design considerations not discussed thus far. For convenience, the design and fabrication phases are described together in Section 4.1. The final performance checkout is the subject of Section 4.2.

4.1 FINAL DESIGN AND FABRICATION

4.1.1 Materials and Fluids

Before proceeding with the description of the various components, the rationale behind the selection of the heat pipe fluid and materials used in the prototype will be discussed. Copper and water were chosen for the heat pipe/demand tank subsystem, and indeed, were the only combination tested in the experimental work described in the previous section. However, a number of other materials and fluids were considered.

Selection of an appropriate heat pipe fluid was dependent largely upon operating temperature. The melting/freezing point of the fluid must be below the minimum operating temperature, and likewise, the fluid's critical temperature must be well above the maximum operating temperature so that the vapor pressure of the fluid remains moderate. For this application the heat pipe fluid operating pressure range was

ENGINEERING PROTOTYPE DESIGN SCHEMATIC

COMPLETELY ASSEMBLED HEATER

ENCLOSURE PARTIALLY REMOVED

COMBUSTION CHAMBER, BURNER,
IGNITION AND CONTROLS

FINNED TUBE EVAPORATOR

FIGURE 4.2

assumed to roughly 50°F to 250°F. Fluids which were considered included water, ammonia, methanol, and R-11. These fluids are listed in Table 4.1 along with their pertinent properties.

Note that ammonia, a commonly used heat pipe fluid, has a critical temperature of 271°F which is very close to the 250°F maximum operating temperature. Even at 150°F, the vapor pressure of ammonia is about 350 psia. Because of the extreme pressures associated with ammonia and the high toxicity of ammonia vapor (UL rating of 2), it was ruled out as a potential fluid.

In terms of the required operating temperature range, water, methanol, and R-11 all appeared to be satisfactory candidates. The next important heat transfer characteristic evaluated was peak nucleate boiling heat flux. This is the maximum heat transfer rate per unit area which the fluid can sustain through boiling heat transfer before making a transition from nucleate to film boiling. This transition is accompanied by a sharp rise in evaporator wall temperature and overheating or "burn out".

Using Equation A.32 of Section A.1.3, peak heat fluxes were calculated for the candidate fluids at 150°F operating temperature and are shown below:

<u>Fluid</u>	<u>Peak Flux</u> Btu/hr - ft ²
water	273000
methanol	251000
R-11	155000

Upon examining the available finned tube geometries, and estimating the outer surface area required to transfer 50,000 to 70,000 Btu/hr, and also, the space restraints on the overall size of the evaporator, it was estimated that less than 1/2 square foot of boiling heat transfer area would be available with the proposed evaporator configuration. This meant that using R-11 would probably approach or exceed the peak nucleate boiling heat flux for that fluid under the anticipated operating conditions. In addition, with a UL rating of 5a, R-11 is highly toxic. Hence, it was ruled out as a potential fluid.

In addition to its heat transfer potential, methanol is extremely attractive in terms of its compatibility with materials. However, it is both toxic and combustible. Although it could not be ruled out as a potential candidate, it was felt that it was not within the scope of

TABLE 4.1

PROPERTIES OF POTENTIAL HEAT PIPE FLUIDS
(@ 150°F)

	Water (H ₂ O)	Methanol (CH ₃ OH)	Ammonia NH ₃	R-11 (CCl ₃ F)
Melting/Freezing Temperature [°F]	32	-144	-108	-170*
Critical Temperature [°F]	706	464	271	388
Saturation Pressure [PSIA]	3.7	15*	350*	52
Vapor Density [lb/ft ³]	0.0103	0.0756	1.67	1.21
Liquid Density [lb/ft ³]	61.2	49.8	33.5	85.7
Latent Heat of Vaporization [Btu/lb]	1008	473	423	70.3
Viscosity [lb/ft-hr]	0.0255	0.033	0.031	0.035

*Values Approximate

this program to address both the toxicity and combustibility of methanol, especially when considering the potential of water.

Water has excellent heat transfer characteristics. It is not toxic or combustible. It operates at a relatively low pressure and is inexpensive. However, there are two drawbacks to using water as a heat pipe fluid. First, water freezes at temperatures potentially reached by the heat pipe evaporator, which could damage fluid passages. Second, water is incompatible with many materials which might be used for the heat pipe container. Despite these shortcomings, it is felt that, with proper design precautions, water was the best choice. Following, freezing and compatibility problems are addressed.

Freezing can occur, since the heat pipe evaporator, by design, is isolated from the heated tank. If the water heater were located in an environment with an ambient temperature below 32°F, the water in the evaporator would freeze in a relatively short time. The freezing problem was overcome by designing the evaporator so that the water can freeze without causing damage. This was accomplished by simply allowing enough room in the evaporator tubes for the water to expand.

There are several basic failure modes caused by incompatibility between the heat pipe fluid and the container. The first is a reaction between the fluid and container which causes the fluid to break down. The second type of failure is caused by the fluid corroding the container wall and solid particles breaking away and collecting in the evaporator section, inhibiting heat transfer. The third is through generation of a noncondensable gas caused by chemical reaction between the fluid and container wall. A noncondensable gas so produced collects in portions of the condenser thereby preventing the vapor from condensing on that section.

There is much published information regarding compatibility of materials for water heat pipes (References 18, 19, 20, 21, 22, 23, 24, 25). The indications are that carbon steel and aluminum are too vulnerable to corrosion by water to function as water heat pipe jackets. Further, stainless steels and nickel have both been shown to generate hydrogen, a noncondensable gas, when used for water heat pipe containers. Materials which have been used successfully for water heat pipes are copper and monel. Copper is the most widely used material for low temperature water heat pipes and is indeed the logical choice. The relative expense of copper compared to carbon steel, which might be used with some fluid other than water, is felt to be outweighed by the reduced amount of material required, and by enabling the use of a nontoxic, noncombustible fluid.

4.1.2 Sizing

As discussed in Section 2.2, an important aspect of the design process at the system level was the proper "sizing" of the various components. In solar system design, the term is typically associated with selecting the most appropriate collector area and solar storage volume. Sizing of conventional gas or electric water heaters usually involves, if anything, selecting the most appropriate combination of tank volume and firing rate. In either case, appropriateness is evaluated in terms of both economics and thermal performance.

Proper sizing of the final prototype must really address all of these considerations; collector area, solar tank size, firing rate, and demand tank size. The first two of these (the solar side of the system) are very sensitive to a myriad of parameters, including location, fuel costs, and available tax credits, to name a few. Fortunately, for the purpose of designing the final prototype, a number of simplifying assumptions concerning the solar side of the system could be made without compromising the objectives of the program. Indeed, at this stage of the commercialization process, it was deemed sufficient to target the prototype for a single, well defined application, while at the same time, acquiring the tools and basic understanding necessary to design for others.

As discussed in Section 2.2, it was assumed that the prototype would be used by a family of 3 to 5 members living in a three bedroom, 1 1/2 bath house with laundry equipment included. Albuquerque, New Mexico, was chosen for the location, although Madison, Wisconsin, was also used in the computer simulations. Based upon data available concerning the volume and temperature of hot water typically used in a residential setting, [33,34,35,36,37], and the hypothetical family assumed, a daily load profile, and two worst case demand scenarios were constructed.

The daily load profile was used to drive the computer simulation and is described in Section 2.2 (see Figure 2.13).

The worst case demands consisted of an early morning peak and a heavy laundry day. The morning peak assumed a 3 gallon sink demand at 105°F, and two, five minute showers at 3 gpm and 105°F and spaced 10 minutes apart. The laundry peak included two, 21 gallon, 5 gpm loads occurring 22 minutes apart; one at a "hot" setting of 140°F, and one at a "warm" setting of 105°F. The morning peak was assumed to have a total duration of 20 minutes, and the laundry peak 30.

Collector areas ranging from 35 to 65 square feet were investigated using the computer model (again see Section 2.2). This range is quite reasonable, considering that a typical solar system for the assumed location and 100 gallon per day load would probably utilize one or two "standard" 4 foot by 8 foot panels. In addition, for Albuquerque, the computer model predicted a 75 percent solar fraction at 35 square feet, and 95 percent at 65, which should bracket the range containing the economic "optimum".

The size of the solar preheat tank was chosen based upon an industry "rule of thumb", which calls for roughly 1 to 2 gallons of storage per square foot of collector. For the collector areas cited, this would correspond to a low of 35 gallons and a high of 130. Of course, the fact that the demand tank is also directly coupled to the preheat tank should enhance the effective solar storage, however, as discussed in Section 2.2, the degree of enhancement, if any, and the optimum distribution of volume between the two tanks could not be determined with the current simulation model. Hence, for the purpose of the final prototype, the preheat tank volume was set at 80 gallons, the middle of the rule of thumb range.

The demand tank volume and heat pipe firing rate were determined in combination, in the same manner as conventional heaters are sized. The philosophy was to take the worst case approach. In other words, from a marketing viewpoint, the system must not inconvenience the user if the sun fails to shine. Of course, some inconvenience may be acceptable in the name of energy conservation, and for the purpose of maximum solar utilization. However, at this stage it seemed reasonable to follow accepted sizing criteria such as outlined by ASHRAE [36], and NBS [37]. The process was still a subjective one, and hence, the final specifications leaned towards the smaller capacity.

The procedure is quite simple, and involves determining the maximum hot water use in any one hour, and selecting a combination of tank volume and firing rate which will yield a Hot Water Supply Rating within 3 gallons. The Hot Water Supplying Rating is defined as the sum of the nominal storage capacity, and the recovery rate at 100°F rise of the heater [37]. The Hot Water Supply Rating of the prototype for various combinations of firing rate and demand tank volume are shown in Table 4.2. This figure assumes 40°F supply water, a 140°F set temperature, no solar preheat (i.e., a 100°F temperature rise), and a usable water ratio of 90 percent due to stratification and dilution by incoming cold water.

TABLE 4.2

Hot Water Supply Rating
[Gallons/Hour]

		Demand Tank Volume [gallons]			
		15	20	25	30
Firing Rate [Btu/hr]	30,000	43	47	52	56
	40,000	52	56	61	65
	50,000	62	66	71	75
	60,000	72	76	81	85
	70,000	81	85	90	117

Now, referring back to the two worst case loads, and accounting for the end use mixing with cold water required to achieve 105°F, the actual requirement seen by the demand tank is 22 gallons in 20 minutes for the morning peak, and 35 gallons in 30 minutes for the laundry peak. In the NBS procedure these loads would be assumed to occur over 1 hour, and a heater rating would be selected accordingly. However, for a residential application, especially one with smaller loads, one hour is much too large because the demands occur in short segments, and in the worst case, these are closely spaced. The one hour interval can result in an undersized system. Instead, the worst case demands were converted to an hourly duration and then compared to the ratings in Table 4.2. The morning peak was then 66 gallons per hour, and the laundry peak 70.

Several combinations of firing rate and tank size would have resulted in the desired 66 to 70 gallon per hour rating. However, based upon the operating experience gained with the heat pipe prototype, and space constraints on the tank volume, a combination of 50,000 Btu/hr and 25 gallons was selected.

It is of interest to compare this selection to the minimum water heater sizes cited in HUD's "Minimum Property Standards for One and Two Family Living Units." For a 3 bedroom, 1 1/2 bath home, HUD calls for 30 gallons of storage, a 36,000 Btu/hr input, a 30 gallon/hr recovery, and a 60 gallon 1 hour draw. The latter is the most critical since any combination of requirements producing the specified draw is acceptable. The size selected for the prototype yields a 71 gallon 1 hour draw; 11 gallons more than the minimum required by HUD [40].

The HUD minimum is probably lower than that usually sought in the marketplace, but the rating selected for the prototype should still be

ample, especially considering that the solar preheat tank will probably seldom, if ever, drop to 40°F as assumed in calculating the ratings in Table 4.2.

A more sophisticated approach to the problem of matching tank size and firing rate to the anticipated load would be to use the computer simulation model to examine the transient response of the system to the worst cases conditions. This was done and is discussed in Section 2.2.3.2.

4.1.3 Evaporator Design

The final evaporator design is illustrated in Figure 4.3. It closely resembles the bench setup except for the headers, which were rectangular in cross section (2 x 3/4 inch) instead of circular. This change resulted in a much larger fluid reservoir (without increasing overall area), thereby adding a margin of safety to the cold startup process. The finned tubes, made of copper, were manufactured by Weiland, and were 7 1/8 inches long, with a 5/8 inch ID, 1 1/2 inch fin diameter, and 9 fins per inch. The vapor and condensate return tubes were also copper (alloy 122), and were 1 inch and 3/8 inch diameter, respectively. Although it would not be included in a production model, a pressure gauge was installed on the vapor tube as a convenient means of monitoring the integrity of the vacuum.

The entire evaporator assembly was fabricated inhouse. Components were held in place with clamping jigs and brazed with brass or phos-copper filler, except for the pressure gauge fitting which was threaded. After brazing, the assembly was flushed with water and pickled with nitric acid to remove the oxide scale.

4.1.4 Burner and Combustion Chamber Design

The final burner design, shown with the evaporator in Figure 4.3, was identical to the one which resulted from the testing described in Section 3.4, except it had a smaller tube array. It consisted of 11, 7 1/2 inch long, 3/8 inch diameter, thin walled steel tubes instead of 13. Also, each tube had only 11, 0.081 inch diameter ports drilled on 1/2 inch centers, resulting in 121 individual flame ports instead of 169. These changes were intended to further reduce losses, and to a limited extent, the thermal capacitance of the burner itself. Gas and "primary" air were supplied through a 3/4 inch venturi mixer [41].

FIGURE 4.3 EVAPORATOR, BURNER
AND COMBUSTION
CHAMBER

The burner was assembled in the same manner as the evaporator. Components were held in place with clamping jigs and brazed with brass filler.

The combustion chamber, also illustrated in Figure 4.3, consisted of an aluminum enclosure (.032 inch walls) positioned tightly around the evaporator, covered with 1 inch of ceramic fiber insulation, and fitted with a stainless steel cap (also .032 inch walls) designed to reduce the rectangular cross section of the duct to a 4 inch, circular flue connector. The combustion chamber, burner, and all controls were attached to the upper portion of the solar tank with a structural framework made of 1/4 inch steel angle.

4.1.5 Condenser/Demand Tank Design

The 25 gallon jacketed demand tank, illustrated in Figure 4.4, was made entirely of copper alloy 101 (oxygen free), had an 18 inch inside diameter, and was 18 inches tall (excluding heads). The main shell and heads were 3/16 inch thick, and the jacket wall 0.067 inch. The jacket had a 1 1/2 inch fitting at the top for the vapor tube, and a 1/2 inch fitting at the bottom for the condensate return tube. In addition, a 2 1/2 inch brass flange was welded to the center of the bottom head for connecting to the solar tank.

The jacketed tank represented the most expensive and complex step in the fabrication process. First, the main shell was rolled, the heads spin formed, and these three components TIG welded with a phos-copper filler. The jacket was then wrapped and bonded to the main shell with 1/2 inch plug welds on 2 1/2 inch centers. Finally, holes were drilled for the vapor and condensate return tube fittings, the thermostat well, and the connecting flange, and these items welded in place.

4.1.6 Solar Tank Design

The 80 gallon solar traced tank is shown in Figures 4.1 and 4.2. It was constructed of 10 gauge carbon steel, was 24 inches in diameter, and 40 inches tall (excluding the heads). The traced coil was formed by wrapping and soldering 44 inches of rectangular copper tubing (3/4 inch by 3/8 inch) seven times around the bottom 18 inches of the tank. The ends of the coil had 5/8 inch threaded female fittings for coupling to the collector loop. In addition, the tank itself had three 1/2 inch threaded male fittings; one for connecting to the water main, and the other two to facilitate bypassing the traced coil in the event a drain down or external exchanger configuration were desired. A magnesium

FIGURE 4.4 FINAL DEMAND TANK DESIGN

sacrificial anode was suspended from the top head. Also welded to the top head was a 2 1/2 inch steel flange for mounting the demand tank. The split tube connector assembly was made of plastic and suspended in the solar tank flange. It was 10 inches long (overall) with 3 inch "offsets".

The protective tank lining became the major design obstacle. Initially, several options were available; a stainless steel tank, galvanizing the inner wall, a teflon lining, an epoxy lining, or a glass lining. In production, a glass lined tank would be used, but for the prototype, it originally appeared that a teflon lining would be the most practical option.

The key to the lining question was that a decision had to be made before soldering the traced coil. This was because the galvanizing and glass lining processes required temperatures high enough to melt the traced coil solder, and the traced coil soldering process required temperatures high enough to damage the epoxy or teflon linings. Hence, the fabrication sequence depended on the type of tank protection selected.

It was only after the decision was made to use teflon, and the coil soldered to the tank, that the supplier changed his recommendation and decided the teflon process was not suitable for this application. The only option then was to use the epoxy coating. This was done inhouse, and applied with a brush in two coats. Unfortunately, the resulting lining was almost ten times thicker than the teflon coating would have been (20 mils instead of 2).

The traced coil was designed for 65 square feet of collector delivering 150 Btu/hr ft² at noon (9,750 Btu/hr total). A nominal temperature difference of 20°F was assumed between the average temperature of the collector fluid passing through the coil, and the bulk temperature of the water in the portion of the tank covered by the coil.

The convection heat transfer coefficient on the inner tank wall was estimated as follows:

$$h = 0.10 \frac{k_f}{L} (GrPr)^{.33} \quad (4.1)$$

Gr = Grashof Number

Pr = Prandtl Number

k_f = thermal conductivity of water

L = the vertical extent of tank wall covered
by coil

Under the assumed conditions, the product of the Grashof and Prandtl numbers reduces to the following:

$$\text{GrPr} = 5.2 \times 10^8 \Delta T L^3 \quad (4.2)$$

The term ΔT is the difference between the average temperature of the wall covered by the coil, and the bulk temperature of the water in that portion of the tank. However, the "known" temperature difference is between the collector fluid and tank water. The two are related by the "fin efficiency" of the tank wall between consecutive turns of the coil, as follows:

$$\Delta T = F \Delta T' \quad (4.3)$$

where

$\Delta T'$ = the difference between the average temperature of the collector fluid passing through the coil and the bulk temperature of the water in the portion of the tank covered by the coil

F = tank wall fin efficiency

$$F = \frac{\tanh m (W-D)/2}{m (W-D)/2} \quad (4.4)$$

where

$$m^2 = \frac{h}{K\delta}$$

δ = thickness of tank wall

k = thermal conductivity of tank wall

W = coil pitch

D = coil tube diameter

Equation 4.3 assumes that the thermal resistance of the coil tube wall and convective film on the inside surface of the tube are negligible.

For any coil geometry, Equations 4.1 through 4.4 had to be solved in an iterative fashion to determine h and in turn, the overall heat transferred. According to the design criteria, the latter was to be 9,750 Btu/hr. In addition, the thickness of the tank wall, the thermal

conductivity of the tank wall material, and the diameter of the coil tubes were also known. Therefore, it was a simple matter to work through the equations to determine a suitable combination of coil pitch and area of tank wall covered. The final choice was a coil pitch of 2 1/2 inches, covering the bottom 18 inches of tank wall in seven turns. With this design, h was estimated to be 71 Btu/hr ft² °F and the overall heat transfer coefficient area product of the coil was estimated to be 473 Btu/hr °F.

At the time this calculation was performed, it was assumed that the resistance of the tank lining could be neglected (the teflon coating was to be only a few mils thick). However, the thicker epoxy coating that was finally applied did represent a significant thermal resistance. This effect is discussed in Section 4.2.

4.1.7 Controls, Insulation and Enclosure

The prototype controls consisted of a water temperature thermostat, pressure gauge, two overheat safety thermostats, pressure relief valve, and intermittent ignition system. All but the pressure gauge would be included on a production model.

The water temperature thermostat is mechanically activated by a pressure bulb located in a well in the side of the demand tank. The well was positioned 6 inches from the lower head, at a point where a temperature corresponding to the average bulk tank temperature was expected to occur based upon the results of the stratification test described in Section 3.5. The thermostat control was located below the burner, suspended from the combustion chamber support structure.

The two overheat thermostats were located on the fin tip of the center evaporator tube, and outside surface of the vapor tube. These were connected in series to a control relay and reset switch for the main thermostat. If either the vapor temperature entering the condenser exceeds 250°F (15 psia), or the fin tip temperature exceeds 350°F, the system is shut off and must be manually reset. The former condition would signify an empty tank, and the latter, evaporator dryout.

The burner was fitted with a Robert Shaw intermittent ignition system. When signaled by the thermostat, it ignites the pilot with an electric spark, and after the pilot is sensed, opens the main gas valve.

As shown in Figures 4.1 and 4.2, the entire heater was enclosed in a cylindrical shell, fabricated in two halves from 20 gauge sheet steel,

and joined along the two vertical seams with sheet metal screws to facilitate disassembly. The top and bottom of the enclosure were cut from a sheet of 1 inch plywood. As with a conventional heater, the shell served two functions; to make the final product more aesthetically pleasing, and to contain the insulation. The space between the two tanks and the shell was filled with fiberglass insulation; 5 inches on top, and 2 inches on the bottom. The completely assembled unit was 72 inches tall and 28 inches across; just small enough to fit through a standard door frame.

4.2 TESTING

The end product of this effort was to be a working engineering prototype, with only enough checkout testing to make sure that it worked, and to understand its performance during operation. No funding was included for endurance testing (so as to understand any possible degradation mechanisms), since this was thought to depend too much on fabrication details to be significant at the engineering prototype stage. No funding was included for detailed, long-term performance tests of the heater subjected to realistic solar inputs and hot water demands (in order to obtain a realistic estimate of the improvement in "solar fraction" provided by this heater concept). Quoting from the Altas proposal, in the work statement on Experimental Work: "Only the component testing work can be estimated at this time. A testing program for a final prototype (or prototypes) can be quite simple, or it could be very ambitious. We are in no position to make a judgement on the desirable level of testing at this point."

What is described below is hence the limited checkout and performance testing program funded on this first effort. Some testing to determine solar fraction results is planned at the National Bureau of Standards, to which the prototype is to be shipped at the end of the current program. Some further understanding of long-term performance, and of how to optimize it, will come out of a Gas Research Institute follow-on and other subsequent efforts.

Checkout of the final prototype consisted of four distinct tests; efficiency, startup, cycling, and simulated short-term solar input. Since all of the components, except for the traced coil, had been thoroughly tested as part of the development effort, it was not necessary to perform extensive checkout tests on the final prototype, but only to insure that it had been properly assembled, and that the minor changes made in the final design did not result in any unexpected performance characteristics.

The experimental setup for measuring the steady state firing efficiency of the prototype was identical to the one used for the bench model shown in Figure 3.11. Firing rate was monitored with a bellows gas meter, and energy input to the tank was measured by collecting and weighing a small throughput of "cooling" water. Flue gas, fin tip, and vapor operating temperatures were also recorded. Firing efficiency was determined to be 81.4 percent at 50,000 Btu/hr, with a flue gas temperature of 260°F and a heat pipe operating pressure of 3 psia.

