Analytical and Experimental Investigation of Thermosyphon Solar Hot Water Systems

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ANALYTICAL AND EXPERIMENTAL INVESTIGATION
OF THERMOSYPHON SOLAR HOT WATER SYSTEMS

BY

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B.S.E., Florida Technological University, 1975

THESIS
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1976
ABSTRACT

A computer simulation of a thermosyphon system allowing load drawoff and non-ideal weather conditions has been developed. The model is restricted to the more common single cover, flat plate collector system. Using an analysis based on the present literature, this model calculates the energy absorbed by the collector, the temperature distribution through the system, and the corresponding flow rate.

Experimental data for a non-ideal day is compared to the computer simulation. Results of this comparison indicate that the desired parameters, flow rate, collector inlet and outlet temperatures, and the mean tank temperature can be predicted by this model to within 10 percent.
ACKNOWLEDGEMENTS

Since this work would not have resulted if it were not for the work of many people, I would like to acknowledge those who have been so helpful. Special thanks is extended to my thesis advisor, Dr. Bruce Nimmo, for allowing me the opportunity to become involved in solar research and for lending his moral and academic support in the development of the following analysis. Certainly, Jeff Pearce is owed a great deal for offering his expert assistance in computer programming and the confirmation of many of the equations used in this text. As a fellow student, he has also made the "trip up the river without a paddle" a more enjoyable one. The work of Mike Hackathorn and Rick Larsen in assisting me to obtain experimental data is greatly appreciated.

Thanks is also given my thesis committee members, Dr. Ronald Evans, Dr. Richard Rapson, and Dr. Harold Klee for their time spent evaluating this report.

Much gratitude is extended to Peggy Semo for the many hours she spent typing this paper.

Finally, I wish to offer a special thank you to my wife, Fran, for bearing with me through the preparation of this paper.
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LIST OF VARIABLES

A_c  collector surface area
A_i  circumferential area of node i of storage tank
B    distance between tube centers
C    convection coefficient
C_a  lumped capacitance of absorber and fluid per unit area
C_a  lumped capacitance of absorber and fluid
C_p  specific heat of working fluid
d    inside tube diameter
d_o  outside tube diameter
E_LOAD energy supplied by the storage tank to the load
ΔE_TANK daily energy increase of storage tank
F    fin efficiency
F'   collector efficiency factor
F_{i-j} configuration factor
f    friction factor
g    gravitation constant
H_f  total friction head around flow circuit
H_T  total thermosyphon head
h_f  film coefficient on inside of tubes
h_i  friction head contributed by section i of collector tube
\( h_{cij} \) convection coefficient from node \( i \) to node \( j \)

\( h_{rij} \) radiation coefficient from node \( i \) to node \( j \)

\( h_w \) wind convection coefficient

\( I \) incident radiation on collector surface

\( I_o \) incident radiation normal to the collector surface

\( I_T \) energy transmitted through the cover plate

\( K \) extinction coefficient of cover plate

\( K_T \) conductivity of collector absorber sheet

\( K_3 \) conductivity of storage tank insulation

\( k_i \) friction loss coefficient

\( L \) thickness of cover plate

\( l_i \) total length of tubing in collector

\( l_i \) length of section \( i \) of collector tube

\( m_L \) mass of fluid supplied to the load

\( m_T \) mass of fluid in storage tank

\( m_a \) system flow rate

\( m_{L_a} \) system flow rate averaged over a time interval

\( m_L \) load flow rate from storage tank

\( (mc)_i \) capacitance of node \( i \)

\( n_1 \) refractive index of air

\( n_2 \) refractive index of cover plate

\( Q_u \) useful energy gain

\( Q_{\text{BACK}} \) energy loss from back of absorber sheet
\( Q_{\text{EDGE}} \)  
energy loss from absorber through edges of collector

\( Q_{\text{TOP}} \)  
energy loss from top of absorber

\( Q_{\text{LOSS}} \)  
total energy loss from absorber

\( Q_{si} \)  
energy stored in node \( i \) of collector

\( Q_{cij} \)  
energy convected from node \( i \) to node \( j \)

\( Q_{kij} \)  
energy conducted from node \( i \) to node \( j \)

\( Q_{rij} \)  
energy radiated from node \( i \) to node \( j \)

\( Re \)  
Reynold's number

\( r_t \)  
radius of tank

\( S \)  
total energy absorbed by absorber plate

\( S_g \)  
total energy absorbed by cover plate

\( S_1 \)  
energy absorbed by cover plate due to internal reflections

\( S_2 \)  
energy absorbed by cover plate due to multiple reflections between cover and absorber