As described in Section 3.3, experiments with the bench prototype had shown that the evaporator was most likely to overheat under certain startup conditions. To insure that this was not a problem with the final prototype, it was run through a series of startup transients over a range of demand tank temperatures. Initially, some evaporator overheating was encountered at very low tank temperatures. It was believed that this was caused by an excessive pressure drop between the evaporator and condenser, causing the condensate return flow to be pushed up into the condenser. This was a new performance characteristic attributed to the fact that the prototype condenser was larger and more effective than any of the previous experimental designs. The problem was completely eliminated by bleeding a small amount of air into the heat pipe, thereby raising the condenser pressure and lowering the pressure drop. The system then functioned properly under all startup conditions, with no trace of evaporator overheating.

Another condition found conducive to evaporator overheating during the experimental stage was quick cycling. The final prototype had a 30 second delay built into the intermittent ignition system, so this represented the shortest cycling frequency. Under this worst case cycling condition, no problems were encountered.

The final test involved the solar tank traced coil. A solar collection loop was simulated by connecting the traced coil to a pump and a 4 kw electric heater, controlled with a 240 volt variable transformer. With this arrangement it was possible to mimic the flow conditions the traced coil would experience when connected to a typical collector array, and in particular, to measure the impact of the epoxy lining.

During the test, the flow rate through the coil was 1,092 lbm/hr, the coil inlet temperature 137°F, the tank temperature 101.5°F, and the resulting coil outlet temperature 127°F. From these data, the energy input to the tank and the overall heat transfer coefficient-area

product of the coil were calculated to be 10,920 Btu/hr and 357 Btu/hr °F, respectively.

Referring back to the coil sizing calculations described in Section 4.1.6, the overall heat transfer coefficient-area product predicted was 437 Btu/hr °F, but this did not account for the epoxy lining. The exact thickness and thermal conductivity of the lining were not known, but using Equation 4.1 through 4.4 and the experimental results, the overall conductance of the lining was estimated to be 120 Btu/hr ft °F. For a 20 mil coating, this corresponds to a thermal conductivity of 0.2 Btu/hr ft °F.

With the thermal resistance of the epoxy coating now known, it was possible to predict how the traced coil would perform when coupled to a typical collector array. Specifically, the heat exchanger factor could be calculated. This factor represents the ratio of useful energy delivered to the tank, to the energy that would be delivered if the traced coil were not present, i.e. if tank water were circulated directly through the collectors.

The energy collected by a flat plate collector coupled to a traced tank can be calculated using the following modification of Whillier's well known linear expression [31]:

$$Q_c = \left[1 + \frac{F_R U_c A_c}{C_c} \left(\frac{1}{\epsilon} - 1 \right) \right]^{-1} \left[Q_i F_R \tau \alpha A_c - F_R U_c A_c (T_B - T_a) \right] \quad (4.5)$$

Q_c = useful energy collected

Q_i = the energy incident on the collector

F_R = collector heat removal factor

$\tau \alpha$ = transmittant absorption product of the glazing

A_c = collector

U_c = collector heat loss coefficient

T_B = tank temperature

T_a = ambient temperature

C_c = flow stream capacity rate of collector fluid

ϵ = traced coil effectiveness

$$\epsilon = 1 - \exp \left(\frac{-UA}{C_c} \right) = \frac{T_{in} - T_{out}}{T_{in} - T_B} \quad (4.6)$$

UA = overall heat transfer coefficient of the traced coil

T_{in} = temperature of fluid entering coil

T_{out} = temperature of fluid leaving coil (entering collector)

The group of parameters in the first set of brackets in Equation 4.5 represents the heat exchanger factor.

In Equation 4.5, the overall heat transfer coefficient of the traced coil is a function of the tank wall fin efficiency, F , the convection heat transfer coefficient on the inner tank wall, h , (both defined in Section 4.1.6), and the resistance of the epoxy lining:

$$UA = F \left[\frac{1}{h} + \frac{\Delta X}{k_e} \right]^{-1} A \quad (4.7)$$

where

$\frac{\Delta X}{k_e}$ = thermal resistance of the epoxy lining

For any given collector and traced coil geometry, Equations 4.5 through 4.7 can be solved iteratively to arrive at the heat exchanger factor. For the purpose of evaluating the present design, assume a system with 65 square feet of single glazed, flat black, flat plate collector with the following characteristics:

$$\begin{aligned} F_R &= 0.9 \\ U_C &= 1.0 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F} \\ \tau\alpha &= 0.85 \\ C_C &= 650 \text{ Btu/hr } ^\circ\text{F} \\ Q_i &= 300 \text{ Btu/hr ft}^2 \\ A_C &= 65 \text{ ft}^2 \end{aligned}$$

Assume also that the tank temperature is 100°F and the ambient temperature is 50°F . (Note that the incident energy, Q_i , would correspond to noon on a clear summer day.)

Iterating through Equations 4.1 through 4.7 yields the following:

$$\begin{aligned} \text{heat exchanger factor} &= 0.88 \\ Q_C &= 10,603 \text{ Btu/hr} \\ T_{in} - T_{out} &= 16.3^\circ\text{F} \\ \epsilon &= 0.407 \\ h &= 73.6 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F} \\ F &= 0.791 \\ UA &= 340 \text{ Btu/hr } ^\circ\text{F} \end{aligned}$$

It is of interest to examine what these results are for the identical system with no epoxy lining:

heat exchanger factor = 0.93

$Q = 11,109 \text{ Btu/hr}$

$T_{in} - T_{out} = 17.1^{\circ}\text{F}$

$\epsilon = 0.531$

$h = 78.4 \text{ Btu/hr ft}^2 ^{\circ}\text{F}$

$F = 0.695$

$UA = 513 \text{ Btu/hr } ^{\circ}\text{F}$

Hence, for a typical collector array, the collector penalty associated with the final prototype traced coil and lining is 12 percent, 5 percent of which is due to the lining.

It is generally accepted that the heat exchanger factor should be well above 90 percent in a well designed system (although this is dependent upon the "quality" of the collector). Clearly, the traced coil design will have to be improved for future prototypes, and indeed, a different lining alone could reduce the collector penalty by as much as 5 percent.

Under this program, no funding was allocated to develop an optimum traced coil design, however, considerable work as collector-to-storage heat transfer is planned for future work, and in fact, at this writing an extensive follow-on is already in progress.

Section 5

COMMERCIALIZATION AND FUTURE WORK

This contract was for the development of an engineering prototype or technical feasibility prototype, which is normally the first step in the development of a new product. The next step is normally the development of more realistic hardware, from a cost and manufacturing point of view. Subsequently there is a detailed design phase, special tooling and an associated production line must be set up, marketing, installation and servicing information and material must be developed, and organizations must be set up for production, distribution, marketing, installation, servicing, etc. At many points along this path, decisions are made whether to pursue or drop the project.

The most inexpensive part of this process is normally the preliminary R&D to get a concept from the idea phase to the technical feasibility model phase. It is, however, a very critical pump priming operation. Performance projections on a piece of paper may have credibility problems. A piece of working hardware does not. If working hardware does not exist, it may be impossible to get a product off the ground.

During the DOE contract phase, a continuing effort was made to make information on the project available to the Gas Research Institute (GRI), the American Gas Association, the Institute of Gas Technology, the Copper Development Association, Pacific Gas & Electric Company, and others in the gas research, appliance manufacturing, materials research or utility field.

During this contract there was continuing evidence that the design concept was sound and that its development was progressing in a satisfactory fashion. The Gas Research Institute had already allocated funds for the development of a gas-fired water heater especially designed for a solar DHW backup function, and when the time came to choose a contractor with a design for this purpose, the Altas Corporation and the DOE-sponsored design was chosen over three other competitors.

At present, Altas Corporation has a Gas Research Institute contract (No. 5014-343-0279) for \$323,000 which started in February 1980, and is to run for two years, for the development of two pre-production prototypes and the performance of the R&D necessary to produce these

units. Several other organizations are involved in this work. The Copper Development Association, through Mr. Carl Gaffoglio, is providing guidance on design requirements and manufacturing processes with copper alloys. Messrs. Robert Cook and John Eising of the A.O. Smith Corporation are providing guidance on the requirements and practices of the appliance industry. Dr. Allan Michaels and Mr. Joseph Pearson of the Argonne National Laboratory will continue to provide guidance and to provide an interface with the other DOE-sponsored work in solar energy and in DHW systems.

In the meantime, hardware produced by Altas on the DOE and GRI programs will begin to undergo tests by independent laboratories. Altas has tested the DOE engineering prototype under simulated solar input and under gas firing conditions, but has not yet performed tests with simulated consumption and solar energy inputs so as to get an understanding of the solar fractions (of DHW energy) which will be possible with the equipment in actual applications. The NBS has agreed to perform such tests in late summer 1980, and the results will be directly comparable with those obtained on five or six other configurations it has tested.

SECTION 6

REFERENCES CITED

1. Chapman, A., Heat Transfer, Third Edition, MacMillan Publishing Co., N.Y., 1974, pp.368-379.
2. Rohsenow, W.M., and Choi, H., Heat Mass and Momentum Transfer, Prentice-Hall Inc., New Jersey, 1961, pp.237-247.
3. Chi, S.W., Heat Pipe Theory and Practice: A Source Book, McGraw-Hill, 1954.
4. McAdams, W.H., Heat Transmission, Third Edition, McGraw-Hill, 1961.
5. Perry, H.P., and Chilton, C.H., eds., Chemical Engineers Handbook, Fifth Edition, McGraw-Hill, 1973, pp. 10,19.
6. Rohsenow, W.M., and Griffith, P., "Correlation of Maximum Heat Flux Data for Boiling of Saturated Liquids", AIChE-ASME Heat Trans. Symposium, Louisville, KY, March 1955.
7. Kays, W., and London A.L., Compact Heat Exchangers, 2nd Edition, McGraw Hill, 1964.
8. Briggs, D.E., and Ward, E.H., "Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes", CEP Symposium, Series No. 41, 59, 1.
9. Ward, D.J., and Young, E.H., "Heat Transfer and Pressure Drop of Air in Forced Convection Across Triangular Pitch Banks of Finned Tubes", CEP Symposium, Series No. 29, 55, 7, 1959.
10. "Heat Transfer and Pressure Drop Characteristics of Dry Tower Extended Surfaces, Part I: Heat Transfer and Pressure Drop Data", Prepared for Battelle Memorial Institute by PFR Engineering Systems, Inc., Marina Del Rey, CA, March 9, 1976, Report No. BNWL-PFR-100.
11. Ibid., "Part II: Data Analysis and Correlation".
12. Steen, D.A., and Walks, G.B., AEC Report NYO-3114-2, 1964.
13. ESCOA Engineering Manual, ESCOA Fintube Corporation, Pryor, OK, 1979.
14. TRNSYS: A Transient Simulation Computer Program, Version 9.2, Engineering Experiment Station Report No. 38, Solar Energy Laboratory, University of Wisconsin, Madison, WI, Feb. 1978.
15. Levy, E.K., "Theoretical Investigation of Heat Pipes Operating at Low Vapor Pressure", J.Eng.Ind., Vol.90, pp.547-552, 1968.
16. Steen, D.A. and Wallis, G.B., AEC Report NYO-2114-2, 1964.

17. Wallis, G.B., "One Dimensional Two-Phase Flow", McGraw-Hill, New York, 1969, p.387.
18. Anderson, W.T., "Hydrogen Evolution in Hickel Water Heat Pipes", AIAA 8th Thermophysics Conference, Palm Springs, California, July 16-18, 1973, AIAA Paper No. 73-726.
19. Monzel, W.D., "Compatibility Tests of Various Heat Pipe Working Fluids and Structural Materials at Different Temperatures", 3rd Int. Heat Pipe Conference, Tech. Papers, Palo Alto, California p.98-101, May 22-24, 1978, AIAA, Inc., New York, NY.
20. Basiulis, A., and Hummel, T.A., "Designers Guide to Heat Pipes", Design News, March, 1974, Cahners Publishing Co.Inc.
21. Busse, C.A., Loens, J., and Campanile, A., "Hydrogen Generation in Water Heat Pipes at 250°C". Proceed. 1st International Heat Pipe Conference, Stuttgart, Federal Republic of Germany, October 1973.
22. Basiulis, A. and Prager, R.C., "Compatibility and Reliability of Heat Pipe Materials", AIAA Paper No. 75-660, AIAA 10th Thermophysics Conference, Denver, CO., May 1975.
23. Schwartz, J., "Performance Map of the Water Heat Pipe and the Phenomenon on Noncondensable Gas Generation", ASME Preprint 69-HT-15 (1969).
24. Petrick, S.W., "Hydrogen Generation in Water/Stainless Steel Heat Pipes", ASME Preprint 72-WA/HT-37 (1972).
25. Kreeb, H., Groll, M., and Zimmermann, P., "Lifetest Investigations with Low Temperature Heat Pipes", Proceed. 1st International Heat Pipe Conference, Stuttgart, Federal Republic of Germany, October 1973.
26. Schneider, P.J., Conduction Heat Transfer, Addison-Wesley Publishing Company, Inc., Cambridge, MA, 1955, p.84.
27. "TRNSYS-A Transient Simulation Program", Engineering Experiment Station Report No. 38, February 1978.
28. Sweigert, R.L., and Beardsley, M.W., "Empirical Specific Heat Equations Based Upon Spectroscopic Data", Georgia School Technol. Bull., Vol.1, No. 3, June 1938.
29. Kreith, F., Principles of Heat Transfer, Third Edition, Intext Educational Publishers, New York, New York, 1973.
30. Holman, J.P., Heat Transfer, Third Edition, McGraw-Hill, 1972.
31. de Winter, F. and Horel, J.D., Investigation of Methods to Transfer Heat From Solar Liquid Heating Collectors to Heat Storage Tanks, U.S.Department of Energy, Final Report, Contract No. E-(04-3)-1238, April 20, 1978.
32. Suh, C.H., Computer Aided Design of Mechanisms, University of Colorado Press, Boulder, CO, Jan. 1973.

33. "Appliance Usage Study", Marketing Research Department, San Diego Gas & Electric Company, 1974.
34. Mutch, J.J., "Residential Water Heating: Fuel Conservation, Economics, and Public Policy", Rand Corporation, Santa Monica, 1974.
35. Clinton, J.R., Simulation Analysis of Solar Water Heating with Applications to the San Diego Region, Energy Center, University of California, San Diego, LaJolla, CA, August 1978.
36. American Society of Heating, Refrigeration, and Air-Conditioning Engineers, ASHRAE Handbook and Product Directory, 1976 Systems.
37. National Bureau of Standards, Household Water Heaters, LC 1058.
38. A 500,000 Btu annual heat loss can be expected for the backup heater corresponding to a 25 gallon (95 liter) spherical tank, at 140°F (60°C) in a 60°F (16°C) environment with R-16 (0.38 W/m² °C) insulation and with three thin-walled 3/4 inch (2 cm) diameter stainless steel tubes as penetrations. If copper tubes instead of stainless tubes are used, one must use R-22 (0.29 W/m² °C) to get the same heat loss.
39. This suggests that final models may have to be leveled during installation. The same is needed for many washing machines and absorption refrigerators.
40. The 1-hour draw for a 25 gallon tank with a usable water ratio of 90 percent, a 50,000 BtuH burner operating at 81.4 percent producing a 100°F temperature rise is given by
$$0.9 \times 25 + \frac{50,000 \times 0.814}{100 \times 1 \times 8.34 \text{ lb/gal}} = 71.3 \text{ gal}$$
41. These design simplifications (fewer tubes, fewer burner ports) were made based on confidence obtained on the earlier, more complex geometries. It was expected that the changes would not affect the possible firing efficiency significantly. This proved to be true.

APPENDIX A

Derivation of the Heat Pipe Computer Model

As discussed in Section 2.1, the primary objective in modeling the heat pipe assembly is the ability to predict, given the backup tank temperature and the conditions of combustion (i.e., the flame temperature and mass flow rate), the rate of energy transfer into the tank. In addition, intelligent design and optimization require knowledge of: 1) the amount of fuel burned (and hence, the overall efficiency); 2) the temperature at key locations; 3) the resistance to heat transfer associated with each component; 4) the pressure inside the heat pipe; 5) the pressure drop in each section; 6) the sonic, entrainment, and boiling limitations to heat transfer; and 7) the quantity of fluid entrained in the entire assembly. The current heat pipe model will supply this information for heat pipes conceptually identical to the one shown in Figure 2.1; a tubular evaporator partially filled with heat transfer fluid; an adiabatic connecting section; and a condenser employing film condensation on the inner surface and free convection on the outer surface.

The remainder of this appendix describes the development of the heat pipe model; Section A.1 deals with estimating overall heat transfer, Section A.2 describes performance limitations, and Section A.3 combustion considerations. A simplified schematic illustrating the system geometry and important design variables is presented in Figure A.1.

FIGURE A.1 HEAT PIPE MODELING NOMENCLATURE

A.1 Heat Transfer Rate

The total energy transferred into the backup tank via the heat pipe is determined using a modification of the heat exchanger effectiveness (ϵ) and number of transfer units (NTU) method developed by Kays and London [7]. The effectiveness compares the actual heat transfer rate Q , to the thermodynamically limited maximum heat transfer rate achievable in a counterflow heat exchanger of infinite heat transfer area, $Q_{\max}$. In other words, when heat is transferred between hot and cold fluid streams, the effectiveness is a measure of how close the temperature of the exiting cold fluid approaches the temperature of the entering hot fluid.

$$\epsilon = \frac{Q}{Q_{\max}} = \frac{C_h (T_{h,in} - T_{h,out})}{C_{\min} (T_{h,in} - T_{c,in})} = \frac{C_c (T_{c,out} - T_{c,in})}{C_{\min} (T_{h,in} - T_{c,in})} \quad (A.1)$$

where

- Q = actual heat transfer rate, Btu/hr
- $Q_{\max}$ = maximum possible heat transfer rate, Btu/hr
- T = temperature, $^{\circ}\text{F}$
- C = fluid capacity rate = $(\dot{m}c_p)$, Btu/hr $^{\circ}\text{F}$
- $\dot{m}$ = mass flow rate, lbm/hr
- c_p = fluid specific heat, Btu/lbm $^{\circ}\text{F}$
- h = hot fluid
- c = cold fluid
- $\min$ = the smaller of C_h and C_c

Referring to Figure A.2 and considering only the evaporator portion of the heat pipe assembly, the maximum heat transfer is defined as:

$$Q_{\max} = (\dot{m}c_p)_{\text{prod}} (T_{\text{flame}} - T_{\text{ve}}) \quad (A.2)$$

where

- $(\dot{m}c_p)_{\text{prod}}$ = capacity rate of combustion products flowing across exchanger, Btu/hr $^{\circ}\text{F}$
- T_{flame} = temperature of combustion products, $^{\circ}\text{F}$
- T_{ve} = saturation vapor temperature inside of evaporator, $^{\circ}\text{F}$

Note that for this case, the gas side fluid capacity is taken as the minimum capacity since the apparent capacity of the boiling heat pipe fluid, being a constant temperature process, is infinite.

Combining Equations A.1 and A.2, the actual heat transfer rate is:

$$Q = \epsilon_x (\dot{m}c_p)_{\text{prod}} (T_{\text{flame}} - T_{\text{ve}}) \quad (\text{A.3})$$

where

$$\epsilon_x = \text{the effectiveness of the evaporator exchanger}$$

The NTU is a nondimensional quantity representing the "heat transfer size" of the heat exchanger. It is the ratio of the heat exchange capacity per degree of mean temperature difference, UA, and the minimum fluid capacity rate defined above, $C_{\min}$ (i.e. the heat transferred per degree temperature rise of the fluid).

$$\text{NTU} = \frac{UA}{C_{\min}} \quad (\text{A.4})$$

In general, effectiveness is a function of the NTU, the fluid capacity rates, and the exchanger geometry. For the exchanger configuration of interest here, in which the apparent cold side fluid capacity rate is infinite, effectiveness is simply:

$$\epsilon_x = 1 - e^{-\text{NTU}_x} \quad (\text{A.5})$$

where

$$\text{NTU}_x = \text{NTU of the evaporator portion of the heat pipe}$$

Now, referring back to Equation A.3, the only remaining unknown is the vapor temperature inside the evaporator, T_{ve} . The heat transfer between the evaporator and the tank is described by the following:

$$Q = (UA)_{\text{pipe}} (T_{\text{ve}} - T_t) \quad (\text{A.6})$$

where

$$(UA)_{\text{pipe}} = \text{overall heat transfer coefficient/surface area product for the heat pipe excluding the evaporator, Btu/hr } ^\circ\text{F}$$

$$T_t = \text{the tank temperature, assumed to be uniform, } ^\circ\text{F}$$

Upon rearranging, Equation A.6 can be solved for T_{ve} and substituted into Equation A.3 to yield:

$$Q = \epsilon_x (\dot{m}c_p)_{\text{prod}} \left[T_{\text{flame}} - \frac{Q}{(UA)_{\text{pipe}}} - T_t \right] \quad (\text{A.7})$$

and solving for Q:

$$Q = \left[\frac{\epsilon_x}{1 + \frac{\epsilon_x (\dot{m}c)_p}{(UA)_{\text{pipe}}}} \right] (\dot{m}c)_p \text{ prod } (T_{\text{flame}} - T_t) \quad (\text{A.8})$$

Equation A.8 is of the same form as Equation A.3 in which Q is defined in terms of effectiveness and known temperatures. Hence, an overall heat pipe effectiveness can be defined as follows:

$$\epsilon_{\text{tot}} = \frac{\epsilon_x}{1 + \frac{\epsilon_x (\dot{m}c)_p}{(UA)_{\text{pipe}}}} \quad (\text{A.9})$$

Equation A.8 then becomes:

$$Q = \epsilon_{\text{tot}} (\dot{m}c)_p \text{ prod } (T_{\text{flame}} - T_t) \quad (\text{A.10})$$

In order to solve Equation A.10 the overall conductance of the evaporator, $(UA)_x$, and that of the remaining heat pipe components, $(UA)_{\text{pipe}}$, must be determined. The reciprocal of $(UA)_x$ is the overall thermal resistance of the evaporator which is simply a series summation of the resistance of the hot-side convective boundary layer, the tube wall, and the boiling taking place inside of the tube.

$$\frac{1}{(UA)_x} = R_{\text{conv}} + R_{\text{ewall}} + R_{\text{boil}} \quad (\text{A.11})$$

Likewise, the reciprocal of $(UA)_{\text{pipe}}$ is the sum of the resistance associated with the pressure drop (and hence temperature drop) along the length of the flow passages, the condensing film, the tank wall, and the convective film on the inside of the tank.

$$\frac{1}{(UA)_{\text{pipe}}} = R_{\text{pdrop}} + R_{\text{film}} + R_{\text{cwall}} + R_{\text{free}} \quad (\text{A.12})$$

The assumptions and calculation procedures for determining the components of resistance in Equations A.11 and A.12 are discussed below in Sections A.1.1 through A.1.7.

A.1.1 Evaporator Film Convection

Heat transfer data for extended surface heat exchangers are typically available in two forms. The first, and most desirable, is actual experimental data for the specific tube arrangement and extended surface geometry of interest. Such data have been generated by numerous researchers for a wide variety of exchanger configurations [7, 8, 9, 10, 11] and are usually presented graphically (as illustrated by the sample plot in Figure A.2 reproduced from Kays and London [7]) in the form of:

$$St Pr^{2/3} = \phi_1(Re) \quad (A.13)$$

where

$$St = \text{Stanton number} = \frac{h}{Gc_p}$$

$$Re = \text{Reynolds number} = \frac{dG}{\mu}$$

$$Pr = \text{Prandtl number} = \frac{c_p \mu}{K}$$

Or in nomenclature consistent with that used here:

$$\begin{aligned} St &= \frac{h_e}{G_{\text{prod}} c_{\text{prod}}} \\ Re &= \frac{G_{\text{prod}} D_{\text{eo}}}{\mu_{\text{prod}}} \\ Pr &= \frac{c_{\text{prod}} \mu_{\text{prod}}}{K_{\text{prod}}} \end{aligned} \quad (A.14)$$

where

h_e = convection heat transfer coefficient for the evaporator finned tube surface, Btu/hr ft² °F

G_{prod} = mass velocity of combustion products based upon minimum free flow area, lbm/ft² hr

c_{prod} = specific heat of combustion products, Btu/lbm °F

μ_{prod} = viscosity of combustion products, lbm/ft hr

K_{prod} = thermal conductivity of combustion products, Btu/ft hr °F

D_{eo} = outer diameter of primary tube surface, in

FIGURE A.2 FINNED TUBE HEAT TRANSFER AND PRESSURE DROP DATA (REPRODUCED FROM KAYS AND LONDON[7])

For finned tubes, the form of $\emptyset_1$ is the following:

$$\text{StPr}^{2/3} = K_1 (\text{Re})^{-K_2} \quad (\text{A.15})$$

If data exist for the surface and tube arrangement of interest, K_1 and K_2 are determined graphically and used in Equation A.15 to determine h_e for any Reynolds number.

In lieu of actual experimental data, there are available generalized correlations for predicting performance of finned tube heat exchangers [11,13]. These were arrived at by determining the variables effecting finned tube performance (i.e. flow conditions, tube arrangement, and fin geometry) and performing a regression analysis for all of the available experimental data. Although the error resulting from this method can be as high as 10 to 20% [11,13], it is still a valuable optimization tool in that it accurately reflects the relative effects of varying important design variables.

The correlation incorporated into the heat pipe model was developed by ESCOA [13] and has the following form:

$$\text{StPr}^{2/3} = 0.25 (\text{Re})^{-0.35} C_1 C_2 (D_f/D_{eo})^{0.5} (\bar{T}_{\text{prod}}/\bar{T}_s)^{0.25} \quad (\text{A.16})$$

where

$$C_1 = 0.35 + 0.65 e^{(-0.25 l_f/s_f)} \quad \text{staggered arrays} \quad (\text{A.17})$$

$$= 0.20 + 0.65 e^{(-0.25 l_f/s_f)} \quad \text{inline arrays} \quad (\text{A.18})$$

$$C_2 = 0.70 + [0.70 - 0.8e^{(-0.15 N_r^2)}] e^{(-P_1/P_t)} \quad \text{staggered} \quad (\text{A.19})$$

$$= 1.1 - [0.75 - 1.5e^{(-0.70 N_r^2)}] e^{(-2.0 P_1/P_t)} \quad \text{inline} \quad (\text{A.20})$$

l_f = fin length, in

s_f = fin spacing, in⁻¹

P_t = transverse tube pitch, in

P_l = longitudinal tube pitch, in

$\bar{T}_{\text{prod}}$ = average combustion products temperature across array, °R

$$= (T_{\text{flame}} - T_{\text{flue}})/2 \quad (\text{A.21})$$

$$\begin{aligned}\bar{T}_s &= \text{average fin temperature, } ^\circ\text{R} \\ &= T_{ve} + 0.3(\bar{T}_{prod} - T_{ve})\end{aligned}\quad (\text{A.22})$$

$$D_f = \text{fin diameter, in}$$

Given the evaporator geometry, it is then only necessary to solve Equation A.16 for the Stanton number and in turn Equation A.14 for h_e .

Once the convection heat transfer coefficient is known, the overall surface effectiveness η_o must be determined in order to account for the temperature gradient in the fin. This is a weighted average of the effectiveness of the primary tube surface (equal to 1) and the fin effectiveness η_f .

$$\eta_o = 1 - \frac{A_f(1 - \eta_f)}{A} \quad (\text{A.23})$$

where

$$A = \text{total outer surface area of evaporator (including fin area), ft}^2$$

$$A_f = \text{total fin area, ft}^2$$

The effectiveness for circular fins is obtained according to the analysis presented by Schneider [26]:

$$\eta_f = \frac{\sqrt{2}/\xi}{(1 + D_{2c}/D_{eo})} \left[\frac{I_1(R_a \xi) K_1(R_b \xi) - I_1(R_b \xi) K_1(R_a \xi)}{I_1(R_a \xi) K_0(R_b \xi) + I_0(R_b \xi) K_1(R_a \xi)} \right] \quad (\text{A.24})$$

where

$$R_a = \sqrt{2}/(1 - D_{eo}/D_{2c})$$

$$R_b = (D_{eo}/D_{2c})$$

$$D_{2c} = D_f + t_f/2$$

$$\xi = (D_{2c} - D_{eo}) \sqrt{\frac{h_e}{K_{ew} t_f}} \quad (\text{A.25})$$

$$t_f = \text{fin thickness, in}$$

$$K_{ew} = \text{thermal conductivity of fin material (evaporator wall), Btu/hr ft } ^\circ\text{F}$$

$$I_0, I_1 = \text{zero and first order Bessel function of the first kind}$$

$$K_0, K_1 = \text{zero and first order Bessel function of the second kind}$$

The thermal resistance of the convective boundary layer on the outer surface of the evaporator is then:

$$R_{\text{conv}} = \frac{1}{h_e A \eta_o} \quad (\text{A.26})$$

A.1.2 Evaporator Wall Resistance

The thermal resistance of a cylindrical wall is simply:

$$R_{\text{ewall}} = \frac{\ln(D_{eo}/D_{ei})}{M 2\pi l_e K_{ew}} \quad (\text{A.27})$$

where

D_{ei} = inner diameter of evaporator tube, in

l_e = length of evaporator tube, ft

M = number of evaporator tubes

A.1.3 Evaporator Boiling

For a wickless heat pipe operating in the heat flux range required for this application, the desired evaporation mechanism is nucleate boiling. The analysis employed here is taken primarily from the work of Rohsenow [2] and assumes : 1) the fluid in the evaporator is not flowing at any appreciable velocity; and 2) the geometry of the boiling surface does not significantly affect the nucleate boiling mechanism [5].

The nucleate boiling heat transfer coefficient is expressed in terms of the excess temperature above the boiling point:

$$h_{\text{boil}} = \frac{Q}{A_b \Delta T_x} \quad (\text{A.28})$$

where

ΔT_x = excess temperature

= $(T_{ei} - T_{\text{sat}}) = (T_{ei} - T_{ve})$

T_{ei} = temperature of inner wall of evaporator, $^{\circ}\text{F}$

T_{sat} = saturation temperature of heat pipe fluid in evaporator, $^{\circ}\text{F}$

T_{ve} = temperature of evaporator vapor, $^{\circ}\text{F}$

$$A_b = \text{boiling surface area, in}^2$$

The excess temperature is calculated using an experimental correlation developed by Rohsenow:

$$\frac{c\Delta T_x}{h_{fg} Pr^{1.7}} = C_{sf} \left[\frac{Q/A_b}{\mu h_{fg}} \sqrt{\frac{g_o \sigma}{g(\rho - \rho_v)}} \right]^{0.33} \quad (A.29)$$

where

C_{sf} = empirical surface/fluid combination coefficient

σ = surface tension coefficient for heat pipe fluid, lbf/ft

c = specific heat of heat pipe fluid, Btu/lbm OF

h_{fg} = latent heat of vaporization of heat pipe fluid, Btu/lbm

μ = viscosity of heat pipe fluid, lbm/ft hr

ρ = density of heat pipe fluid, lbm/ft³

ρ_v = density of heat pipe vapor, lbm/ft³

g_o = constant, 32.17 lbm/ft/lb_fsec²

g = gravitational acceleration, ft/sec²

The boiling heat transfer area A_b will depend upon the number and size of the evaporator tubes and also the level to which they are filled with heat pipe fluid:

$$A_b = \pi D_{ei} l_e M C_{fill} \quad (A.30)$$

where

C_{fill} = a constant representing the level to which the evaporator tubes are filled with heat transfer fluid

The boiling resistance to heat transfer is then:

$$R_{boil} = \frac{1}{h_{boil} A_b} \quad (A.31)$$

In order to insure that the heat pipe is indeed operating in the nucleate boiling regime, the maximum and minimum nucleate boiling heat fluxes are calculated. For predicting the maximum heat flux, Rohsenow and Griffith [6] developed the following correlation:

$$\frac{(Q/A_b)_{\max}}{\rho_v h_{fg}} = 143g^{0.25} \left[\frac{\rho - \rho_v}{\rho_v} \right]^{0.6} \quad (\text{A.32})$$

The lower nucleate boiling range will vary depending upon the magnitude of natural convection heat transfer but can be assumed to be between 10 and 20% of the maximum predicted by Equation (A.32) [5].

A.1.4 Pressure Drop

The total vapor pressure drop along the length of the heat pipe is the sum of the average pressure drop in the evaporator, the pressure drop in the adiabatic section, and the average pressure drop in the condenser section as illustrated in Figure A.3. For the case of laminar flow, assuming negligible dynamic effect, the total pressure drop is:

$$P_e - P_c = \frac{F_e Q l_e}{3} + F_a Q l_a + \frac{F_c Q l_c}{3} \quad (\text{A.33})$$

where

$$F = \frac{f \text{Re}_v \mu_v}{2 r_h^2 A_v \rho_v h_{fg}} \quad (\text{A.34})$$

f = friction factor

Re_v = vapor Reynolds number

r_h = hydraulic radius, in

μ_v = vapor viscosity, lb_f/ft sec

P_e and P_c = evaporator and condenser pressure, psi

l_e , l_a and l_c = length of the evaporator, adiabatic, and condensing sections

For the turbulent flow case, accounting for the dynamic effect:

$$P_e - P_c = (0.267 F'_e Q^{7/4} l_e + 0.677 D_e Q^2) + F'_a Q^{7/4} l_a + (0.267 F'_c Q^{7/4} l_c + 0.677 D_c Q^2) \quad (\text{A.35})$$

where

$$F' = \left[\frac{0.0396 \mu_v}{A_v r_h^2 \rho_v h_{fg}} \right] \left[\frac{2 r_h}{A_v \mu_v h_{fg}} \right]^{.75} \quad (\text{A.36})$$

$$D = \frac{1}{A_v^2 \rho_v h_{fg}} \quad (\text{A.37})$$

FIGURE A.3 PRESSURE DROP AND MASS FLOW IN THE HEAT PIPE

Knowing the pressure drop, the Clausius-Clapeyron equation is used to relate temperature and pressure:

$$T_{ve} - T_{vc} = \frac{T_v (P_e - P_c)}{\rho_v h_{fg} J} \quad (A.38)$$

where

J = mechanical equivalent of heat, 778 ft lb_f/Btu

The thermal resistance attributable to the pressure drop is then:

$$R_{pdrop} = \frac{Q J h_{fg} \rho_v}{T_v (P_e - P_c)} \quad (A.39)$$

A.1.5 Film Condensation

The film heat transfer coefficient is calculated assuming the following:

1. laminar film condensation
2. fully mixed backup tank
3. linear temperature distribution in film
4. negligible momentum effects in film
5. the film surface is at the saturation temperature of the vapor

The analysis, following the derivations of Rohsenow [2] and McAdams [4], accounts for the effect of shear stress at the liquid-vapor interface and results in the following equation for the average film heat transfer coefficient:

$$\bar{h}_{film} = \frac{h_{fg}}{\ell_c \Delta T} \left[\frac{g \rho (\rho - \rho_v) \delta_l^3}{3 \mu} - \frac{\tau \delta_l^2 \rho}{2 \mu} \right] \quad (A.40)$$

where

$$\Delta T = (T_{vc} - T_{co})$$

T_{co} = temperature of outer surface of tank wall, °F

h'_{fg} = average enthalpy change of the vapor in condensing to liquid and subcooling to average liquid temperature (assuming linear temperature gradient in film), Btu/lbm

$$= h_{fg} + \frac{3}{8} C (T_{ve} - T_{co}) \quad (A.41)$$

$$\begin{aligned}\tau_v &= \text{vapor shear stress} \\ &= \frac{f G_{vm}^2}{2\rho_v}\end{aligned}\tag{A.42}$$

f = friction factor

G_{vm} = average vapor mass velocity, $\text{lbm/ft}^2\text{hr}$
 $= 0.4 G_{in}$

$$\begin{aligned}G_{in} &= \text{mass velocity of vapor entering condenser, } \text{lbm/ft}^2\text{hr} \\ \delta_\ell &= \left\{ \left[\frac{\Delta T k_{hf} \ell_c}{h'_{fg}} \right] / \left[\frac{g\rho(\rho - \rho_v)\delta_\ell}{4\mu} \frac{\tau_v \rho}{3\mu} \right] \right\}^{1/3}\end{aligned}\tag{A.43}$$

k_{hf} = thermal conductivity of heat pipe fluid, $\text{Btu/hr ft}^\circ\text{F}$

The film area will depend upon the geometry of the condenser. If the entire outer tank wall is used for condensing, the area is:

$$A_{\text{film}} = \pi D_t \ell_c \tag{A.44}$$

where

D_t = diameter of backup tank, in

If distinct "condenser channels" are to be used as illustrated in Figure A.4, the area is:

$$A_{\text{film}} = Nw\ell_c \tag{A.45}$$

where

N = number of condensor channels

w = width of channels, in

The resistance to heat transfer of the film is then the following:

$$R_{\text{film}} = \frac{1}{\bar{h}_{\text{film}} A_{\text{film}}}\tag{A.46}$$

Top View Of Tank

FIGURE A.4 BACKUP TANK FINS

A.1.6 Tank Wall Resistance

Because the tank wall thickness is small relative to the tank diameter, its resistance can be estimated as:

$$R_{cwall} = \frac{t_w}{K_{tw} \ell_c \pi D_t} \quad (A.47)$$

where

t_w = tank wall thickness

K_{tw} = thermal conductivity of tank wall, Btu/hr ft °F

Or, if the condenser does not cover the entire tank, but is divided into distinct vertical "channels", the resistance is:

$$R_{cwall} = \frac{t_w}{K_{tw} Nw \ell_c} \quad (A.48)$$

where

w = width of individual channels

Another case of interest is that where the condenser channels are created by brazing a tube of a "D-shaped" or square geometry to the tank wall. Here, the total wall resistance includes that of the tube wall, the bond, and the tank wall as follows:

$$R_{cwall} = \frac{1}{Nw \ell_c} \left[\frac{t_w}{K_{tw}} + \frac{t_b}{K_b} + \frac{K_{tb}}{t_{tb}} \right] \quad (A.48a)$$

where

K_b = thermal conductivity of bond

K_{tb} = thermal conductivity of tube wall

t_b = average bond thickness

t_{tb} = tube wall thickness

A.1.7 Free Convection Inside Tank

Assuming a fully mixed backup tank and a negligible vertical temperature gradient in the tank wall, the free convection heat transfer coefficient can be calculated using the analysis presented by Chapman [1]:

Laminar Case ($Gr < 10^9$)

$$Nu_L = 0.678 (0.952 + Pr)^{-1/4} Pr^{1/2} Gr^{1/4} \quad (A.49)$$

Turbulent Case ($Gr > 10^9$)

$$Nu_L = 0.246 (1 + 0.494 Pr^{2/3})^{-2/5} Pr^{7/15} Gr^{2/5} \quad (A.50)$$

where the Nusselt and Grashoff numbers are defined as:

$$Nu_L = \frac{\bar{h}_{free} \ell_c}{K_{tf}} \quad (A.51)$$

$$Gr = \frac{\ell_c^3 g \beta \Delta T_{free} \rho_t^2}{\mu_t^2} \quad (A.52)$$

μ_t = viscosity of water in tank, $lb_f/ft \text{ hr}$

ρ_t = density of water in tank, lbm/ft^3

K_{tf} = thermal conductivity of water in tank, $Btu/hr \text{ ft } ^\circ F$

ΔT_{free} = surface fluid temperature difference

= $(T_{ci} - T_t) ^\circ F$

T_{ci} = temperature at inner surface of tank wall

$\bar{h}_{free}$ = average heat transfer coefficient for a finite tank height, $Btu/hr \text{ ft}^2 ^\circ F$

If it is assumed that entire tank wall is used as condensing surface, the thermal resistance is:

$$R_{free} = \frac{1}{h_{free} \ell_c \pi D_t} \quad (A.53)$$

If portions of the tank wall are not used for condensing, then the area between consecutive "channels" is treated as a straight fin of constant cross section as shown in Figure A.4. Two heat transfer coefficients are calculated, one for the surface adjacent to a channel of width w , and another average coefficient for a fin of length $W/2$. The average coefficient over the fin is always smaller than that adjacent to the channel since the temperature difference driving force decreases with distance from the fin base. For this case, R_{free} is given by:

$$R_{\text{free}} = 1/(h_{fc} w N l_c + h_{ff} W \eta_c N l_c) \quad (\text{A.54})$$

where

w = condensor channel width, in

W = distance between consecutive channels, in

h_{fc} = free convection coefficient for surface adjacent to channel, Btu/hr ft² °F

h_{ff} = free convection coefficient for fin surface (based upon average fin temperature), Btu/hr ft² °F

$$\eta_c = \text{fin efficiency} = \frac{\tanh(m W/2)}{m W/2} \quad (\text{A.55})$$

$$m^2 = \frac{h_{ff}}{K_{tw} t_w} \quad (\text{A.56})$$

A.2 Performance Limitations

For any given heat pipe configuration there are both thermodynamic and practical design constraints to be addressed. The mechanisms which determine the maximum heat pipe heat transfer rate include sonic and entrainment limitations, and the peak nucleate boiling heat flux. The latter was discussed in Section A.1.3, and the sonic and entrainment limitations are described below in Sections A.2.1 and A.2.2. In addition, the final design must insure that the heat pipe components do not experience excessive pressures and temperatures under the anticipated operating conditions. These calculations are discussed in Section A.2.3. Finally, as described in Section A.2.4, the volume of heat transfer fluid associated with the vapor, and more importantly, held up in the condenser, is estimated to insure proper startup and operation.

A.2.1 Sonic Limitation

The acceleration and deceleration of vapor in the heat pipe (due to vapor addition and removal in the evaporator and condenser) is similar to flow in a converging-diverging nozzle [3]. Pressure decreases in the converging section (evaporator) with an increase in velocity up to the throat (the end of the evaporator), and is recovered in the diverging section (the condenser) with a decrease in velocity. A critical or choked flow condition exists when lowering of the back pressure (lowering the condenser temperature) results in no further increase in the flow rate (heat transfer rate). This occurs when the vapor velocity at the evaporator becomes sonic and the conditions at the condenser are no longer transmitted upstream to the evaporator.

The heat pipe heat flux corresponding to the sonic limit can be determined using the Levy equation [15] as described by Chi [3]. Here, it is assumed that the sonic limit occurs at low vapor densities and high vapor velocities and hence 1) the vapor behaves as an ideal gas; 2) inertial effects dominate; and 3) frictional effects may be neglected. The sonic limit for heat transfer is then:

$$Q_{sl} = A_v \rho_o h_{fg} \left[\frac{v_v R_v T_o}{2 (v_v + 1)} \right]^{1/2} \quad (A.57)$$

where

v_v = vapor specific heat ratio

R_v = gas constant for vapor, Btu/lbm °R

T_o = stagnation temperature, °F

ρ_o = density of vapor at stagnation conditions, lbm/ft³

A_v = vapor flow area in evaporator, in²

The stagnation state can be found if the temperature at the upstream end of the evaporator (where velocity = 0) is known. In this case it is not but it can be assumed equal to T_{ve} which is in effect the average vapor temperature in the evaporator.

A.2.2 Entrainment Limitation

Entrainment occurs in a heat pipe when the velocity of the vapor flowing over the returning condensate produces a shear force at the liquid-vapor interface large enough to pull droplets of liquid into the vapor stream. The mechanism is "self-feeding" in that the onset of entrainment restricts the condensate return, which increases the evaporator temperature, which in turn increases the entrainment. This can result in noisy liquid slugging, reduced heat transfer, and possibly the complete drying out of the evaporator. In any event, prudent design dictates operation below the point at which entrainment begins.

The onset of entrainment can be predicted using the criterion developed experimentally by Steen [16]:

$$\frac{j_g \mu_g}{\sigma} \left[\frac{\rho_g}{\rho_f} \right]^{1/2} = \pi_2 > 2.46 \times 10^{-4} \quad (A.58)$$

where

j_g = volumetric flux of gas, ft/hr

μ_g = viscosity of gas, lbf/ft hr

σ = surface tension coefficient, lbf/ft

ρ_g = density of gas, lbm/ft³

ρ_f = density of liquid, lbm/ft³

Steen's parameter π_2 can be interpreted as a dimensionless "critical gas velocity", which must remain below 2.46×10^{-4} to insure the absence of entrainment. Equation A.58 is applicable as long as the liquid does not occupy a major portion of the duct, and viscous forces in the liquid are not significant. It should also be noted that experimental observations have shown that the critical gas velocity is relatively insensitive to duct dimensions, orientation, and liquid flow rate [17].

A.2.3 Pressure and Temperatures

In order to select the appropriate materials for construction and to insure safe, reliable operation, the internal pressure and the temperature at key points must be determined. The temperature distribution along the heat pipe assembly can be calculated by multiplying the overall heat transfer rate (see Section A.1) by the applicable thermal resistance. For example, the temperature at the outer condenser surface (inner tank wall) is given by:

$$T_{co} = T_t + Q R_{free} \quad (A.59)$$

The pressure inside of the heat pipe is simply that corresponding to the saturation temperature of the heat pipe vapor. For the purpose of the computer program, the saturation pressure-temperature relationship taken from the steam tables was curve fit as follows:

$$\ln \hat{P}_s = \sum_{j=1}^n b_j (T_s/10)^{j-1} \quad (A.60)$$

$$b_1 = -2.4147$$

$$b_4 = 7.3911 \text{ E-4}$$

$$b_2 = 0.71598$$

$$b_5 = -1.2275 \text{ E-5}$$

$$b_3 = -0.026808$$

$$b_6 = 8.9646 \text{ E-8}$$

where

$\hat{P}_s$ = saturation pressure, psi

T_s = saturation temperature, °C

A.2.4 Fluid Level Considerations

Another practical design consideration is the amount of heat transfer fluid needed in the heat pipe to insure the desired performance during firing and startup. In particular, if an excessive volume of fluid is held up in the condenser film and the flowing vapor, it is possible that when cooled, the evaporator will be flooded, eliminating the vapor flow area and causing "slugging" during startup. The problem is then to determine to what level to fill the evaporator so that there is adequate fluid for heat transfer and yet sufficient vapor flow area to prevent slugging during startup and any entrainment during normal operation.