\( T \)  
local fluid temperature in collector

\( T_a \)  
ambient temperature

\( T_{\text{OLD}} \)  
last known value of local fluid temperature

\( \Delta T_{\text{avg}} \)  
temperature rise across the collector averaged over a time interval

\( \Delta T_{\text{LOAD}} \)  
temperature difference between hot fluid supplied to load and cold return fluid

\( \Delta T_{\text{TANK}} \)  
rise in mean tank temperature for the day

\( T_i \)  
temperature of node \( i \) in collector

\( T_{S_i} \)  
temperature of node \( i \) in storage tank
t  optical path length through cover plate;
     time

U_L  loss coefficient of collector

U_T  loss coefficient of storage tank

u_i  velocity in section i of collector tube

v  kinematic viscosity

W  average fin width

x  optical distance in cover plate;
     distance along tube in collector

Δx  thickness of storage tank insulation

Δx_3 thickness of collector back insulation

y  vertical position in system

α  absorptance of absorber sheet

α_g effective absorptance of cover plate

β  angle of refraction

γ  specific gravity

δ  thickness of absorber sheet

ε_i emissivity of node i of collector

η  collector efficiency

η_s system efficiency

η_{day} daily collector efficiency

θ  angle of incidence

ρ  reflectance at incident angle θ

ρ_o reflectance at normal incidence
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho_i$</td>
<td>density in section i of collector tube</td>
</tr>
<tr>
<td>$\rho_s$</td>
<td>standard value of density</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Stephan-Boltzman constant</td>
</tr>
<tr>
<td>$T$</td>
<td>transmittance of cover plate</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>length of time interval</td>
</tr>
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Solar energy systems used for heating water primarily in residences can be categorized as either pumped or thermosyphon types. As the classification implies, pumped systems are those employing a mechanical device for circulating the working fluid. Thermosyphon types have no such devices and operate on the principle of natural circulation of the working fluid. Since no electrical power is required, the thermosyphon system is ideal for use in rural areas. A typical unit is shown in Figure 1.

Considerable work has been done in developing useful computer simulations of solar energy systems.\textsuperscript{1-6} Compared to thermosyphon units, pumped systems analyses have received much more attention since the latter type is presently in widespread use. In order to provide a capability for transient analysis of in-service thermosyphon units, a simulation model, herein referred to as the TSP, is developed.

General thermosyphon analyses are complicated not only by a varying flow rate but by the non-ideal conditions of transient effects and load drawoff. By generating a model which accounts for these conditions, a basis for several other system simulations is
Figure 1. A Typical Thermosyphon System
established. For example, a steady state pumped system model is readily derived from the TSP by neglecting the transient effects and setting the flow rate to a constant.

Development of the equations used in the computer simulation is contained in the following four chapters. Much of the material presented is based on equations presently available in the literature, notably references 7, 8, 9. However, the overall approach to the problem is believed to be unique to this analysis.

Chapters VI and VII discuss the computer model itself and its validation. Chapter VI discusses the basic working of the model. The output parameters of the model are also presented. Chapter VII deals with verification of the TSP. Verification includes a comparison between the TSP and experimental data and a model existing in the literature.
This chapter presents an analysis of the radiation input to a collector with a single cover plate. The energy absorbed by the primary absorber surface and the cover plate is considered. The performance of any solar energy unit is dependent on the amount of energy it absorbs. For this analysis, it is assumed that solar radiation data is available for the tilted surface of the collector and that this data contains both direct and diffuse values. As a result of this assumption, complications arising from diffuse radiation reflected from surroundings are eliminated. The problem of obtaining the system energy input reduces to establishing the fraction of incident energy absorbed by the collector plate and the cover plate.

The cover plate on a collector serves to reduce energy losses from the primary absorber surface to the environment. In addition, if the cover material is transparent to the incoming short wavelength radiation, the cover plate will cause the collector to behave as a trap for solar radiation. There may be several covers on a collector. An increase in the number of covers reduces the energy losses. However, there is an accompanying decrease in the amount of
energy reaching the primary surface. For the sake of simplicity, this report is concerned only with a single cover system. However, analyses of energy transmission through several cover plates exist in the present literature.\textsuperscript{10}

The transmittance of the cover plate to incident radiation can be obtained by the use of Bouguer's absorption law and reflection concepts for optically smooth surfaces.