The volume of liquid associated with the vapor flow during firing is given by:

$$V_l = \frac{V_v \rho_v}{\rho} \quad (\text{A.61})$$

where

V_v = volume of vapor during firing, ft^3

V_l = volume of liquid associated with corresponding
volume of vapor, ft^3

The volume of fluid entrained in the condensing film per unit width of condenser is determined by integrating the film thickness over the vertical length of tank wall which yields the following:

$$V_f = 0.8 \sqrt{\frac{4 K_{hf} \mu \Delta T \ell_c}{g \rho (\rho - \rho_v) h_{fg}}} \quad (\text{A.62})$$

where

V_f = volume of fluid entrained in condensing film, ft^3

A.3 Combustion Conditions

For the purpose of modeling heat pipe heat transfer, the combustion conditions are characterized by the temperature and mass flow rate of the products, which in turn are set by specifying the "heater rating" and the percent theoretical air used in the burner. As is typically the case for gas fired appliances, the heater rating is based upon the higher heating value of the fuel, which for natural gas (methane) is assumed equal to 23,861 Btu/lbm. However, if condensation of the water vapor in the combustion products is to be avoided, then only the lower heating value of the fuel (assumed equal to 21,502 Btu/lbm) is available to heat the tank. Thus it can be said that the efficiency of the heater based upon the higher heating value is inherently limited to 90%. Once the heater rating has been

specified, the mass of fuel burned per unit time is determined simply by dividing by the higher heating value.

The temperature and mass flow rate of the resulting combustion products depend solely on the percent theoretical air used in the burner. The reaction of interest is expressed as follows:

Here, x represents the percent theoretical air, which is equal to 100 for the "chemically correct" or "complete-combustion" mixture, and N_2 represents the apparent nitrogen, which includes the trace inert gases found in air. Now, using Equation A.63 and the molecular weights of the reactants and products, the mass of products per pound of methane burned is given by:

$$M_{\text{prod}} = 1 + 17.14 \left[\frac{x}{100} \right] \quad (\text{A.64})$$

The mass flow rate of the products per pound of methane burned is then:

$$\dot{m}_{\text{prod}} = \dot{m}_{\text{gas}} M_{\text{prod}} \quad (\text{A.65})$$

It should be noted that the combustion equation shown above is valid for mixtures of 100 percent or greater theoretical air. For the purpose of optimizing the fuel-air mixtures, it is of interest to examine mixtures below 100 percent. The combustion equation then becomes:

Equations A.64 and A.65 for calculating the mass flow rate of the combustion products per pound of methane burned apply regardless of whether the percent theoretical air is greater or less than the 100 percent.

The flame temperature attained during combustion can be determined using a First Law energy balance, which for constant pressure combustion states that the enthalpy of the reactants equals that of the products. When expressed in terms of the specific enthalpies of the individual reactants, the energy balance can be solved for the enthalpy per mole of products as follows:

$$\bar{h}_p = \frac{1}{n_p} \left[\bar{m}_f (h_f - h'_f - h_{rp}) + m_a (h_a - h'_a) \right] + \bar{h}'_p \quad (\text{A.67})$$

where

- h_p = enthalpy per mole of combustion gases at the adiabatic flame temperature
- $\bar{h}_p'$ = enthalpy per mole of combustion gases at 77°F
- h_f = enthalpy of methane at the original temperature
- h_f' = enthalpy of methane at 77°F
- h_A = enthalpy of air at the original temperature
- h_A' = enthalpy of air at 77°F
- $\bar{m}_f$ = molecular weight of methane
- m_a = number of pounds of air
- n_p = number of moles of products
- h_{rp} = heat of combustion of methane

The problem then is to determine the temperature of the products corresponding to the enthalpy which results when the heat of combustion of the fuel is added to the original enthalpy of the reactants. The enthalpy of various common gases at temperatures ranging from 520 to 7000 °R is available in tabular form [18], and can be used to determine the enthalpy of the product mixture at any temperature. Thus, the flame temperature is determined iteratively by calculating the enthalpy of the product mixture for an assumed temperature using the tabular data, comparing the result to the enthalpy of the reactants calculated in Equation A.68, and repeating the process until the correct temperature is found. This was done for methane for air-fuel mixtures ranging from 80 to 400 percent theoretical air and the results are shown plotted in Figure A.5. For use in the computer program, the data was curve fit as follows:

For $x < 100\%$

$$T_{\text{flame}} = 4189 + 632.93 x' + 114.93 x'^2 \quad (\text{A.68})$$

For $x > 100\%$

$$T_{\text{flame}} = 4189 - 2698.0 x' - 508.81 x'^2 - 400.50 x'^3 \quad (\text{A.69})$$

where

$$\begin{aligned} T_{\text{flame}} &= \text{flame temperature } ^\circ\text{R} \\ x' &= 1 - 100/x \end{aligned}$$

It is important to note that the theoretical flame temperature calculated here will not be attained in the actual burner because the reaction is never

FIGURE A.5 FLAME TEMPERATURE AS A FUNCTION OF PERCENT THEORETICAL AIR FOR THE COMBUSTION OF METHANE

complete. In addition, even if the reaction were complete, radiative and convective losses lower the combustion products temperature before they reach the evaporator. It was deemed impractical to attempt to quantify these effects analytically, so instead, the analysis will be modified according to results obtained in the laboratory.

The heat transfer calculations described in Section A.1. assume that the specific heat of the combustion products is known. For the purpose of many engineering applications, the value for pure air can be used without encountering a significant error. However, for this computer application, where a wide range of temperatures and fuel-air mixtures may be encountered, a much greater degree of accuracy is required.

The specific heat at constant pressure of a mixture of gases can be calculated by means of the Gibbs-Dalton Law from the composition of the product mixture and the specific heat of the individual components. Simply stated, the specific heat of the mixture is equal to the sum of the specific heat of the constituents evaluated at the temperature of interest and weighted according to their respective mole fractions.

$$c_{\text{prod}} = W_{\text{prod}} \sum_i x_i c_{pi} \quad (\text{A.70})$$

where

x_i = mole fraction of the i th component

W_{prod} = molecular weight of the product mixture, lbm/lb mole

c_{pi} = mole specific heat of the i th component, Btu/lb mole $^{\circ}\text{R}$

Specific heat equations for the combustion product constituents are available and are included in Appendix B. The mole fractions can be determined from Equations A.63 and A.66 for any percent theoretical air. These equations can also be rearranged to yield the following equation for the molecular weight of the mixture, W_{prod} , as a function of percent theoretical air:

$$W_{\text{prod}} = \frac{[1 + 17.137(x/100)] 16.043}{1 + 9.528(x/100)} \quad (\text{A.71})$$

Another important consideration related to combustion is the pressure drop across the finned tube evaporator. It is assumed that the appliance will incorporate a draft hood and hence, should operate with neutral (atmospheric) pressure at the inlet to the exhaust vent. Any excess capacity resulting from the natural draft in the vent will draw in draft hood dilution and prevent draft hood spillage. It is of interest then to compare the pressure drop across the evaporator with an estimate of the buoyancy driving force associated with the hot combustion gases moving from the

actual point of combustion to the draft hood. The intent is to insure no backup of combustion products or unburned fuel below the evaporator.

Pressure drop data for flow across finned tubes are available in two forms, analogous to the corresponding heat transfer data for finned tubes discussed in Section A.1.1. Experimental data for the specific tube arrangement and extended surface of interest are typically plotted as a function of Reynolds number along with the heat transfer data. The relationship is of the same form as Equation A.15:

$$f = L_1 (Re)^{-L_2} \quad (A.72)$$

where

f = mean friction factor

L_1 and L_2 = constants

The friction factor can also be predicted using generalized correlations developed in the same manner as the heat transfer correlation given by Equation A.16. The friction factor correlation used here was developed by ESCOA [3] and has the following form:

$$f = C_3 C_4 C_5 (D_f/D_{eo})^{C_6} \quad (A.73)$$

where

$$\begin{aligned} C_3 &= 0.07 + 8.0 Re^{-0.45} \\ C_4 &= 0.11(0.05 P_t/D_{eo})^{[-0.7(1_f/s_f)^{0.20}]} \text{ staggered arrays} \\ &= 0.08(0.15 P_t/D_{eo})^{[-1.1(1_f/s_f)^{0.15}]} \text{ inline arrays} \\ C_5 &= 1.1 + [1.8 - 2.1 e^{(-0.15 N_r^2)}] e^{(-2.0 P_1/P_t)} \\ &\quad - [0.7 - 0.8 e^{(-0.15 N_r^2)}] e^{(-0.6 P_k/P_t)} \text{ staggered} \\ &= 1.6 - [0.75 - 1.5 e^{(-0.7 N_r)}] e^{-0.2(P_k/P_t)^2} \text{ inline} \\ C_6 &= 0.5 \text{ staggered} \\ &= 1.0 \text{ inline} \end{aligned}$$

Once the friction factor is known, the pressure drop can then be calculated as shown by Kays and London [7]:

$$\frac{\Delta P}{P_1} = \frac{G^2 v_1}{2g P_1} \left[(1 + \sigma^2) \left(\frac{v_2}{v_1} - 1 \right) + f \frac{A}{A_c} \frac{v_m}{v_1} \right] \quad (A.74)$$

where

- G = mass velocity, lbm/hr ft²
- v_1 = specific volume of gases entering the evaporator, ft³/lbm
- v_2 = specific volume of gases exiting the evaporator
- v_m = mean specific volume
- σ = ratio of free flow area to frontal area
- A = total surface area, ft²
- A_c = free flow area, ft²
- P_1 = pressure at entrance to evaporator

In Equation A.73, the first term inside of the brackets represents flow acceleration, and the second term represents core friction. It should be noted that entrance and exit loss are accounted for in the friction factor.

The buoyancy driving force in the combustion chamber, DF_{prod} , (i.e. below the draft hood) is simply the difference in weight between a column of combustion gases and an equal column of ambient air. The column of combustion gases can be divided into three isothermal segments: 1) a column between the burner and the bottom of the finned tube array, which is assumed to be at the flame temperature; 2) a column the height of the finned tube array itself, at a temperature which is the average of the entrance and exit temperatures; and finally, 3) a column between the top of the array and the draft hood, at a temperature equal to that exiting the finned tube array.

$$DF_{prod} = \rho_o \frac{g}{g_c} (H_1 + H_2 + H_3) - \frac{g}{g_c} (\rho_{flame} H_1 + \rho_{ave} H_2 + \rho_{flue} H_3) \quad (A.75)$$

where

- ρ_o = density of ambient air, lbm/ft³
- ρ_{flame} = density of combustion gases entering finned tube array
- ρ_{ave} = average density of gases across array
- ρ_{flue} = density of gases exiting array
- H_1 = distance between burner and bottom of array, ft
- H_2 = height of finned tube array
- H_3 = distance between top of array and draft hood

Appendix B

This appendix contains equations for the physical properties of water and various gases as a function of absolute temperature. Unless otherwise specified, all curve fits were performed by Atlas personnel under Contract EM-78-C-02-4696.A000.

1. Viscosity of Saturated Water

$$\ln \hat{\mu}_\ell = \sum_{j=1}^n b_j (T/10)^{j-1} \quad \text{centipoise}$$

$$\begin{aligned} b_1 &= 0.69315 & b_2 &= -0.36389 & b_3 &= 0.025868 \\ b_4 &= -0.0012168 & b_5 &= 3.1076\text{E-}5 & b_6 &= -3.2126\text{E-}7 \end{aligned}$$

T in °C

Saturation Pressure

Range: 0- 374.1 °C

RMS = 0.017 centipoise

2. Density of Saturated Water

$$\hat{\rho}_\ell = b_1 + \sum_{j=2}^n b_j (647.27 - T)^{1/(j-1)} \quad \text{lbm/ft}^3$$

$$\begin{aligned} b_1 &= 24.758 & b_2 &= -0.68745 \\ b_3 &= 14.804 & b_4 &= -7.9058 \end{aligned}$$

T in °K

Saturation Pressure

Range: 10 - 374.1 °C

RMS = 0.19 lbm/ft³

3. Thermal Conductivity of Saturated Water

$$\hat{k}_l = \sum_{j=1}^n b_j (T/10)^{j-1} \quad \text{Btu/hr ft } ^\circ\text{F}$$

$$b_1 = 0.32417 \quad b_2 = 0.016393 \quad b_3 = -0.0011969$$

$$b_4 = 3.8806\text{E-}5 \quad b_5 = -5.8900\text{E-}7$$

T in $^\circ\text{C}$

Saturation Pressure

Range: 0 - 374.1 $^\circ\text{C}$

RMS = 0.005 Btu/hr ft $^\circ\text{F}$

4. Specific Heat of Saturated Water

$$\hat{c}_p = \sum_{j=1}^n b_j (647.27 - T)^{(2-j)} \quad \text{Btu/lbm } ^\circ\text{F}$$

$$b_1 = 0.0047628 \quad b_2 = 0.66376 \quad b_3 = 6.1985$$

$$b_4 = -9.5208 \quad b_5 = 9.3237$$

T in $^\circ\text{K}$

Saturation Pressure

Range: 0 - 360 $^\circ\text{C}$

RMS = 0.002 Btu/lbm $^\circ\text{F}$

5. Latent Heat of Saturated Water

$$\hat{h}_{fg} = \sum_{j=1}^n b_j (647.27 - T)^{1/(j+1)} \quad \text{Btu/lbm}$$

$$b_1 = -586.60302 \quad b_2 = 5390.4459$$

$$b_3 = -9789.4194 \quad b_4 = 5255.0767$$

T in °K

Saturation Pressure

Range: 0 - 374.1 °C

RMS = 1.1 Btu/lbm

6. Thermal Expansivity of Water

$$\hat{\beta} = \left[\sum_{j=1}^n b_j T^{j-1} \right] \times 10^{-5} \quad 1/^{\circ}\text{C}$$

$$b_1 = -4.90694 \quad b_2 = 1.47412$$

$$b_3 = -0.0115287 \quad b_4 = 4.82008\text{E-}5$$

T in °C

Pressure = 1 Atmosphere

Range: 8 - 100 °C

RMS = 0.5%

7. Density of Steam

$$\ln \hat{\rho}_v = \sum_{j=1}^n b_j (T/10)^{j-1} \quad \text{lbm/ft}^3$$

$$b_1 = -8.1528 \quad b_2 = 0.71923 \quad b_3 = -0.033825$$

$$b_4 = 0.0013395 \quad b_5 = -3.2505\text{E-}5 \quad b_6 = 3.3834\text{E-}7$$

T in °C

Saturation Pressure

Range: 10 - 374.1 °C

RMS = 0.0012 lbm/ft³

8. Viscosity of Steam

$$\frac{1}{\hat{\mu}_v} = \sum_{j=1}^n b_j (647.27 - T)^{j-1}$$

$$b_1 = 35.1511 \quad b_2 = 2.2171$$

$$b_3 = -0.074653 \quad b_4 = 0.0018579$$

T in °K

Saturation Pressure

Range: -17.8 - 399 °C

RMS = 0.38 1/centipoise

9. Surface Tension Coefficient of Water

$$\hat{\sigma} = \left[\sum_{j=1}^n b_j T^{j-1} \right] \times 10^{-4} \text{ lbf/ft}$$

$$b_1 = 51.694 \quad b_2 = -9.3907E-2$$

$$b_3 = -2.3677E-4 \quad b_4 = 2.9513E-7$$

T in °C

Saturation Pressure

Range: 0 - 360 °C

RMS = 0.12%

10. Specific Heat of Oxygen*

$$C_{pO_2} = 11.515 - \frac{172}{\sqrt{T}} + \frac{1530}{T} \text{ Btu/mole } ^\circ R$$

T in °R

Range: 540 - 5000 °R

Maximum Error = 1.1%

Source: Ref. 27

11. Specific Heat of Nitrogen*

$$C_{pN_2} = 9.47 - \frac{3.47(10^3)}{T} + \frac{1.16(10^6)}{T^2} \text{ Btu/mole } ^\circ R$$

T in °R

Range: 540 - 9000 °R

Maximum Error: 1.7%

Source: Ref. 27

12. Specific Heat of Steam^{*}

$$C_{pH_2O} = 19.86 - \frac{597}{\sqrt{T}} + \frac{7,500}{T} \quad \text{Btu/mole } ^\circ\text{R}$$

T in $^\circ\text{R}$

Range: 540 - 5400 $^\circ\text{R}$

Maximum Error: 1.8%

Source: Ref. 27

13. Specific Heat of Carbon Dioxide^{*}

$$C_{pCO_2} = 16.2 - \frac{6.53(10^3)}{T} + \frac{1.41(10^6)}{T^2} \quad \text{Btu/mole } ^\circ\text{R}$$

T in $^\circ\text{R}$

Range: 540 - 6300 $^\circ\text{R}$

Maximum Error = 0.8%

Source: Ref. 27

* Assuming ideal gas state

APPENDIX C

This appendix contains a complete listing of the heat pipe computer model developed under Contract DE-AC02-78CS34696-A002 and cited in this report.

```

      PROGRAM DRIVER(INPUT,OUTPUT,PAR,TAPE1=PAR,TAPE2=OUTPUT)
C*****
C      THIS PROGRAM IS DESIGNED TO DRIVE
C      THE HPIPE HEAT PIPE VERSION ESCOA
C      12/78 ND
C*****
      DIMENSION XIN(2),OUT(65),PAR(39)
      INFO = -1
      READ(1,*) PAR
      WRITE(2,1006) PAR
1006  FORMAT(* PARAMETERS*/10(4E10.3/))
      READ(1,*) XIN(1),XIN(2)

C
C----- CALL HEAT PIPE
C
      CALL HPIPE(XIN,OUT,PAR,INFO)
      INFO = INFO + 1

C
C----- OUTPUTS
C
      WRITE(2,3000) (OUT(I),I=1,37)
3000  FORMAT(1H ,*Q      =*,F10.0,*[BTU/HR]*/* MDOT    =*,E10.3,*[LB/HR]*/
1/* PRESUR =*F6.2*[PSI]*/* TFLUE  =*F8.1*[DEG F]*/
1* TFLAME  =*,F8.1/* TEO      =*F8.1/* TEI      =*F8.1/
1* TVE     =*,F8.1/* TVC     =*,F8.1/* TCO     =*,F8.1/* TCI     =*,
2F8.1/* TCIC  =*,F8.1/* PTOT   =*,E10.3,*[PSI]*/
2* PAVE    =*,E10.3/* PAD     =*E10.3/* PAVC    =*E10.3/
4* REVE    =*,F10.0/* REVA    =*,F10.0/* REVC    =*,F10.0/
5* XMACHE  =*,E10.3/* QSONIC  =*E10.3*[BTU/HR]*/
9* ETE     =*E10.3/* ETA     =*E10.3/* ETC     =*E10.3/
9* QBMAXR  =*E10.3*[BTU/HR]*/* QBMAXZ =*E10.3/* QBMIN  =*E10.3/
9* HFILM   =*E10.3*[BTU/HR DEG F FT2]*/* DTFM    =*E10.3/
1* DTL     =*E10.3*[FT]*/* DTL    =*E10.3/* VCON    =*E10.3*[IN3]*/
9* REFLML  =*E10.3/
9* HFRET   =*,E10.3/* HFREF   =*E10.3/* FIN      =*F5.3/* HBOIL  =*
9E10.3)
      WRITE(2,3003) (OUT(I),I=38,65)
3003  FORMAT(* VFGAS  =*E10.3*[FT3/HR]*/* MFPROD =*E10.3*[LB/HR]*/
1* REBURN  =*E10.3/* HBURN   =*E10.3*[BTU/FT2 HR DEGF]*/
2* TSA     =*F7.3*[FT2]*/* FEFF  =*F6.3/* SEFF   =*F6.3/
3* UAX     =*E10.3/* NTUX    =*F6.3/* UAPIPE =*E10.3/* XEFF   =*
4F6.3/* RSBURN =*E10.3*[HR DEGF/BTU]*/* RSEW    =*E10.3/* RSBO   =*
5E10.3/* RSPD  =*E10.3/* RSFLM  =*E10.3/* RSCB   =*E10.3
6/* RSFRE   =*E10.3/* ICOUNT =*F4.0/* FFAC    =*F7.4/
6* DRAFT    =*F7.4* [IN. OF H2O]*/* PLOSS  =*F7.4* [IN. OF H2O]*
7/* ANET    =*F7.4*[FT2]*/* VEVF    =*E10.4*[IN3]*/* VADB    =*
7E10.4*[IN3]*/* VSTEAM =*E10.4*[IN3]*
7/* VTOT    =*E10.4*[IN3]*/* OEF    =*F6.3)
      STOP
      END

```

```

        PROGRAM DATAPT(INPUT,OUTPUT,PAR,QHPIPE,TAPE1=PAR,
1              TAPE2=OUTPUT,TAPE7=QHPIPE,TAPE5=INPUT)
C*****
C      THIS PROGRAM IS DESIGNED TO DRIVE THE HPIPE
C      HEAT PIPE AND GENERATE A PERFORMANCE MAP GIVEN
C      PTA, TANK TEMP. ,DELT'S, AND MATRIX SIZE
C      (1/25/79) ND
C*****
        DIMENSION XIN(2),OUT(68),PAR(37),Q(10,10)
        READ(1,*) PAR
        WRITE(2,1006) PAR
1006 FORMAT(* PARAMETERS*
1/* RATE      =*F8.0* [BTU/HR]*,T35,* BP      =*F6.2* [IN HG]*
1/* TAMB      =*F7.1* [DEG F]*,T35,* AD      =*F6.3* [FT2]*
1/* HCC       =*F6.2* [IN]*,T35,* M          =*F3.0
1/* NR        =*F3.0,T35,* ISTAG =*F3.0
1/* IFIN      =*F3.0,T35,* XLE      =*F6.3* [FT]*
1/* DFIN      =*F6.3* [IN]*,T35,* DOE      =*F6.3* [IN]*
1/* DIE       =*F6.3* [IN]*,T35,* TFIN     =*F6.3* [IN]*
1/* WFIN      =*F6.3* [IN]*,T35,* NF       =*F6.3* [1/IN]*
1/* PT        =*F5.2* [IN]*,T35,* PL       =*F5.2* [IN]*
1/* XC        =*F6.3* [FT2]*,T35,* RFO     =*F6.3* [HR FT2 DEG F/BTU]*
1/* RFI       =*F6.3* [FT2 HR DEG F/BTU]*,T35,* XKPE =*F6.1
1* [BTU/HR FT DEG F]*
1/* CSF       =*F6.3,T35,* FF          =*F4.2
1/* XLA       =*F5.2* [FT]*,T35,* DIA      =*F5.2* [IN]*
1/* XLC       =*F5.2* [FT]*,T35,* N        =*F3.0
1/* TTW       =*F6.3* [IN]*,T35,* DEL      =*F6.3* [IN]*
1/* W         =*F6.3* [IN]*,T35,* XKPC     =*F6.1* [BTU/HR FT DEG F]*
1/* RCON      =*F7.4* [BTU/LBM DEG R]*,T35,* GAMV =*F5.2
1/* CVRGQ     =*F5.2,T35,* CVRGT =*F5.2
1/* IMAX      =*F3.0)
C
C----- READ MATRIX PARAMETERS
C
C      TFL IS NOW PTA
C      READ(5,*) TFL,DELTF,M,TTL,DELTT,N
C
C----- FILL MATRIX
C
        INFO = -1
        DO 1 I=1,M
        K = I - 1
        XIN(1) = TFL + DELTF * K
        DO 1 J=1,N
        L = J - 1
        XIN(2) = TTL + DELTT * L
        CALL HPIPE(XIN,OUT,PAR,INFO)
        INFO = 1
        Q(I,J) = OUT(1)
1      CONTINUE
C
C----- WRITE MATRIX
C

```

```
      WRITE(7,4444) TFL,DELTF,M,TTL,DELTT,N
4444  FORMAT(1H ,F6.1,F6.1,2X,I2,F6.1,F6.1,2X,I2)
      DO 2 I=1,M
      WRITE(7,3000) (Q(I,J),J=1,N)
3000  FORMAT(1H ,(10F8.0))
      2  CONTINUE
      STOP
      END
```

```

      SUBROUTINE HPIPE(XIN,OUT,PAR,INFO)
      DIMENSION XIN(2),OUT(65),PAR(39),C(6),P(3),REV(3),Z(14),
1  XLABEL(3),XLIM(6)
      DATA XLABEL/4HEVAP,4HADTC,4HCOND/,HHV/23861./,IERR/2/
      REAL MFGAS,MFPROD,K,NTUX,LF,JC,NF
C*****
C      HEAT PIPE/BURNER/XCHNGR MODEL (4/79)
C      BASED ON THE FOLLOWING ASSUMPTIONS
C      1. INCOMPRESSIBLE FLOW IN ALL SECTIONS
C      2. UNIFORM HEAT FLUX AT EVAP AND COND
C      3. 100% PURE METHANE GAS
C      4. COM PROD PROP ARE THOSE OF PURE AIR
C      5. RATE BASED ON HIGHER HEATING VALUE
C      6. LAMINAR FILM CONDENSATION IN COND CHANNELS
C      7. NEGLECT MOMENTUM EFFECTS IN FILM
C      8. LINEAR TEMP DIST IN FILM
C      9. FULLY MIXED BACKUP TANK
C      10. NUCLEATE BOILING
C      11. ESCOA XCHGR DATA USED
C      12. VAPOR TEMP EQUAL TO TEMP AT VAPOR-FLUID INTERFACE
C      ENGLISH UNITS ARE USED
C*****
      WRITE(IERR,9876)
      9876 FORMAT(1H ,*++++++EPIPE SUMMARY++++++*)
C
C-----SET INPUTS
C
      ICOUNT = 0
      PTA = XIN(1)
      TT = XIN(2)
      IF(INFO.NE.-1) GOTO 101
C
C-----EQUATE PARAMETERS
C
      RATE = PAR(1)
      BP = PAR(2)
      TAMBR = PAR(3) + 459.7
      HB = PAR(4)/12.
      HA = PAR(5)/12.
      M = INT(PAR(6))
      Z(6) = FLOAT(M)
      NR = INT(PAR(7))
      ISTAG = INT(PAR(8))
C      ISTAG 1=INLINE,2=STAGGERED
      IFIN = INT(PAR(9))
C      IFIN 1=SOLID,2=SEGMENTED
      XLE = PAR(10)
      Z(7) = XLE
      DFIN = PAR(11)
      DOE = PAR(12)
      DIE = PAR(13)
      TFIN = PAR(14)
      WFIN = PAR(15)

```

```

NF      = PAR(16)
PT      = PAR(17)
PL      = PAR(18)
XC      = PAR(19)
RFO     = PAR(20)
RFI     = PAR(21)
XKPE    = PAR(22)
CSF     = PAR(23)
Z(5)    = CSF
FF      = PAR(24)
Z(9)    = FF
XLA     = PAR(25)
Z(10)   = XLA
DIA     = PAR(26)
XLC     = PAR(27)
Z(1)    = XLC
N       = INT(PAR(28))
Z(2)    = FLOAT(N)
TTW     = PAR(29)/12.
Z(13)   = TTW
DEL     = PAR(30)/12.
Z(4)    = DEL
W       = PAR(31)/12.
Z(3)    = W
XKPC    = PAR(32)
Z(14)   = XKPC
RCON    = PAR(33)
GAMV    = PAR(34)
CNVRGQ  = PAR(35)
CNVRGT  = PAR(36)
IMAX    = INT(PAR(37))
WCAP    = PAR(38)/12.
CB      = PAR(39)
Z(12)   = WCAP
C       RCON IS FOR HEAT PIPE FLUID
C       GAMV IS FOR HEAT PIPE FLUID
IF(NR.EQ.1) ISTAG = 2
C       SINGLE ROWS USE CORRELATIONS FOR STAGGERED ARRAYS
ROE     = DOE/24.
RIE     = DIE/24.
RIA     = DIA/24.
R2C     = DFIN + TFIN/2.
RA      = 1.414/(1. - DOE/R2C)
RB      = RA*(DOE/R2C)
RC      = 1 + R2C/DOE
Z(11)   = RIA
Z(8)    = RIE
PI      = 3.14159
GY      = 4.170E8
C       GY IS GRAVITY IN FT/HR2
XJ      = 788.
C       J IS FT LBF/ BTU
C
C----- CALCULATE PIPE CONSTANTS

```

C

```

AVE = PI * RIE ** 2.
AVEF = AVE * (1. - FF)
AVA = PI * RIA ** 2.
AVC = W * DEL
HMID = ((DFIN*PAR(7))+(DFIN-PL)*(PAR(7)-1.))/12.
HTOT = HB + HA + HMID
C(1) = ALOG(ROE/RIE) / (2.0 * PI * XLE * XKPE * M)
C(2) = 1./(XLE*M*2.*PI*RIE)

C(3) = 1./(CB * XLC * N)
C(4) = XLC * N * W
C(6) = XLC * N * WCAP
XXQ = CNVRGQ / 100.
XXT = CNVRGT / 100.

```

C

C----- BURNER XCHANGER GEOMETRY CONSTANTS

C

```

SF = 1./NF - TFIN
APO = PI*DOE*(1.-NF*TFIN)/12.
LF = (DFIN - DOE)/2.
IF(IFIN.EQ.2) GOTO 96
AFO = PI * NF*(2.*LF*(DOE+LF)+TFIN*(DOE+2.*LF))/12.
GOTO 97
96 AFO = PI*DOE*NF*(2.*LF*(WFIN+TFIN)+WFIN*TFIN)/(12.*WFIN)
97 CONTINUE
AO = APO + AFO
TSA = AO * XLE * FLOAT(M)
AC = (DOE+2.*LF*TFIN*NF)/12.
IF(NR.NE.1) GOTO 93
NT = M
GOTO 99
93 TEST= (M/2)*2. - M
IF(TEST.LT.0.0) GOTO 98
NT = (M/NR)
GOTO 99
98 NT = (M/NR) + 1
99 CONTINUE
AD = ((PT*FLOAT(NT))*XLE)/12.
AN = AD - AC*XLE*FLOAT(NT) - XC
B = LF + TFIN/2.
RAT = LF/SF
IF(ISTAG.EQ.1) GOTO 110

```

C

C

C

CORRELATIONS FOR STAGGERED AND SINGLE ROW ARRAYS

```

IF(IFIN.EQ.2) GOTO 94
ZC1 = .35
ZC2 = .65
ZC3 = -.25
ZC4 = .20
GOTO 95
94 ZC1 = .55
ZC2 = .45

```



```

      ZC3 = -.35
      ZC4 = .23
95  CONTINUE
      C3 = ZC1 + ZC2*EXP(ZC3*RAT)
      C5 = .7+((.7-.8*EXP(-.15*(FLOAT(NR)**2.)))*(EXP(-PL/PT))
      C4 = .11*((.05*PT/DOE)**(-.7*(RAT**ZC4)))
      C61 = EXP(-.15*FLOAT(NR)**2.)*2.1
      C62 = EXP(-2.*PL/PT)
      C63 = .8*EXP(-.15*FLOAT(NR)**2.)
      C64 = EXP(-.6*PL/PT)
      C6 = 1.1 + (1.8-C61)*C62 - (.7-C63)*C64
      GOTO 111

C
C  CORRELATIONS FOR INLINE ARRAYS
C
110 IF(IFIN.EQ.2) GOTO 89
      ZL1 = .2
      ZL2 = .65
      ZL3 = -.25
      ZL4 = .15
      GOTO 90
89  ZL1 = .35
      ZL2 = .50
      ZL3 = -.35
      ZL4 = .20
90  CONTINUE
      C3 = ZL1 + ZL2*EXP(ZL3*RAT)
      C5 = 1.1-((.75-1.5*EXP(-.7*FLOAT(NR)))*EXP(-2.*PL/PT)
      C4 = .08*((.15*PT/DOE)**(-1.1*(RAT**ZL4)))
      C6 = 1.6-((.75-1.5*EXP(-.7*FLOAT(NR)))*EXP(-.2*(PL/PT)**.2)

111 CONTINUE

C
C----- BURNER OPERATING CONDITIONS
C
      MFGAS = RATE/HHV
C      MFGAS = MASS FLOW RATE OF GAS [LBM/HR]
      VFGAS = MFGAS/.04144
C      VFGAS = VOLUMETRIC FLOW RATE OF GASS [FT3/HR]
      WRITE(IERR,5050) TSA,AN,MFGAS,VFGAS
5050 FORMAT(* CALCULATED CONSTANTS*/ * TSA      =*F7.3* [FT2]*
1,T30,* AFREE =*F7.4* [FT2]*/ * MFGAS =*F7.2* [LBM/HR]*,
1T30,* VFGAS =*F6.2* [FT3/HR]*)
101 MFPROD = (1+(17.137*(PTA/100.)))*MFGAS
C      MFPROD = MASS FLOW OF PRODUCTS [LBM/HR]
      XPT = 1. - 100./PTA
      IF(PTA.LE.100.) GOTO 29
      YPT = -1498.9*XPT - 282.67*XPT**2. - 222.5*XPT**3.
      GOTO 28
29  YPT = 351.63*XPT + 63.851*XPT**2.
28  TFRANK = (YPT*1.8) + 4189.
C      TFRANK = FLAME TEMPERATURE IN RANKINE
      TFLAME = TFRANK-459.7
C      TFLAME = FLAME TEMP IN DEG F

```

```

      G = MFPROD/AN
      IF(INFO.NE.-1) GOTO 100
C      G = COM PROD MASS VELOCITY [LBM/HR FT2]
C
C----- INITIAL GUESSES FOR PROPERTY EVALUATION
C
      TFLUE = 250.
      TV = TT + 120.
      TVE = TV
      TFLM = TT + 50.
      TFLMX= TFLM
      DTFM = 10.
      TCI = TT + 20.
      Q = 50000.
      XMDOT= 50.
C
C##### BEGIN ITERATION LOOP
C
      100 ICOUNT = ICOUNT + 1
      IF(ICOUNT.GT.IMAX) GOTO 104
C
C----- COMBUSTION PROD PROPERTIES
C
      TGPROP = (TFLUE+459.7+TFRANK)/2.
C      TGPROP = AVERAGE COMB PROD TEMPERATURE FOR PROP EVALUATION [DEG R]
      CP = CPPROD(PTA,TGPROP)
      TVAL = TGPROP
      IF (TGPROP.GT.3000) TVAL = 3000.
      TK = (TVAL-491.7) * .05556
      CMIN = MFPROD * CP
      VISC = 2.629E-3*((TGPROP**1.5)/(TGPROP+198.7))
C      TK = AVERAGE COM PROD TEMP FOR EVAL OF K [DEG C]
      K=.013967+TK*(3.8143E-4+TK*(-1.4287E-6+TK*(1.4098E-8+
1  TK*(-6.9972E-11))))
      RE = (G*DOE)/(VISC*12.)
      PR = (CP*VISC)/K
C
C----- HEAT TRANSFER COEF FOR BURNER
C
      C1 = .25*(RE**(-.35))
      TI = TVE + 459.7
      TS = TI + .3*(TGPROP - TI)
      JC = C1 * C3 * C5 * SQRT(DFIN/DOE) * (TGPROP/TS)**.25
      HBURN= JC * G * CP / (PR**.67)
      HBURN = 1./((1./HBURN)+RFO)
C
C----- FIN AND SURFACE EFFECTIVENESS
C
      IF(IFIN.EQ.2) GOTO 120
      ETA = (R2C-DOE)*SQRT((HBURN*12.)/(XKPE*TFIN))/12.
      ARGA = RA * ETA
      ARGB = RB * ETA
      U1 = BESI1(ARGA)*BESK1(ARGB)
      U2 = BESI1(ARGB)*BESK1(ARGA)

```

```

      B1 = BESI1(ARGA)*BESKO(ARGB)
      B2 = BESIO(ARGB)*BESK1(ARGA)
      FEFF = ((1.4142/ETA)/RC)*(U1-U2)/(B1+B2)
      GOTO 121
120  XM = SQRT(HBURN*(TFIN+WFIN)/(6.*XKPE*TFIN*WFIN))
      X = (TANH(XM*B))/(XM*B)
      FEFF = X * (.9 + .1*X)
121  SEFF = (FEFF*AFO + APO)/AO
      C(5) = 1./(HBURN * SEFF * TSA )

C
C----- DETERMINE HEAT PIPE FLUID PROPERTIES
C
C----- HEAT PIPE VAPOR
      RHOV = H2O(7,TV)
      HFGV = H2O(5,TV)
C----- CONDENSING FILM
      XMEWF = H2O(1,TFLMX)
      RHOF = H2O(2,TFLM)
      XKF = H2O(3,TFLM)
      CPF = H2O(4,TFLM)
      HFG = H2O(5,TFLM)

C
C----- CALCULATE HTC'S
C
      CALL HTC(HFILM,REFLML,DTLL,HFRET,HFREF,HBOIL,QBMAXR,QBMAXZ,QBMIN,
12  1Z,INFO,TV,TFLM,TFLMX,DTFM,TCI,TCIC,XMDOT,TX,TT)

C
C----- REYNOLDS # AND PRESSURE DROP
C
      CALL PDROP(Q,TV,Z,INFO,P,REV)
      INFO = 1
      PTOT = P(1) + P(2) + P(3)
      PTOT IS IN LBF/FT2

C
C----- CALCULATE Q
C
      QOLD = Q
      PDROPR = (TV+459.7) * PTOT/(RHOV*HFGV*XJ*Q)
      IF(WCAP.LE.0.) GOTO 127
      XX = SQRT(HFREF/(XKPC*TTW))*WCAP/2.
      FIN = TANH(XX)/XX
      GOTO 128
127  FIN = 1.0
128  FREEC = C(4)*HFRET + C(6)*HFREF*FIN
      UAPIPE = 1./(PDROPR + 1./(C(4)*HFILM)+C(3)+1./FREEC)
      HBOIL = 1./((1./HBOIL) + RFI)
      UAX = 1./(C(5)+C(1)+C(2)/(HBOIL*FF))
      NTUX = UAX/CMIN
      XEFFX = 1. - EXP(-NTUX)
      XEFF = 1./((1./XEFFX)+(CMIN/UAPIPE))
      Q = XEFF * CMIN * (TFLAME - TT)
      UA = 1./(1./UAX + 1./UAPIPE)
      TFAVE = Q/UA + TT
      XMDOT = Q / HFGV

```

```

C
C----- NEW TEMPERATURES
C
    TFLUE = TFLAME - Q/CMIN
    TEO = TFAVE - C(5)*Q
    TEI = TEO - C(1) * Q
    TVE = TEI - (C(2) * Q)/(HBOIL*FF)
    TVC = TVE - PDROPR * Q
    TCO = TVC - Q / (C(4)*HFILM)
    TCI = TCO - C(3) * Q
    TTZ = TCI - Q/FREEC

C
    TV = (TVC + TVE) / 2.
    TFLM = (TCO + TVC) / 2.
    TFLMX = TCO + .25 * (TVC - TCO)
    DTFM = (TVC - TCO)

C
C----- TANK TEMP CHECK
C
    CHKTT = ABS((TT - TTZ) / TT)
    IF(CHKTT.LE.XXT) GOTO 103
    WRITE(IERR,1000) CNVRGT,TT,TTZ
1000 FORMAT(1H ,*CALCULATION ERROR; TANK TEMPS GIVEN AND CALCULATED
1DO NOT MATCH WITHIN*,3X,F4.1,*PERCENT*,/,1H ,*THE VALUES ARE
2; GIVEN =*,3X,E10.3,3X,*DEG F AND CALCULATED =*,3X,E10.3,3X,
3*DEG F*)
    STOP

C
C----- ITERATE AGAIN?
C
    103 CHKQ = ABS((Q - QOLD) / Q)
    IF(CHKQ.GT.XXQ) GOTO 100
    QOLD = Q
    GOTO 105

C
C----- ERROR MESSAGES
C
    104 WRITE(IERR,1001) IMAX,Q,QOLD
    1001 FORMAT(1H ,*WARNING; HEAT PIPE LOOP HAS FAILED TO CONVERGE IN*,
    1I4,1X,*Q =*,E10.3,*QOLD =*,E10.3)

C
C----- BURNER PRESSURE DROP
C
    105 C2 = .07 + 8. * (RE**(-.45))
    EXPON = .5
    IF(ISTAG.EQ.1) EXPON = 1.
    FFAC = C2 * C4 * C6 * (DFIN/DOE) ** EXPON
    BETA2 = (AN/AD) ** 2.
    ACCEL = ((1. + BETA2)/(4.*FLOAT(NR)))*((TFLUE+459.7)
1 -TFRANK)/TGPROP
    PLOSS = ((FFAC+ACCEL) *G*G *FLOAT(NR))/(1.083E9*1.325*BP/TGPROP)

C
C----- COMBUSTION CHAMBER DRAFT (INCHES OF WATER)
C

```

```

DRAFT=.2554*BP*(HTOT/TAMBR-HB/TFRANK-HMID/TGPROP-HA/(TFLUE+459.7))
IF(PLOSS.LT.DRAFT) GOTO 115
WRITE(IERR,2222) DRAFT,PLOSS
2222 FORMAT(* WARNING; CORE PRESSURE DROP EXCEEDS DRAFT*/
1 * DRAFT =*E10.3*[IN. H2O]*/* PLOSS =*E10.3*[IN. H2O]*)
115 CONTINUE
C
C----- REGIME/LIMITATIONS CALCULATIONS
C
STC = H2O(9,TV)
XMEWV = H2O(8,TV)
G11 = Q/(RHOV*HFGV*(GAMV*XJ*GY*RCON*(TV+459.7)) ** .5)
XMACHE = G11 / (AVEF*M)
XMACHA = G11 / ((RIA ** 2.) * PI)
XMACHC = G11 / (W * DEL * N)
C
C XLIM(1) = SONIC LIMIT
C XLIM(4-6) = ENTRAINMENT ND # PI2 E,A,C
C CALC ASSUMES FLOW AREA = .85 C.S. AREA
C
XLIM(1) = M*AVEF*RHOV *HFGV* ((GAMV*RCON*XJ*GY*(TV+459.7))/
1 (2. * (GAMV + 1.))) ** .5
XLIM(4) = (XMDOT*XMEWV*(RHOV/RHOF)**.5)/
1 (STC*GY*RHOV*AVEF*M)
XLIM(5) = XLIM(4) * AVEF * M / AVA * .85
XLIM(6) = XLIM(5) * AVA / (AVC * N)
DO 10 I=1,3
IF(XLIM(1).GE.Q) GOTO 10
WRITE(IERR,9002) XLIM(1)
9002 FORMAT(1H ,* SONIC LIMIT EXCEEDED, LIMIT =*,E10.3*[BTU/HR]*)
10 CONTINUE
DO 11 I=1,3
IF(XLIM(I+3).LE.2.46E-4) GOTO 11
WRITE(IERR,9003) XLABEL(I),XLIM(I+3)
9003 FORMAT(1H ,A4,* ENTRAINMENT LIMIT EXCEEDED, PI2 = *,E10.3)
11 CONTINUE
C
C COND FILM TURBULENCE CHECK
C
IF(REFLML.LE.300.) GOTO 20
WRITE(IERR,2000) REFLML
2000 FORMAT(* WARNING; CONDENSOR FILM MAY BE TURBULENT*/
1 * FILM REYNOLDS # =*E10.3)
20 CONTINUE
C
C NUCLEATE BOILING RANGE CHECK
C
IF(Q.GE.QBMIN.AND.Q.LE.QBMAXZ) GOTO 21
WRITE(IERR,2001) Q,QBMIN,QBMAXZ
2001 FORMAT(* WARNING; Q IS NOT IN NUCLEATE BOILING RANGE*/
1 * Q =*E10.3/* QBMIN =*E10.3/* QBMAXZ =*E10.3)
21 CONTINUE
C
C BURNER HTC RANGE CHECK

```

```

C
    IF(RE.GE.1700) GOTO 22
    WRITE(IEERR,2002) RE
2002 FORMAT(* WARNING; HBURN CORELATION IS OUT OF RANGE*/
1 * RE # OF BURNER FLOW =*E10.3)
22 CONTINUE
C
C----- POINTS OF INTEREST
C
C    PRESSURE IN PIPE PSI
C
    PRESUR = H2O(11,TV)
C
C    FILM THICKNESS IN COND ASUM. NO SHEAR STRESS
C
    HFGP = HFG + .375 * CPF * DTFM
    VCON = (.8*W*N*((4.*XKF*XMEWF*(TVC-TCO)*XLC**5.)/(GY*RHOF*
1 (RHOF-RHOV)*HFGP))**.25)*1728.
    DTL = VCON / (XLC*.8*W*N*1728.)
C
C----- TOTAL VOLUME FLUID IN SYSTEM
C
    V EVP = (M * AVE * FF * XLE)*1728.
    VADB = .15 * AVA * XLA*1728.
    VSTEAM= (VEVP*(1.-FF)/FF + VADB*5.67 + (W*DEL*XLC*1728. -
1 VCON)) * H2O(7,TV)/H2O(2,TV)
    VTOT = V EVP + VADB + VCON + VSTEAM
C
C----- SUMMARY WRITES
C
    OEF = Q/RATE
    WRITE(IEERR,9004) PTA,RE,TT,HBURN,Q,VTOT,TFLAME,VEVP,
1TFLUE,XEFF,XLIM(4),XLIM(5),XLIM(6),XLIM(1),TV,PLOSS,PRESUR,DRAFT
2,OEF,MFPROD
9004 FORMAT(* PTA =*F5.0***,T30,* REBURN =*E10.4
1/* TT =*F7.1* [DEG F]*,T30,* HBURN =*F6.2* [BTU/HR FT2 DEG F]*
2/* Q =*F8.0* [BTU/HR]*,T30,* VTOT =*E10.4* [IN3]*
3/* TFLAME=*F7.1* [DEG F]*,T30,* V EVP =*E10.4* [IN3]*
4/* TFLUE =*F7.1* [DEG F]*,T30,* XEFF =*F5.3
4/* ETE =*E10.4,T30,* ETA =*E10.4
4/* ETC =*E10.4,T30,* QSONC =*E10.4*[BTU/HR]*
5/* TV =*F7.1* [DEG F]*,T30,* PLOSS =*F7.4* [IN H2O]*
6/* PRESUR=*F6.2* [PSI]*,T30,* DRAFT =*F7.4* [IN H2O]*
7/* OEF =*F6.3,T30,* MFPROD=*F6.3* [LBM/HR]*
7/* ++++++*)
C
C----- OUTPUTS
C
    OUT(1) = Q
    OUT(2) = XMDOT
    OUT(3) = PRESUR
    OUT(4) = TFLUE
    OUT(5) = TFLAME
    OUT(6) = TEO

```

```

OUT(7) = TEI
OUT(8) = TVE
OUT(9) = TVC
OUT(10) = TCO
OUT(11) = TCI
OUT(12) = TCIC
OUT(13) = PTOT/144.
OUT(14)= P(1)/144.
OUT(15)= P(2)/144.
OUT(16)= P(3)/144.
OUT(17)= REV(1)
OUT(18)= REV(2)
OUT(19)= REV(3)
OUT(20)= XMACHE
OUT(21)= XLIM(1)
OUT(22) = XLIM(4)
OUT(23) = XLIM(5)
OUT(24) = XLIM(6)
OUT(25)= QBMAXR
OUT(26)= QBMAXZ
OUT(27)= QBMIN
OUT(28)=HFILM
OUT(29)=DTFM
OUT(30) = DTLL
OUT(31) = DTL
OUT(32) = VCON
OUT(33) = REFLML
OUT(34) = HFRET
OUT(35) = HFREF
OUT(36) = FIN
OUT(37) = HBOIL
OUT(38) = VFGAS
OUT(39) = MFPROD
OUT(40) = RE
OUT(41) = HBURN
OUT(42) = TSA
OUT(43) = FEFF
OUT(44) = SEFF
OUT(45) = UAX
OUT(46) = NTUX
OUT(47) = UAPIPE
OUT(48) = XEFF
C FLAME TO EVAP WALL RES.
  OUT(49) = C(5)
C EVAP PIPE WALL RES.
  OUT(50)= C(1)
C BOILING RES.
  OUT(51)= C(2)/(HBOIL*FF)
C PRESSURE DROP RES.
  OUT(52)= PDROPR
C FILM RES.
  OUT(53)= 1./(C(4)*HFILM)
C CONDENSOR BOND RESISTANCE

```

```
      OUT(54)= C(3)
C  FREE CON. RES.
      OUT(55)= 1./FREEC
      OUT(56) = ICOUNT
      OUT(57)=FFAC
      OUT(58)=DRAFT
      OUT(59)=PLOSS
      OUT(60)=AN
      OUT(61) = V EVP
      OUT(62) = VADB
      OUT(63) = VSTEAM
      OUT(64) = VTOT
      OUT(65) = OEF
      RETURN
      END
```



```

SUBROUTINE HTC(HFILM,REFLML,DTLL,HFREE,HBOIL,QBMAXR,QBMAXZ,QBMIN,
1      PAR,INFO,TV,TFLM,TFLMX,TFRE,DTFM,DTFE,XMDOT,TX)
  DIMENSION PAR(11)
C*****
C      THIS SUBROUTINE IS DESIGNED TO CALCULATE FILM AND FREE
C      CONVECTION HEAT TRANSFER COEFFICIENTS FOR THE
C      HEAT PIPE MODEL      (11/78) N.D.
C*****
      IF(INFO.NE.-1) GOTO 101
      XLC  = PAR(1)
      N    = PAR(2)
      W    = PAR(3)
      DEL  = PAR(4)
      CSF  = PAR(5)
C      CSF IS SURFACE/FLUID CONSTANT FOR HBOIL CORRELATION
      M    = INT(PAR(6))
      XLE  = PAR(7)
      RIE  = PAR(8)
      FF   = PAR(9)
      PI   = 3.14159
      GY   = 4.170E8
C      GY IS GRAVITY IN FT/HR 2
      ASP  = DEL/W
      AS   = (1. - ASP) ** 2.
      BUG1 = (14.2 + AS * (4.47 + AS * 5.33))
      DTLL = 3.0E-4

C
C----- HEAT PIPE VAPOR
101 XMEWV = H2O(8,TV)
      RHOV = H2O(7,TV)
      HFG  = H2O(5,TV)
C----- HEAT PIPE LIQUID
      XMEWF = H2O(1,TFLMX)
      RHOF  = H2O(2,TFLM)
      XKF   = H2O(3,TFLM)
      CPF   = H2O(4,TFLM)
C----- TANK FLUID
      XMEWT = H2O(1,TFRE)
      RHOT  = H2O(2,TFRE)
      XKT   = H2O(3,TFRE)
      CPT   = H2O(4,TFRE)
      BETA  = H2O(6,TFRE)
C----- BOILING POOL PROPERTIES
      XMEWB = H2O(1,TV)
      RHOB  = H2O(2,TV)
      XKB   = H2O(3,TV)
      CPB   = H2O(4,TV)
      STC   = H2O(9,TV)

C
      ZIP  = GY * RHOF * (RHOF - RHOV) / XMEWF
C

```

C***** FILM HEAT TRANSFER COEFFICIENT

C

GVM = (0.4 * XMDOT) / (W * DEL * N)
 REVM= (2. * GVM * W * DEL) / (XMEWV * (W + DEL))
 IF(REVM.LT.2000) GOTO 1
 F = 0.0791 / (REVM ** .25)
 GOTO 2

1 F = BUG1 / REVM

2 CONTINUE

TAUV = (F * GVM ** 2.) / (2. * RHOV)

C

C----- FILM THICKNESS AT BOTTOM

C

HFGP = HFG + (.375 * CPF * DTFM)
 TOP = (DTFM * XKF * XLC) / HFGP
 ZAP = (TAUV * RHOF) / XMEWF

3 DTLLZ= DTLL

B1 = (ZIP * DTLLZ / 4.) - (ZAP/3.)

IF(B1.GE.0.) GOTO 7

WRITE(2,1001) ZIP,ZAP, TOP,B1

1001 FORMAT(* ERROR IN HTC- CALC OF FILM COND HTC HAS BECOME

1 MEANINGLESS*/ * DUMP OF ZIP ZAP TOP AND B1 FOLLOWS*/4E10.3)

7 CONTINUE

DTLL = (TOP / B1) ** .333

CHK = ABS((DTLL - DTLLZ)/ DTLL)

IF(CHK.GT..01) GOTO 3

C

C----- MASS FLOW PER UNIT WIDTH AT BOTTOM

C

FLOW = (ZIP*(DTLL**3.)/3.)-(ZAP*(DTLL**2.)/2.)

C

C----- AVERAGE FILM HEAT TRANSFER COEF.

C

HFILM = (HFGP * FLOW) / (XLC * DTFM)

REFLML = 4. * FLOW / XMEWF

C

C***** FREE CONVECTION COEFFICIENT

C

PR = (CPT * XMEWT) / XKT

GR = (GY*BETA*DTFE*(XLC**3.)*(RHOT**2.))/(XMEWT**2.)

IF(GR.GT.1.E9) GOTO 4

DUM = -.25

HFREE= (.6773/XLC)*XKT*(PR**.5)*((0.952*PR)**DUM)*(GR**.25)

GOTO 5

4 EXP = -0.4

HFREE = (XKT/XLC) * 0.0246 * ((1. + .494 * (PR **.666)) **EXP) *

1 (PR **.467) * (GR ** 0.4)

5 CONTINUE

C

C***** NUCLEATE POOL BOILING

C

Q = XMDOT * HFG

A = 2. * PI * RIE * XLE * FF * M

D1=SQRT(STC/(RHOB-RHOV))

```

      D2=(Q/A)/(XMEWB*HFG)
      D3=CSF*HFG/CPB
      PR=CPB*XMEWB/XKB
C
      TX=((D1*D2)**.33)*D3*(PR**1.7)
C
      HBOIL = (Q/A)/TX
C
C----- MIN AND MAX Q/A
C
C      QMAXR IS MAX FROM ROHSENHOW
C      QMAXZ IS MAX FROM ZUBER
      QBMAXR = A*RHOV*HFG*143.*((RHOB-RHOV)/RHOV)**.6
      E1 = SQRT(RHOB/(RHOB+RHOV))
      E2 = ((STC*(RHOB-RHOV)*GY*GY)/(RHOV**2.))**.25
      QBMAXZ = RHOV*HFG*A*.13*E1*E2
      QBMIN = .2 * QBMAXZ
      RETURN
      END

```

```

      SUBROUTINE PDROP(Q,TV,PAR,INFO,P,REV)
      DIMENSION PAR(11),P(3),REV(3),F(6),G(4),R(3)
C*****
C      THIS SUBROUTINE IS DESIGNED TO CALCULATE PRESSURE
C      DROP FOR THE TYPE29 HEAT PIPE MODEL      (11/78)
C*****
      IF(INFO.NE.-1) GOTO 101
      XLE = PAR(7)
      XLA = PAR(10)
      XLC = PAR(1)
      N = PAR(2)
      M = PAR(6)
      RVE = PAR(8)
      RIA = PAR(11)
      W = PAR(3)
      DEL = PAR(4)
      PI = 3.14159
      GY = 4.170E8
C      GY IS GRAVITY IN FT/HR2
C
C----- CONSTANTS
C
      ASP = DEL / W
      AS = (1. - ASP) ** 2.
      F(1) = 8. / (PI * RVE ** 4.)
      F(2) = 8. / (PI * RIA ** 4.)
      BUG1 = (14.2 + AS * (4.47 + AS * 5.33))
      F(3) = (BUG1 * (W + DEL) ** 2.) / (2. * (W*DEL)**3.)
      F(4) = .00898 / (RVE ** 4.75)
      F(5) = .00898 / (RIA ** 4.75)
      F(6) = (.0666 * (W + DEL) ** 1.25) / (W * DEL) ** 3.
      D11 = 1. / ((PI * RVE ** 2.) ** 2.)
      D13 = 1. / (W * DEL) ** 2.
      R(1) = 2. / (PI * RVE * M)
      R(2) = 2. / (PI * RIA)
      R(3) = 2. / (N * (W + DEL))
C
C----- PROPERTIES
C
      101 XMEWV = H2O(8,TV)
          RHOV = H2O(7,TV)
          HFG = H2O(5,TV)
C
C----- REYNOLDS #'S
C
      G(1) = XMEWV / (RHOV * HFG)
      G(2) = Q / (XMEWV * HFG)
      G(3) = (XMEWV ** .25) / (RHOV * (HFG ** 1.75))
      G(4) = 1. / (RHOV * HFG ** 2.)
      DO 1 I=1,3
      1 REV(I) = G(2) * R(I)
C
C----- PRESSURE DROP (LAMINAR AND TURBULENT)

```

C

```
IF(REV(1).GT.2300.) GOTO 5
FVE = F(1) * G(1)
P(1) = (Q * FVE * XLE) / (3. * M)
GOTO 6
5 FVEP = G(3) * F(4)
DVE = G(4) * D11
P(1) = (.267 * FVEP * ((Q/M) ** 1.75) * XLE + .667 * DVE * ((Q/M)
1 ** 2.))
6 IF(REV(2).GT.2300.) GOTO 7
FVA = F(2) * G(1)
P(2) = (Q * FVA * XLA)
GOTO 8
7 FVAP = G(3) * F(5)
P(2) = (FVAP * (Q ** 1.75) * XLA)
8 IF(REV(3).GT.2300.) GOTO 9
FVC = F(3) * G(1)
P(3) = (Q * FVC * XLC) / (3. * N)
GOTO 10
9 FVCP = G(3) * F(6)
DVC = G(4) * D13
P(3) = (.267 * FVCP * ((Q/N) ** 1.75) * XLC - .667 * DVC *
1 ((Q/N) ** 2.))
10 CONTINUE
DO 11 I=1,3
P(I) = P(I) / GY
11 CONTINUE
RETURN
END
```

```

FUNCTION H2O(IP,TV)
  DIMENSION C(10,11),RABEL(11)
  DATA RABEL/4HVISW,4HDENW,4HTHMW,4HSHTW,4HLTHT,
*4HBETA,4HDENS,4HVISS,4HSTCW,4HDDIF,4HPSAT/
C*****
C  THIS FUNCTION EVALUATES THE PROPERTIES OF WATER AND STEAM
C  GIVEN THE TEMPERATURE. 'IP' DESIGNATES THE PROPERTY;
C      1 = VISCOSITY [LBM/HR FT],(32 TO 705.4 DEG F)
C      2 = DENSITY [LBM/FT3],(50 TO 705.4 DEG F)
C      3 = THERMAL COND. [BTU/HR FT DEG F],(32 TO 705.4 DEG F)
C      4 = SPECIFIC HEAT [BTU/LBM DEG F],(32 TO 680 DEG F)
C      5 = LATENT HEAT [BTU/LBM],(32 TO 705.4 DEG F)
C      6 = BETA [1/DEG F],(46 TO 212 DEG F) ATM. PRESRE
C      7 = STEAM DENSITY [LBM/FT 3],(50 TO 705.4 DEG F)
C      8 = STEAM VISCOSITY [LBM/FT HR],(0 TO 750 DEG F)
C      9 = SURFACE TENSION COEFFICIENT [LBF/FT],(32 TO 680 DEG F)
C     10 = DENSITY DIFERENCE FROM 32 DEG F [LBM/FT3] (32 TO 212 DEG
C     11 = SATURATION PRESSURE [PSI], (50 TO 705.4 DEG F)
C  INPUT TEMPERATURE IS IN DEG F, PRESSURE IS SAT UNLESS GIVEN
C  EQUATIONS USE VARIUOS TRANSFORMATIONS OF DEG C
C*****
C  DATA ARE POSITIONED AS FOLLOWS;
C      C(1-6,IP) = EQUATION COEFFICIENTS
C      C(7,IP)   = TEMPERATURE TRANSFORMATION ADDITIVE FACTOR
C      C(8,IP)   = TEMPERATURE TRANSFORMATION MULTIPLICATIVE FACTOR
C      C(9,IP)   = CONVERSION FACTOR TO LB/FT/HR SYSTEM
C      C(10,IP)  = UPPER LIMIT OF FIT
C*****
C      DATA C/.69315,-.36389,.025686,-.0012168,3.1076E-5,
* -3.2126E-7,-32.,.05556,2.419,705.4,
* 24.758,-.68745,14.804,-7.9058,0.,
* 0.,-705.4,-.05556,1.,705.4,
* .32417,.016393,-.0011969,3.8806E-5,-5.8900E-7,
* 0.,-32.,.05556,1.,705.4,
* .0047628,.66376,6.1985,-9.5208,9.3237,
* 0.,-705.4,-.05556,1.,680.,
* -586.60302,5390.4459,-9789.4194,5255.0767,0.,
* 0.,-705.4,-.05556,1.,705.4,
* -4.90694,1.47412,-.0115287,4.82008E-5,0.,
* 0.,-32.,.5556,.5556E-5,212.,
* -8.1528,.71923,-.033825,.0013395,-3.2505E-5,
* 3.3834E-7,-32.,.05556,1.,705.4,
* 35.1511,2.2171,-.074653,.0018579,0.,
* 0.,-705.4,-.05556,2.19,750.,
* 51.694,-9.3907E-2,-2.3677E-4,2.9513E-7,0.,
* 0.,-32.,.5556,1.E-4,680.,
* 0.,-.500429E-5,-.546020E-5,.132166E-7,0.,
* 0.,-32.,.5556,62.428,212.,
* -2.4147,.71598,-.026808,7.3911E-4,-1.2275E-5,
* 8.9646E-8,-32.,.05556,1.,705.4/
C
C  TEMPERATURE IN RANGE OF FIT?
C

```

```

      T=TV
      IF(T.LE.C(10,IP)) GOTO 100
      T = C(10,IP)
      WRITE(2,2000) RABEL(IP),TV,T
2000  FORMAT(* WARNING FROM H2O, TEMP OUTSIDE RANGE FOR *,A4/
      1* T =*,E10.3/* T SET TO*,E10.3)

C
C      TEMPERATURE TRANSFORMATION
C
100  T = (T + C(7,IP)) * C(8,IP)

C
C      CURVE FITS
C
      GOTO(1,2,1,4,5,1,1,1,1,1,1),IP
1    SUM = C(6,IP) * T
      DO 11 N=1,4
      I = 6 - N
      SUM = (SUM + C(I,IP)) * T
11   CONTINUE
      SUM = SUM + C(1,IP)
      GOTO(90,1,91,1,1,91,90,92,91,91,90),IP
2    H2O=(C(1,IP)+C(2,IP)*T+C(3,IP)*SQRT(T)+C(4,IP)*(T**.333))*C(9,IP)
      RETURN
4    SUM = 0.0
      DO 44 N=1,5
      E = 2. - FLOAT(N)
      SUM = SUM + C(N,IP)*(T**E)
44   CONTINUE
      GOTO 91
5    SUM = 0.0
      DO 55 N=1,4
      E = 1./FLOAT(N+1)
      SUM = SUM + C(N,IP)*(T**E)
55   CONTINUE
      GOTO 91
90   H2O = EXP(SUM) * C(9,IP)
      RETURN
91   H2O = SUM * C(9,IP)
      RETURN
92   H2O = (1./SUM) * C(9,IP)
      RETURN
      END

```

```

      FUNCTION CPPROD(PTA,T)
C      PTA = PERCENT THEORETICAL AIR
C      T   = TEMPERATURE IN DEG R
      REAL MCO2,MH2O,MO2,MN2,MCH4
C
C----- MOLES OF CONSTITUENTS IN PRODUCT
C
      RAT = PTA/100.
      IF(PTA.LT.100.) GOTO 1
      MCO2 = 1.
      MH2O = 2.
      MO2  = 2. * (RAT - 1.)
      MN2  = 7.528 * (RAT)
      MCH4 = 0.0
      CPO2 = 11.515 - 172./SQRT(T) + 1530./T
      CPCH4 = 0.0
      GOTO 2
1      MCO2 = RAT
      MH2O = 2. * RAT
      MN2  = 7.528 * RAT
      MO2  = 0.0
      MCH4 = 1. - RAT
      CPCH4 = 3.385 + T*(1.003E-2 - 1.328E-6 * T)
      CPO2 = 0.0
C
C----- TOTAL MOLES OF PRODUCT PER MOLE METHANE BURNED
C
2      TMP = 1. + 9.528*RAT
      WT   = (16.043 + 274.9*RAT)/TMP
C
C----- SPECIFIC HEAT OF CONSTITUENTS
C
      CPCO2 = 16.2 - 6.53E3/T + (1.41E6/(T**2.))
      CPH2O = 19.86 - 597./SQRT(T) + 7500./T
      CPN2  = 9.47 - 3.47E3/T + 1.16E6/(T**2.)
C
C----- SPECIFIC HEAT OF MIXTURE [BTU/LBM DEG R]
C
      CPPROD = (MCO2*CPCO2 + MH2O*CPH2O + MO2*CPO2 + MN2*CPN2 + MCH4*
1      CPCH4) / (16.043 + 274.9*RAT)
      WRITE(1,6000) MCO2,MH2O,MO2,MN2,MCH4
6000 FORMAT(' MCO2,MH2O,MO2,MN2,MCH4'/5E10.3)
      WRITE(1,6001) CPCO2,CPH2O,CPO2,CPN2,CPCH4,CPPROD,TMP,WT
6001 FORMAT(' CPCO2,CPH2O,CPO2,CPN2,CPCH4,CPPROD,TMP,WT'/8E10.3)
      RETURN
      END

```


APPENDIX D

Derivation of the Traced Tank Computer Model

As discussed in Section 2.2.1.3, the traced tank behaves approximately like a cross-flow heat exchanger with fins on both sides. The tank fluid is considered to be mixed whereas the collector fluid is unmixed. The average temperature of the inside surface of the traced coil, $T_{cs}(x)$, at a given distance, x , above the base of the tank, the corresponding average temperature of the tank's inside surface, $T_{vs}(x)$, and the corresponding convection heat transfer coefficients, h_c and h_v , must be solved for simultaneously. The rate of heat transfer per unit tank wall area, $\dot{q}_v$, the fin efficiencies, η_c and η_v , and the fin base temperatures, T_{cfb} and T_{vfb} , are related to the unknowns enumerated above and are used in the basic equations for the sake of clarity.

Referring to Figures 2.11 and 2.12 in Section 2.2, the heat transferred from the collector fluid to the tank fluid per unit tank wall area is:

$$\frac{Q_v}{A_{vt}} = \dot{q}_v = h_v [T_{vfb} - T_b] \eta_v \quad (D.1)$$

where

$$\eta_v = \frac{\tanh (mL)_v}{(mL)_v} = \text{tank fin efficiency}$$

$$m_v = \sqrt{\frac{h_v}{K_v t_v}}$$

T_b = temperature at bottom of tank

A_{vt} = total wetted surface area of tank fins

Q_v = total heat transfer across traced coil

K_v = thermal conductivity of tank wall

Also

$$\dot{q}_v = \frac{(UA)_B}{A_{vt}} (T_{cfb} - T_{vfb}) \quad (D.2)$$

and

$$\dot{q}_v = h_c \frac{A_c}{A_{vt}} [T_c - T_{cfb}] \eta_c \quad (D.3)$$

where

$(UA)_B$ = overall conductance of coil bond

T_c = collector fluid temperature

$\frac{A_c}{A_v} = \frac{\pi d_c}{L_v} =$ ratio of coil area to tank fin area

$\eta_c = \frac{\tanh (mL)_c}{(mL)_c} =$ coil fin efficiency

$$m_c = \sqrt{\frac{h_c}{K_c t_c}}$$

$L_c = \pi (d_c + t_c) / 2 =$ coil fin length

$T_{vs} = T_b + \eta_v (T_{vfb} - T_b) =$ average tank fin temperature (D.4)

$T_{cs} = T_c - \eta_c (T_c - T_{cfb}) =$ average coil fin temperature (D.5)

To solve for the total heat transferred by the traced tank wall coil, the vertical extent of the wall is divided into several finite elements or nodes. Each element has a wall area A_{vt}/N where N is the total number of nodes. Although the collector fluid temperature varies continuously in the vertical direction, the temperature change over each element is sufficiently small that average tank-side and coil-side film temperatures may be used to evaluate local fluid properties at each element.

To do this an expression for the average collector fluid temperature in terms of known temperatures (the collector fluid temperature as it enters the i^{th} node, $T_{c,i}$, and T_b) must be substituted for T_c in equation D.3:

$$T_c = T_b + (T_{c,i} - T_b) \frac{G_c N}{U_o A_{vt}} \left[1 - \exp\left(\frac{-U_o A_{vt}}{G_c N}\right) \right] \quad (D.6)$$

where

$$G_c = \dot{m}_c C_{pc} = \text{thermal capacitance of the collector fluid}$$

and

$$U_o = \left[\frac{1}{h_v \eta_v} + \frac{A_{vt}}{(UA)_B} + \frac{A_{vt}}{A_c h_c \eta_c} \right]^{-1} \quad (D.7)$$

With this substitution equations (D.1, 2, 3) may be solved at each node in succession by noting that:

$$T_{c,i+1} = T_{c,i} - (T_c - T_b) \frac{U_o A_{vt}}{G_c N} \quad (D.8)$$

and upon substituting Equation D.6:

$$T_{c,i+1} = T_b + (T_{c,i} - T_b) \exp\left[\frac{-U_o A_{vt}}{G_c N}\right] \quad (D.9)$$

where

$$T_{c,i+1} = \text{the temperature of the collector fluid exiting the } i^{\text{th}} \text{ node}$$

The only remaining difficulty is that the temperature of the tank wall at two particular points must be known to compute local values of h_v . (The evaluation of h_v will be discussed presently.) The temperatures at the midpoint of the tank wall $T_w(L/2)$, and at the midpoint of the laminar boundary layer region $T_w(x_t/2)$ must therefore be estimated to get the heat transfer rates at each element. This means that the problem really requires the equations for all the nodes to be solved simultaneously. In practice, however, a solution can be obtained by making an initial guess at $T_w(L/2)$ and $T_w(x_t/2)$, solving the node equations thus obtaining better estimates of $T_w(L/2)$ and $T_w(x_t/2)$, and repeating the process until the change in overall heat transfer rate per iteration is less than the error tolerance. The node equations may be viewed as an inner problem and the procedure to iteratively solve

for $T_w(L/2)$ and $T_w(x_t/2)$ as an outer problem. This structure makes numerical solution of the problem very efficient and straightforward. The outer problem converges quickly because U_o is not a very strong function of the wall temperatures. Moreover, the number of iterations to convergence is independent of the number of nodes; consequently there is not an excessive penalty for increasing the number of nodes into which the wall is divided.

The foregoing development can be extended to a baffled traced tank. A baffle may be employed as a means of enhancing thermal stratification. In terms of the heat transfer analysis, the baffle makes the traced tank heat exchanger behave more like a counterflow heat exchanger. The node equations just presented are retained for this type heat exchanger, but the outer problem becomes more complex, requiring one variable per node (e.g. the node effectiveness, ϵ_i) plus one additional variable, G_t , the thermal capacitance rate on the tank-side. G_t is induced by a chimney effect and is quite variable as a result. Evaluation of h_v is also different for a baffled tank since combined free and forced convection are involved.

The equations used to evaluate h_c and U_B , and to evaluate h_v for the unbaffled tank, are presented in the following subsections.

D.1 Traced Coil Convective Heat Transfer Coefficient

The Nusselt number, $Nu_d = hd/K$, for forced convection heat transfer inside a tube depends on tube geometry and on the Reynolds and Prandtl numbers. It is convenient to represent the function by splicing three analytical expressions which correspond to the laminar, transition and turbulent flow regimes. The recommended expressions (Krieth [29] and Holman [30]) for straight tubes are:

$$h = \frac{K}{d_c} \left[3.66 + \frac{.0668 (2d_c L_v / A_{vt}) Re Pr}{1 + .04 [(2d_c L_v / A_{vt}) Re Pr]^{2/3}} \right] \quad (\text{laminar}) \quad (D.10)$$

$$h = .116 \frac{K}{d_c} (Re^{2/3} - 125) Pr^{1/3} [1 + (2d_c L_v / A_{vt})^{2/3}] \quad (\text{transition}) \quad (D.11)$$

$$h = .023 \frac{K}{d_c} Re^{0.8} Pr^{0.4} \quad (\text{turbulent}) \quad (D.12)$$

and all fluid properties are evaluated at the film temperature:

$$T_{fc} = (T_c + T_{cs})/2.$$

For $Re < 5000$, h is the greater of Equations D.10 and D.11; for $5000 < Re < 20000$, h is the lesser of D.11 and D.12; for $Re > 20000$ h is given by D.12.

According to McAdams [4], for a helically coiled tube the convection heat transfer coefficient, h , is enhanced by a factor of $1 + 3.5 d_c/D_c$ thus:

$$h_c = [1 + 3.5 d_c/D_c]h \quad (D.13)$$

where d_c is the inside diameter of the traced tube and D_c is the helix diameter.

D.2 Bond Conductance

Figure 2.12 depicts the trace-coil to tank-wall bond. The bond conductance per unit length of coil has been shown by de Winter [31] to be well approximated by:

$$C_s = \frac{4 K_v t_v}{d_c + 2t_c} \quad (D.14)$$

This approximation is valid when $K_w t_w \ll K_c t_c$ which is the case when a copper tube is soldered to a thin walled steel tank. The term, $(UA)_B/A_{vt}$, appearing in equation (D.2) is thus defined by:

$$\frac{(UA)_B}{A_{vt}} = \frac{C_s}{2L_v} = \frac{2}{L_v} \frac{K_v t_v}{d_c + 2t_c} \quad (D.15)$$

D.3 Tank Wall Convective Heat Transfer Coefficient

The coefficients of heat transfer from the inner tank wall have been adapted from semi-empirical equations representing free convection along vertical free standing surfaces in fluids of essentially infinite extent.

Two convective flow regimes are encountered along a heated vertical surface. A laminar boundary layer first develops along the lower portion. Then, if either the temperature difference or the vertical extent of the surface is sufficient, the transition to a turbulent boundary layer occurs and turbulent free convection takes over along the upper portion of the surface. The transition takes place at a distance, x_t , from the lower extreme of the heated surface which, according to McAdams [4] is determined by:

$$Ra_{x_t} = 10^9 \quad (D.16)$$

where the dimensionless Rayleigh number is defined as

$$Ra_x = Gr_x Pr_x = \frac{\rho g \beta \Delta T x^3}{\mu \alpha} \quad (D.17)$$

ΔT = wall temperature minus bulk fluid temperature

α = thermal diffusivity of fluid

ρ = fluid density

g = local acceleration due to gravity

β = thermal coefficient of expansion

μ = fluid viscosity

x = characteristic length

The transition point may be obtained explicitly by rearranging equation D.17.:

$$x_t = \left[\frac{10^9 \mu \alpha}{\rho g \beta \Delta T} \right]^{1/3} \quad (D.18)$$

The heat transfer coefficient is uniform over the turbulent boundary layer region and independent of the vertical extent of the turbulent region (Kreith [29]). McAdams [4] recommends the following evaluation:

$$h_v = 0.129 \frac{K}{L} Ra_L^{1/3} \quad (D.19)$$

This expression is also generally used for the average heat transfer coefficient over the entire vertical extent of a free standing surface on which both turbulent and laminar convection are active. At heat fluxes large enough to cause turbulent free convection over most of the surface, equation D.19 may also be used for interior vertical surfaces of fluid containers such as the condenser wall of the upper tank.

However, the local heat transfer coefficient is actually not nearly as great near the lower extreme of a vertical, interior container wall as it is near the leading edge of a vertical free standing surface. This becomes important to the traced tank analysis because most or all of the wall is in the laminar convection regime during operation at low heat fluxes.

The local laminar convection heat transfer coefficient is a function of distance, x , from the leading edge. Figure D.1 is a plot of this function for the case of a vertical free standing wall. Superimposed on this figure is the function used in the traced tank model to estimate the heat transfer coefficient along the laminar boundary layer region. The value of this coefficient is given by:

$$h_v = K \left[\frac{\rho g \beta \Delta T}{\mu \alpha} \right]^{1/3} (.092 - .023 x/x_t) \quad (D.20)$$

This and equations D.18 and D.19 are used to evaluate the local heat transfer coefficients, $h_{v,i}$, for each element of the traced tank heat exchanger. The term $\rho g \beta \Delta T$ is replaced by a mean density difference, $\Delta \rho g$, defined in terms of the density temperature function for water, $\rho(T)$, by:

$$\Delta \rho g = g[\rho(T_b) - \rho(T_w(x_t))] \quad (D.21)$$

in the case of the laminar regime, and:

$$\Delta \rho g = g \left[\frac{1}{L} \int_0^L \rho(T_{c,x}) dx - \rho(T_{w,L/2}) \right] \quad (D.22)$$

in the case of the turbulent regime. $T_c(x)$ is the temperature along the center line of the tank.

FIGURE D.1 LOCAL HEAT TRANSFER COEFFICIENTS OF LAMINAR FREE CONVECTION

Fluid properties appearing in the Grashoff numbers are evaluated using film temperatures based on the wall temperatures at the midpoints of the boundary layer regions:

$$T_{f,x_t/2} = (T_{w,x_t/2} + T_b) / 2 \quad (D.23)$$

$$T_{f,L/2} = (T_{f,L/2} + T_b) / 2 \quad (D.24)$$

Fluid properties appearing in the Prandtl numbers are evaluated at local (i.e. node average) film temperatures:

$$T_{f,x_i} = (T_{w,x_i} + T_b) / 2 \quad (D.25)$$

where x_i is the distance of i^{th} node's midpoint from the lower extreme of the traced tank surface.

APPENDIX E

This appendix contains a complete listing of the routines written for inclusion in the modified TRNSYS system simulation model developed under Contract DE-AC02-78CS43696-A002 and cited in this report.

```

      SUBROUTINE TYPE31(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C*****
C      TOP TANK AND HEATER ROUTINE COMBINED
C      GINY TANK TEMP AND PERCENT THEORETICAL
C      AIR. ASSUMES FULLY MIXED TANK
C      (3/12/79) ND
C      PAR(1) = TANK HEIGHT
C      PAR(2) = TANK DIAMETER
C      PAR(3) = CP OF TANK FLUID
C      PAR(4) = DENSITY
C      PAR(5) = OVERALL HTC TO ENV
C      PAR(6) = TINT
C      PAR(7) = LUN FOR MAP
C      PAR(8) = % THEO. AIR
C      PAR(9) = TSET
C      PAR(10) = NDIV
C*****
      DIMENSION QM(2,2),QX(10,10),XIN(4),OUT(9),INFO(9),PAR(10)
1      ,T(1),DTDT(1)
      COMMON /SIM/DUM1,DUM2,DELT
      REAL MSCP
      DATA PI/3.1315927/

C
C----- INPUTS
C
      TIN = XIN(1)
      FIN = XIN(2)
      QDIODE = XIN(3)
      TENV = XIN(4)
C      QDIODE IS POSITIVE UP
      IF(INFO(7).NE.-1) GOTO 1

C
C----- PARAMETERS/CONSTANTS AND INITIALIZATION
C
      TESAV = 0.0
      TQSAV = 0.0
      TFSAV = 0.0
      TOTENV = 0.0
      TOTQPE = 0.0
      TOTFLW = 0.0
      DELE = 0.0
      GAM = 0.0
      TT = PAR(6)
      TOLD = PAR(6)
      INFO(6) = 7
      IERR = 6
      IU = PAR(7)
      PTA = PAR(8)
      TSET = PAR(9)
      NDIV = PAR(10)
      AT = (PI*PAR(2)**2.)/4.
      AS = PI *PAR(1)*PAR(2)
      A = AS + 2. * AT

```

```

      UA = A*PAR(5)
      VOL = AT * PAR(1)
      MSCP = VOL * PAR(3) * PAR(4)
      DT = DELT/FLOAT(NDIV)
C
C----- READ MAP
C
      READ(IU,1000) TFL,DELTF,M,TTL,DELTT,N
1000 FORMAT(1X,F6.1,F6.1,2X,I2,F6.1,F6.1,2X,I2)
      DO 6 I=1,M
      READ(IU,1001) (QX(I,J),J=1,N)
1001 FORMAT(1X,10F8.0)
      6 CONTINUE
      TTH = DELTT * (N-1) + TTL
      TFH = DELTF * (M-1) + TFL
      WRITE(6,1100) TFL,TFH,DELTF,M,TTL,TTH,DELTT,N
1100 FORMAT(1H ,*+++++*/* HEAT PIPE INTERP SUMMARY */
      1 * PTAL,PTAH,DELPTA,M*/1H ,3F7.1,2X,I2/* TTL,TTH,DELTT,N*/
      2 1H ,3F7.1,2X,I2)
      DO 7 I=1,M
      WRITE(6,234) (QX(I,J),J=1,N)
234 FORMAT(1H ,10E10.3)
      7 CONTINUE
      WRITE(6,233)
233 FORMAT(1H ,*+++++*)
      IF(PTA.GE.TFL.AND.PTA.LE.TFH) GOTO 11
      WRITE(IERR,6060) PTA
6060 FORMAT(1H ,*FATAL ERROR IN HPIPE INTERP ROUTINE*/
      1*PTA OUT OF MAP RANGE*/* PTA =*F6.1)
      STOP
      11 CONTINUE
C
C----- SAVE ENERGY QUANTITIES FOR INTERNAL INTEGRATION
C
      1 IF(TIME.LE.0.0) GOTO 501
      IF(INFO(7).NE.0.0) GOTO 498
      TESAV = TOTENV
      TQSAV = TOTQPE
      TFLSAV = TOTFLW
      TOLD = TT
      GOTO 499
498 CONTINUE
      TOTENV = TESAV
      TOTQPE = TQSAV
      TOTFLW = TFLSAV
      TT = TOLD
499 CONTINUE
C
C***** BEGIN INTERNAL TIME LOOP
C
      DO 500 K=1,NDIV
C
C----- BURNER ON OR OFF? - TEMP RANGE OF MAP?
C

```

```

        IF(TT.GE.TSET) GOTO 5
        IF(TT.GE.TTL) GOTO 3
        WRITE(IERR,6000) TT
6000  FORMAT(1H ,*FATAL ERROR IN HPIPE INTERP ROUTINE*/ * TANK
      1  TEMP OUT OF MAP RANGE*/ * TT =*,F6.1)
      3  IF(TT.LE.TTH) GOTO 4
        WRITE(IERR,6000) TT
      4  CONTINUE
C
C----- LOCATE ABSOLUTE NODE NUMBER
C
      TTZ = (TT - TTL) / DELTT
      ITT = INT(TTZ) + 1
      TFZ = (PTA - TFL) / DELTF
      ITF = INT(TFZ) + 1
C
C----- SET UP 2 BY 2 RELATIVE Q MATRIX 'QM'
C
      DO 2 I=1,2
      DO 2 J=1,2
      II = ITF + I - 1
      JJ = ITT + J - 1
      QM(I,J) = QX(II,JJ)
      2  CONTINUE
C
C----- INTERPOLATE
C
      XT = TTZ - ITT + 1
      XF = TFZ - ITF + 1
      QA = XT *(QM(1,2) - QM(1,1)) + QM(1,1)
      QB = XT *(QM(2,2) - QM(2,1)) + QM(2,1)
      QHPIPE = XF * (QB - QA) + QA
      GAM = 1.
      GOTO 10
      5  QHPIPE = 0.0
      GAM = 0.0
C
C----- TANK ENERGY BALANCE
C
      10  QENV = UA * (TT - TENV)
      QFLOW = FIN*PAR(3)*(TT - TIN)
      TT = TT + (QDIODE+QHPIPE-QFLOW-QENV)*DT/MSCP
      DELE = (TT - PAR(6))*MSCP
      TOTENV = TOTENV + QENV * DT
      TOTQPE = TOTQPE + QHPIPE * DT
      TOTFLW = TOTFLW + QFLOW * DT
      500 CONTINUE
      501 CONTINUE
C
C-----OUTPUTS
C
      OUT(1) = TT
      OUT(2) = XIN(2)
      OUT(3) = TOTFLW

```

```
OUT(4) = TOTENV  
OUT(5) = DELE  
OUT(6) = TOTQPE  
OUT(7) = GAM  
RETURN  
END
```

```

PROGRAM DESIGN
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C   C.H.SUH JAN.1973 UNIVERSITY OF COLORADO, BOULDER           C
C   SEE C.H.SUH *COMPUTER AIDED DESIGN OF MECHANISMS* BOULDER, CO., C
C   UNIVERSITY OF COLORADO PRESS.                               C
C   REVISED OCT.1978, JAN.1979 ALTAS CORP., P.R.ARMSTRONG     C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
LOGICAL CONVRG,LIM,NORM,PRINT,INCMT,ONCE
DIMENSION Z(3),A(3,3),B(3),C(3),D(3),E(3),AA(3,3),BB(3),F(3)
COMMON /SIMEQS/ CONVRG,LIM,NORM,PRINT,NITR
COMMON /NBR/ NBR
COMMON /YCOMP/ ONCE,NOSE,V,IONCE
COMMON /RDUNIT/ IOR1 /WRUNIT/ IOW1
COMMON /FIRSTZ/ FIRSTZ(3)
DATA IOR1,IOW1/2*1/
WRITE (IOW1,1)
1  FORMAT ('ENTER NBR,NITR (2I5)')
20 READ (IOR1,2) NBR,NITR
2  FORMAT (10I5)
WRITE (IOW1,3) NBR,NITR
3  FORMAT (1X,2I5,8X,'NOW ENTER INIT GUESS (8F10).')
ONCE=.FALSE.
IONCE=1
PRINT=.FALSE.
IF (NBR)21,22,23
22 WRITE (IOW1,922)
922 FORMAT (' NBR=0; TRY AGAIN.')
GO TO 20
21 NBR=-NBR
PRINT=.TRUE.
23 INCMT=.FALSE.
5  READ (IOR1,6) (Z(I),I=1,NBR)
6  FORMAT (8F10.7)
DO 7 I=1,NBR
7  FIRSTZ(I)=Z(I)
WRITE (IOW1,8) (Z(I),I=1,NBR)
8  FORMAT (24H INITIAL GUESS OF Z(J)      /(11X,5E15.7))
II=1
JJ=NBR
10 CALL SIMEQS(NBR,II,JJ,Z,A,B,C,D,E,AA,BB,F)
IF (.NOT.CONVRG) WRITE(IOW1,11)
11 FORMAT (15H NOT CONVERGED. )
IF (LIM) WRITE (IOW1,12)
12 FORMAT (21H TOO MANY ITERATIONS. )
IF (.NOT.CONVRG.AND.NORM) WRITE (IOW1,13)
13 FORMAT (16H NORM DECREASED. )
IF (CONVRG) WRITE (IOW1,14)
14 FORMAT (11H CONVERGED. )
WRITE (IOW1,15) (Z(I),I=II,JJ)
15 FORMAT (9H ENDING Z      /(11X,5E15.7))
ONCE=.FALSE.
IONCE=2
GO TO 10
END

```

```

SUBROUTINE SIMEQS (NBR,II,JJ,Z,A,B,C,D,E,AA,BB,F)
LOGICAL CONVRG,LIM,NORM,PRINT,NEWZ
DIMENSION Z(NBR),A(NBR,NBR),B(NBR),C(NBR),D(NBR),E(NBR),
1 AA(NBR,NBR),BB(NBR)
COMMON /SIMEQS/ CONVRG,LIM,NORM,PRINT,NITR
C1 COMMON /YCOMP/ ONCE,NOSE,V,IONCE
COMMON /RDUNIT/ IOR1 /WRUNIT/ IOW1
COMMON /FIRSTZ/ FIRSTZ(3)
NEWZ=.FALSE.
NORM=.FALSE.
CONVRG=.FALSE.
LIM=.FALSE.
NDAMP=0
ITNS=0
100 CONTINUE
C1 ONCE=.FALSE.
140 ONORM=0.0
MM=0
DO 150 I=II,JJ
MM=MM+1
E(MM)=0.
D(I)=-GRAD (Z, I, 0, NBR, F)
150 ONORM = ONORM + ABS(D(I))
IF (ONORM.LT.1.0E-7) GO TO 551
C1 IONCE=3
C... COMPUTE PARTIALS
200 DO 205 J=II,JJ
DO 205 I=II,JJ
B(I) = D(I)
205 A(I,J) = GRAD (Z, I, J, NBR, F)
ITNS = ITNS + 1
IF (NBR .NE. 1) GO TO 207
206 B(II) = B(II) / A(II,II)
GO TO 300
207 NN=0
DO 208 I=II,JJ
NN = 1 + NN
BB(NN) = B(I)
MM = 0
DO 208 J=II,JJ
MM = 1 + MM
208 AA(NN,MM)=A(I,J)
DET = 1.0
IF (LINEQF(NBR, NBR, AA, BB, DET, E) .NE. 1) GO TO 700
NN=0
DO 209 I=II,JJ
NN = 1 + NN
209 B(I) = BB(NN)
C... SEE IF CORRECTION TO Z DECREASED NORM OF FUNCTION VECTOR.
300 DO 305 I=II,JJ
305 C(I) = Z(I) + B(I)
ENORM = 0.0
C1 ONCE=.FALSE.
DO 310 I=II,JJ
D(I) = - GRAD (C, I, 0, NBR, F)
310 ENORM = ENORM + ABS(D(I))

```



```

      IF (PRINT) GO TO 800
315 IF (ENORM.GT.ONORM.AND.ENORM.GT.1.0E-7) GO TO 900
      NORM = .TRUE.
      ONORM =ENORM
      NDAMP = 0
      DO 400 I=II,JJ
400  Z(I)= C (I)
C... CONVERGANCE TST
      DO 500 I=II,JJ
      TEST = ABS(B(I))
      IF (ABS(Z(I)) .GE. 1.0E-7) TEST = ABS(B(I)/Z(I))
      IF (TEST .GT. 1.0E-7) GO TO 600
500  CONTINUE
551  CONVRG = .TRUE.
      RETURN
600  IF (ITNS .LT. NITR) GO TO 200
      LIM = .TRUE.
      RETURN
C... FOR FAILURE OF LINEQF. PRINT ITER NO. AND MATRICES.
700  WRITE(IOW1, 10) ITNS
      10 FORMAT (27H LINEQF FAILED AT ITERATION I2,31H. THE MATRICES B(I)
      1 AND A(I,J) )
      DO 710 I=II,JJ
710  WRITE(IOW1, 20) I, B(I), (A(I,J),J=II,JJ)
      20 FORMAT (I4, E15.7,5X, 5E15.7/(24X,5E15.7))
      RETURN
800  WRITE(IOW1, 30) ITNS,(B(I), I=II,JJ)
      30 FORMAT (13H AT ITERATION I2/11H CORRECTNS 5E15.7/(11X,5E15.7))
      WRITE(IOW1, 40) (C(I), I=II,JJ)
      40 FORMAT (11H VARIABLES 5E15.7/(11X,5E15.7) )
      WRITE (IOW1,50) (D(I),I=II,JJ)
      50 FORMAT (11H VARIABLES 5E15.7/(11X, 5E15.7))
      GO TO 315
C... FOR NO DECREASE IN NORM OF FUNCTION VECTOR. TRY REDUCING
C... (DAMPING) CORRECTION VECTOR IN CASE PROBLEM IS OVERSHOOT INSTABILITY.
900  NORM = .FALSE.
      NDAMP = NDAMP + 1
      IF (NDAMP .GT. 10) GO TO 920
      SCALE = 5.0
      IF (NDAMP .GT. 1) GO TO 910
      DO 905 I=II,JJ
      BI = ABS(B(I))
905  IF (BI .GT. SCALE) SCALE = BI
910  DO 915 I=II,JJ
915  B(I) = B(I)/SCALE
      GO TO 300
920  WRITE(IOW1, 60)
      60 FORMAT (53H NO. REDCTNS (DAMPING) OF THE CRRCTN VECTOR EXCEEDED. )
      IF (NEWZ) RETURN
      NEWZ=.TRUE.
      WRITE (IOW1,70)
      70 FORMAT (' TRY NEW INITIAL GUESS JUST ONCE.')
      DO 930 I=II,JJ
930  Z(I)=FIRSTZ(I)
      GO TO 140
      END

```

```

      FUNCTION GRAD (Z, I, J, NBR, F)
C... APPROXIMATES PARTIAL DERIV OF ITH YCOMP WRT JTH Z. IF J=0,
C... RETURNS VALUE OF ITH YCOMP.
      DIMENSION Z(NBR),F(NBR)
      IF (J .NE. 0) GO TO 100
      F(I) = YCOMP(Z,I)
      GRAD = F(I)
      RETURN
100 Y1 = F(I)
      TEMP = Z(J)
C... ORIGINALLY 1.0E-03
      H = 0.001 * ABS (TEMP)
C... ORIGINALLY 1.0E-04
      IF (H .LT. 0.0001) H = 0.0001
      Z(J) = TEMP + H
      Y2 = YCOMP (Z,I)
      Z(J) = TEMP
      GRAD = (Y2 - Y1)/H
      RETURN
      END

C
C
      FUNCTION LINEQF (M,N,A,B,DTRMNT,Z)
      REAL A(M,M), B(M), Z(M)
C... SOLVES SMULTANEOUS LINEAR EQUATIONS BY GAUSSIAN REDUCTION;
C... AT EACH PIVOT, REMAINING PIVOT CANDIDATES ARE TESTED TO FIND
C... ELEMENT WHICH GIVES LEAST ROUND OFF ERROR.
C... ORIGINALLY ALL E-30 WERE E-30
      EPS = 1.0 E-30
      NM1 = N-1
      DO 40 J=1,NM1
      J1=J+1
      LMAX=J
      RMAX=ABS(A(J,J))
      DO 8 K=J1,N
      RNEXT=ABS(A(K,J))
      IF (RMAX .GE. RNEXT) GO TO 8
      RMAX=RNEXT
      LMAX=K
      8 CONTINUE
      IF (LMAX .NE. J) GO TO 10
      IF (ABS(A(J,J)) .LT. EPS) GO TO 94
      GO TO 20
10 DO 12 L=J,N
      W=A(J,L)
      A(J,L)=A(LMAX,L)
      A(LMAX,L)=W
12 CONTINUE
      W = B(J)
      B(J) = B(LMAX)
14 B(LMAX) = W
      DTRMNT = -DTRMNT
20 Z(J)=1./A(J,J)
      DO 30 K=J1,N

```

```

      IF (A(K,J)) 22,30,22
22  W=-Z(J)*A(K,J)
      DO 24 L=J1,N
      A(K,L)=W*A(J,L)+A(K,L)
24  CONTINUE
26  B(K) = W*B(J) + B(K)
30  CONTINUE
40  CONTINUE
      IF (ABS(A(N,N)) .LT. 1.0E-30) GO TO 94
42  Z(N) = 1./A(N,N)
      B(N) = Z(N) * B(N)
      DO 60 K=1,NM1
      J=N-K
      J1=J+1
      W=0.
      DO 56 I=J1,N
      W = A(J,I) * B(I) + W
56  CONTINUE
58  B(J) = (B(J) - W) * Z(J)
60  CONTINUE
      IF (DTRMNT .EQ. 0.0) GO TO 74
70  DO 72 J=1,N
72  DTRMNT = DTRMNT * A(J,J)
74  LINEQF = 1
80  CONTINUE
      RETURN
C... CONTINUE HERE FOR SINGULAR, NEAR SINGULAR CASE.....
94  LINEQF = 2
      DTRMNT=0.
      GO TO 80
      END

```



```

        SLN3(IP4,IP3)=QMAX
        ZMAP(1) = QHX *SUM /DTHXIN
C      ZMAP(1) = NOMINAL HX UA
        WRITE (6,999) (ZMAP(I),I=1,NFCN)
        SLN4(IP4,IP3)=UAHX
505 MAXPAR=NTPARS
510 IF (IPAR(MAXPAR).NE.NPAR(MAXPAR)) GO TO 520
C
        MAXPAR=MAXPAR-1
        IF (MAXPAR.GT.0) GO TO 510
520 IF (MAXPAR.GE.3) GO TO 700
        WRITE (IOW2,691) PAR(IP2,2)
        WRITE (IOW2,692) ((SLN1(I4,I3),I4=1,NP4),I3=1,NP3)
692 FORMAT(3(3F8.4))
        WRITE (IOW2,692) ((SLN2(I4,I3),I4=1,NP4),I3=1,NP3)
        WRITE (IOW2,692) ((SLN3(I4,I3),I4=1,NP4),I3=1,NP3)
        WRITE (IOW2,692) ((SLN4(I4,I3),I4=1,NP4),I3=1,NP3)
693 FORMAT (3(3X,3F8.3))
        IF (MAXPAR.LE.0) STOP
700 CONTINUE
        IPAR(MAXPAR)=IPAR(MAXPAR)+1
        IF(MAXPAR.EQ.NTPARS) GO TO 710
        MAXPAR=MAXPAR+1
        DO 705 I=MAXPAR,NTPARS
705 IPAR(I)=1
C
        IF (MAXPAR.GT.2) GO TO 710
        WRITE(IOW2,695) PAR(IP1,1)
691 FORMAT (/ ' TCO-TTANKBOTTOM (KELVINS) = ',F8.4)
694 FORMAT (/ ,1X)
695 FORMAT(/ , ' (TTANKAVG-TTANKBOT)/(TCO-TTANKBOT)= ',F8.4)
C
C...  DEFINE PERF MAP PARAS HERE...
710 CONTINUE
        FDELRO = PAR(IP1,1)
        DTHXIN = PAR(IP2,2)
        TTXI = PAR(IP3,3)
        THI=TTXI+DTHXIN
        TH(1) = THI
        QMAX = DTHXIN*CH
        PGRUP7=FDELRO*DTHXIN
        WRITE (1,999) TTXTOP,TTXBOT,TTXAVE,CC
        WRITE (1,999) DTHXIN,FDELRO,TTXI,THI
C
        IONCE=3
800 CONTINUE
        ONCE=.TRUE.
C...  FIRST YCOMP CALL WITH NEW ESTIMATE OF ZZ(*)
C...  CALCULATE INTERMEDIATE VARIABLES WHOSE DEPENDENCE ON ZZ(*) IS
C...  MINOR OR NOT AT ALL, HERE.
C
C..  BEGIN EVALUATION OF ERROR RESIDUALS
1000 Q(1)= 1.
        SUM = Q(1)

```

```

      IF (NQ.LE.1) GO TO 1010
      DO 1005 I=2,NQ
      Q(I) = ABS(ZZ(I))
C..  OR TRY Q(I) = EXP(ZZ(I))
      1005 SUM = SUM + Q(I)
C..  WITH FREE CONVECTION ON COLD SIDE THERMAL CAPACITANCE RATE IS
C..  UNDEFINED, EFFECTIVENESS MUST BE DEFINED IN TERMS OF HOT SIDE
C..  CAPACITANCE RATE. HEAT TRANSFER COEFFICIENT CORRELATION WILL
C..  ENSURE AGAINST THERMODYNAMICALLY INFEASIBILITY IN CONVERGED
C..  SOLUTIONS.
C..  QMAX=DTHXIN/CH   C.. HAS BEEN PRECOMPUTED.
      1010 QHX=QMAX*EXP(-ABS(ZZ(1)))
      TC(1)=TTXI
      QHX = QHX/SUM
C..      IIP = II + 1
      L = 1
      QABOVE = 0.
      DO 1205 I=1,II
      Q(I) = Q(I)*QHX
      IF (I.EQ.II) GO TO 1204
      QABOVE = QABOVE + Q(I)
      GO TO 1205
      1204 THI = TH(1) - QABOVE/CH
      THO = THI - Q(I)/CH
      TH(I+1) = THO
      1205 CONTINUE
      THAVE = (THO + THI)/2
      UASCTE = Q(II)/(THAVE-TTXI)
C.....
C... GP = CH /TRC CP...CH FIXED, TRC CP COULD BE FCN(THAVE).
      GP = .072816
      GPP = GP /AXTRC
      TRC VSC = H2O(1,THAVE)
      TRC RHO = H2O(2,THAVE)
      TRC CND = H2O(3,THAVE)
      RE = GPP * DTUBE /TRCVSC
      WRITE (1,999) RE
      CPDVK = TRC CP /TRC CND
      PR = TRCVSC * CPDVK
      X = DOVRLN * RE * PR
      IF (RE.GT. 5000.) GO TO 1220
C..  LAMINAR
      XNU1 = (3.66 + .0668*X/(1. + .04*X**.6666667))
C..  TRANSITION
      1220 XNU2 = 0.116 * (RE**.6666667 - 125.) * PR**.3333333 *
      + (1. + DOVRLN**.6666667)
      IF (RE.LT. 5000.) GO TO 1240
C..  TURBULENT
      1230 XNU1 = 0.023 * RE**0.8 * PR**0.4
      IF (XNU2.LT.XNU1) XNU1 = XNU2
      GO TO 1250
      1240 IF (XNU2.GT.XNU1) XNU1 = XNU2
      1250 H SUB C = XNU1 * TRCCND/DTUBE * HCFCTR
      FINPAR = SQRT(TR PAR 1 * H SUB C)

```

```

      RCXW = ATDVAW * FINPAR /(H SUB C * TANH(FINPAR)) + RBOND
C.....
      RTXE = ASECT/UASCTE - RCXW
      P = PGRUP4/RTXE
      CALL SOLVEF(P,.0001,E FNPARG)
      WRITE (1,999) P,E FNPARG
C..  ESTIMATED H SUB C = EFNPARG*EFNPARG /PGRUP4
      TWALL = TTXI + Q(II)*PGRUP4 /(ASECT*EFNPARG*EFNPARG)
C..  TWALL = THAVE - Q(II)*RCXW
C..  CHOICE OF TWALL DEFN IS INFLUENCED BY OCCURENCE OF INSTABILITY.
C..  SOME INTERMEDIATE VALUE COULD BE USED ALSO.
      TFLM = (TWALL+TTXI)/2
      DELT = TWALL - TTXI
      IF(DELT.LE.0.)WRITE(1,999)THO,THI,TTXI,THAVE,TWALL,TFLM
C..  X = H**3 RHO**2 *BETA *DELT *CP /VISC /K      DIMENSIONLESS
      IF(TFLM.GT.THI)WRITE(1,999)THO,THI,TWALL,TFLM,TTXI,Q(II),PGRUP4,
+  ASECT,UASCTE,EFNPARG,FINPAR,HSUBC,RTX,UAHX
      FLMVSC = H2O(1,TFLM)
      FLMRHO = H2O(2,TFLM)
      FLMCND = H2O(3,TFLM)/3600
      FLMCP  = H2O(4,TFLM)
      FLMBTA = H2O(6,TFLM)
C..  H SUB C = 0.13 *FLMCND *
C..  + (FLMRHO**2 *9.806 [M2/SEC] *FLMBTA *DELT *FLMCP /FLMCND /FLMVSC)**(1/3)
C..  AS DEL T GOES TO 0, H SUB C SHOULD TEND ASSYMPTOTICALLY TO A MIN
C..  OF XNU = 3.66 TO 8.00 -- I.E. H SUB C = 3.66*K/D.
      ALMST X = FLMRHO *FLMRHO *FLMBTA *DELT *FLMCP /(FLMCND *FLMVSC)
      IF (ALMST X .LT. E9VH3G) WRITE (1,999) ALMST X, E9VH3G
      IF(ALMST X.LE.0.)WRITE(1,999)FLMVSC,FLMRHO,FLMCND,FLMCP,FLMBTA
      ALMST X = ALMST X **.3333333
      ALMST X = AMAX1( ALMST X, H C W MIN )
      H SUB C = .278 *FLMCND * ALMST X
      FINPAR = SQRT(PGRUP4 *H SUB C)
      RTX = FINPAR /(H SUB C *TANH(FINPAR))
      UAHX=ASECT/(RCXW+RTX)
      YCOMP = UAHX - UASCTE
      RETURN
999 FORMAT (1X,6E12.5,(/,1X,6E12.5) )
      END
C
C
      SUBROUTINE SOLVE F (P,TOL,FE)
10 F = FE
      EP = EXP(F)
      EM = EXP(-F)
      S = EP + EM
      TH = (EP - EM)/S
      FE = F + (P - F*TH) /(4*F/(S*S) + TH)
      IF (ABS(F-FE)/FE .GT. TOL) GO TO 10
      RETURN
      END

```

```

BLOCK DATA DESPEC
COMMON /NBR/ NQ
COMMON /DESPEC/PI,AWALL,ASECT,W,H,CH,CPTX,CNDVTX,CPDKTX,DTANK,
+ TRC CP,DTUBE,AXTRC,TRCLEN,DOVRLN,A W TRC,PITRC,TK TUBE,HCFCTR,
+ TRPAR1,ATDVAW, RBOND, PGRUP4, CIRC, WDVH, E9VH3G, PGRUP2, PGRUP3
DATA PI/3.1459265/
C... AREA OF TRACED SECTION OF TANK WALL, M**2
DATA AWALL/1.0/
C.. DATA ASECT / AWALL / NQ/
C... HYDRAULIC DIAMETER (WIDTH) OF ANULUS, M
DATA W/.02/
C... HEIGHT OF TRACED SECTION OF TANK WALL, M
DATA H/.5/
C... COLLECTOR LOOP THERMAL CAPACITANCE RATE, KW/DEG.C
DATA CH /0.3/
C... SPECIFIC HEAT OF TANK FLUID, KJ/KG-KELVIN
DATA CPTX/4.12/
C... THERMAL CONDUCTIVITY OF TANK FLUID, KW/M-KLV
DATA CNDVTX/.00060/
C... CPTX/CNDVTX, M-SEC/KG
DATA CPDKTX/6870./
C... TANK DIAMETER, M
C... COLLECTOR FLUID SPECIFIC HEAT CAPACITY, KJ/KG-KLV
DATA TRC CP / 4.12/
C... TUBE INSIDE DIAMETER, M
DATA D TUBE / .01/
C... TUBE CROSS-SECTIONAL AREA, M2
C... DATA AXTRC / D TUBE * D TUBE * PI /4/
C... TRACE TUBE LENGTH, M
C.. DATA TRC LEN / 5. * D TANK/
C... SURFACE AREA OF TUBE WALL, M2
C.. DATA A W TRC / TRC LEN * D TUBE * PI/
C... DATA TUBE SPACING (/ TWICE WALL-FIN-LENGTH), M/
C.. DATA PITRC / AWALL(1) /TRC LEN/
C... HELICAL TUBE CONVECTION COEFFICIENT ENHANCEMENT FACTOR, DIM'LESS
C.. DATA HCFCTR / 1. + 3.5 *DTUBE /DTANK/
C... TUBE WALL THICKNESS * CONDUCTIVITY, KW/KLV
DATA TK TUBE / .02/
C.. DATA TR PAR 1 / PITRC*PITRC/4/TKTUBE/
C.. DATA ATDVAW / AWTRC/AWALL/
C.. UA BOND = TRCLEN*(CONDUCTANCE PER UNIT LENGTH), KW/KLV
C.. RBOND = AWALL/UABOND, M2-KLV/KW
C.. DATA RBOND / AWALL /(TRCLEN *2.)/
C
C
C... DATA TANK WALL-FIN CHARACTERISTIC, P*L**2/(KA), WHERE A/P/FIN THICK-/
C... DATA NESS PER WETTED SIDE, M**2-KLV/KW, K/FIN MAT'L CNDCTVY, L=FIN LEN/
DATA PGRUP4/ 25./
C.. DATA CIRC/AWALL/H/
C.. DATA WDVH/W/H/
C.. DATA E9VH3G / 1E9 /(H*H*H*9.806)/
C.. DATA PGRUP2/32.*H/(CIRC*W*W*W) / NQ/
C.. DATA PGRUP3/(WDVH)**.55/
END

```



```

SUBROUTINE SPCALC
COMMON /NBR/ NQ
COMMON /DESPEC/PI,AWALL(1),ASECT,W,H,CH,CPTX,CNDVTX,CPDKTX,DTANK,
+ TRC CP,DTUBE,AXTRC,TRCLEN,DOVRLN,A W TRC,PITRC,TK TUBE,HCFCTR,
+ TRPAR1,ATDVAW, RBOND, PGRUP4, CIRC, WDVH, E9VH3G, PGRUP2, PGRUP3
C... TANK DIAMETER, M
D TANK = AWALL(1)/H/PI
C... TUBE CROSS-SECTIONAL AREA, M2
AXTRC = D TUBE * D TUBE * PI /4
C... TRACE TUBE LENGTH, M
TRC LEN = 5. * PI * D TANK
C... RATIO OF TUBE DIAMETER TO TUBE LENGTH, DIMENSIONLESS
DOVRLN = DTUBE /TRCLEN
C... SURFACE AREA OF TUBE WALL, M2
A W TRC = TRC LEN * D TUBE * PI
C... TUBE SPACING (= TWICE WALL-FIN-LENGTH), M
PITRC = AWALL(1) /TRC LEN
C... HELICAL TUBE CONVECTION COEFFICIENT ENHANCEMENT FACTOR, DIM'LESS
HCFCTR = 1. + 3.5 *DTUBE /DTANK
TR PAR 1 = PITRC*PITRC/4/TKTUBE
ATDVAW = AWTRC/AWALL(1)
C.. UA BOND = TRCLEN*(CONDUCTANCE PER UNIT LENGTH), KW/KLV
C.. RBOND = AWALL(1)/UABOND, M2-KLV/KW
RBOND = AWALL(1) /(TRCLEN *2.)
CIRC=AWALL(1)/H
WDVH=W/H
E9VH3G = 1E9 /(H*H*H*9.806)
PGRUP2=32.*H/(CIRC*W*W*W) / NQ
PGRUP3=(WDVH)**.55
WRITE (1,999) (AWALL(I),I=1,25)
RETURN
999 FORMAT (1X,6E12.5,(/,1X,6E12.5) )
END

```

APPENDIX F

NOMENCLATURE

A	total heat transfer surface area of heat pipe evaporator
A _b	boiling heat transfer surface area
A _c	internal surface area of trace coil
A _d	cross sectional area of connecting diode
A _f	fin area of heat pipe evaporator
A _v	vapor cross sectional flow area
A _{vt}	inner solar tank area adjacent to traced coil
c _f	specific heat of heat pipe fluid
C _{fill}	fluid level in heat pipe evaporator
c _{prod}	specific heat of combustion products
C _s	traced coil bond conductance
C _{sf}	surface/fluid combination coefficient for nucleate boiling
D	diameter of connecting diode
d _c	diameter of tubes used for traced coil
D _c	inside diameter of solar tank
D _{ei}	inner diameter of evaporator tubes
D _{eo}	outer diameter of evaporator tubes
D _f	evaporator fin diameter
D _t	backup tank diameter
f	friction factor
F	vapor frictional coefficient
g	acceleration of gravity
G _c	capacitance rate of collector fluid
g _o	constant 32.17 lbmft/lb _f sec ²

G_{in}	mass velocity of vapor entering condenser
G_{vm}	average mass velocity of vapor in condenser
Gr	Grashoff number
h_{boil}	nucleate boiling heat transfer coefficient
h_c	convection heat transfer coefficient inside traced coil
h_e	convective film heat transfer coefficient on outer surface of evaporator
h_{fc}	free convection coefficient for surface adjacent to condenser channel
h_{ff}	free convection coefficient over condenser fin
h_{film}	film condensation heat transfer coefficient
h_{fg}	latent heat of heat pipe fluid
h'_{fg}	average enthalpy change of heat pipe vapor in condensing and subcooling to average liquid temperature
h_{free}	free convection heat transfer coefficient
h_v	convection heat transfer coefficient inside solar tank
J	mechanical equivalent of heat
j_g	volumetric flux of heat pipe vapor
K_{ew}	thermal conductivity of evaporator wall
K_c	thermal conductivity of traced coil wall
K_{cw}	thermal conductivity of condenser (tank) wall
K_{hf}	thermal conductivity of heat pipe fluid
K_v	thermal conductivity of solar tank wall
K_{tf}	thermal conductivity of water in tank
l_e, l_a, l_c	length of heat pipe evaporator, adiabatic and condenser sections
L	overall length of connecting diode; also overall height of solar tank traced section
l_f	fin length
L_v	fin length between traced coils on solar tank

M	number of evaporator tubes
M_t	mass of water in backup tank
$\dot{m}_{\text{prod}}$	mass flow rate of combustion products
$\dot{m}_{\text{gas}}$	mass flow rate of methane burned
M_{prod}	mass of combustion products per pound of methane burned
M_t	mass of water in backup tank
N	number of condenser channels
N_r	number of evaporator tube rows
Nu	Nusselt number
P_e	pressure in evaporator
P_c	pressure in condenser
P_l	longitudinal evaporator tube pitch
Pr	Prandtl number
P_s	saturation pressure of heat pipe vapor
P_t	transverse evaporator tube pitch
Q	heat pipe heat transfer rate
Q_{max}	theoretical maximum heat transfer across evaporator
q_v	heat transfer per unit area across traced coil
Q_v	heat transfer rate across traced coil
Re	Reynolds number
r_h	hydraulic radius
s_f	fin spacing
St	Stanton number
T_b	temperature at bottom of solar tank
T_c	mean node temperature of collector fluid in trace coil
t_c	thickness to traced coil tubes
T_{cfb}	temperature at base of traced coil fin

T_{ci}	temperature at inner surface of condenser
T_{co}	temperature at outer surface of condenser
T_{cs}	average temperature of inside surface of traced coil
T_{ei}	temperature at inner surface of evaporator
T_{eo}	temperature at outer surface of evaporator
t_f	fin thickness
T_{flame}	temperature of combustion products entering evaporator tube array
T_{flue}	temperature of combustion products exiting evaporation array
T_{prod}	average temperature of combustion products across evaporator array
T_t	temperature of water in backup tank
T_{vc}	vapor temperature inside condenser
T_{ve}	vapor temperature inside evaporator
T_{vfb}	temperature at base of solar tank wall fin
T_{vs}	average temperature of solar tank wall adjacent to traced coil
t_w	backup tank wall thickness
t_v	solar tank wall thickness
ΔT_x	excess temperature for nucleate boiling
U_B	traced coil bond conductance
v	specific volume
V	fluid velocity
V_l	volume of liquid associated with corresponding volume of vapor
V_v	volume occupied by heat pipe vapor
V_f	volume held up in condenser film
w	condenser channel width
W	condenser channel spacing
x	percent theoretical air used in burner; also distance from base of traced tank
x_t	point of transition from laminar to turbulent free convection along traced tank wall

Greek

α	thermal diffusivity of fluid
ϵ	heat exchanger effectiveness
β	coefficient of thermal expansion
δ	condensing film thickness
ρ	density
σ	surface tension coefficient and ratio of free flow area to frontal area
μ	viscosity
ν	specific heat ratio
η_c	backup tank wall fin efficiency; also solar tank trace coil fin efficiency
η_f	evaporator fin efficiency
η_o	surface effectiveness
η_v	solar tank wall fin efficiency
π_2	Steen's entrainment parameter
χ	mole fraction of constituents in combustion gases
τ	time