According to Bouguer,\textsuperscript{11} the absorption of radiation in a partially transparent medium is described as

\[
\frac{\Delta I}{I} = IKJ\text{ }dx
\]

where
\[
\begin{align*}
I &= \text{intensity of incident light} \\
K &= \text{extinction coefficient} \\
x &= \text{optical thickness}
\end{align*}
\]

Assuming the extinction coefficient is a constant throughout the material for the solar spectrum, an integration along the optical path (refer to Figure 2) yields:

\[
\frac{I_2}{I_1} = e^{-Kt}
\]

where
\[
\begin{align*}
I_1 &= \text{radiation intensity at point 1} \\
I_2 &= \text{radiation intensity at point 2} \\
t &= \text{optical path length from point 1 to point 2}
\end{align*}
\]
Figure 2. Transmission through Cover Plate
From Snell's law,

\[ \frac{\sin \beta}{\sin \theta} = \frac{n_1}{n_2} \]

where

- \( n_1 \) = refractive index of air
- \( n_2 \) = refractive index of cover
- \( \theta \) = angle of incidence
- \( \beta \) = angle of refraction

so that

\[ \beta = \sin^{-1}\left(\left(\frac{n_1}{n_2}\right) \sin \theta\right) \]

The optical path length, \( t \), can now be calculated as

\[ t = \frac{L}{\cos \beta} \]

It now becomes obvious that the optical path length is dependent on the angle of incidence, \( \theta \). Duffie and Beckman\(^{12}\) present this angle as a function of the earth's declination to the sun, the time of day, and the collector orientation. To simplify calculations, it is assumed that the optical path length remains constant. Since the measured radiation data is assumed normal to the collector surface, the path length, \( t \), through the cover is assumed to be \( L \), the cover thickness. This is equivalent to allowing the angles
\( \beta \) and \( \theta \) to remain zero and considering only normal radiation values. Thus, equation II.2 becomes

\[
\frac{I_2}{I_1} = e^{-KL}
\]

In addition to the absorption effect, there exists an effect due to reflection. We can employ Fresnel's relation for the reflection of radiation passing from medium one to medium two if we assume

1. the cover plate is optically smooth
2. incident radiation is composed of two linearly polarized components.

For an incident angle \( \theta \)

\[
\rho = \frac{1}{2} \left[ \frac{\sin^2(\beta - \theta)}{\sin^2(\beta + \theta)} + \frac{\tan^2(\beta - \theta)}{\tan^2(\beta + \theta)} \right]
\]

where \( \beta \) is defined by equation II.3

\( \rho = \text{reflectance at incident angle } \theta \)

Under the assumption that radiation data is for the plane of the collector surface, \( \theta \) and \( \beta \) will again be considered zero for this analysis. Equations II.7 and II.3 can be combined to yield the reflectance for normal incidence, \( \rho_0 \),

\[
\rho_0 = \left[ \frac{n_1 - n_2}{n_1 + n_2} \right]^2
\]
This is the reflection of normal incidence for one interface of the cover plate. There is also a reflection at the second interface. Neglecting absorption, the amount of energy transmitted into the material from the first interface is proportional to $(1 - \rho)$ as shown in Figure 3. Of this, an amount proportional to $(1 - \rho)^2 \rho$ is reflected back into the material at the second interface. At the same surface, a quantity proportional to $(1 - \rho)(1 - \rho)$ leaves the material. There are an infinite number of these reflections occurring at the two surfaces.

Applying equation II.6 to each of these reflected components, the radiation component transmitted through the cover material can be calculated (refer to Figure 4). Summing all terms representing transmission of energy through the cover material and considering only the normal reflectance yields

$$II.9 \quad I_T = (I - \rho_o)^2 e^{-KL} I_o + (1 - \rho_o)^2 \rho_o^2 e^{-3KL} I_o + 
(I - \rho_o)^2 \rho_o^2 e^{-5KL} I_o + \ldots$$

where

$I_T$ = transmitted energy

$I_o$ = incident energy normal to the collector

or

$$I_T = (1 - \rho_o)^2 I_o \sum_{n=0}^{\infty} \rho_o^{2n} e^{-(2n+1)KL}$$
Figure 3. Transmission through Cover Plate due to Reflectance
Figure 4. Transmission through Cover Plate due to Reflectance and Absorptance
This can be rearranged to yield

\[ I_T = (1 - \rho_0)^2 I_0 e^{-KL} \sum_{n=0}^{\infty} (\rho_0 e^{-KL})^{2n} \]

For positive values of \( K \) and \( L \), the quantity \( (\rho_0 e^{-KL}) \) is always less than one. Therefore, this sum can be expanded to yield

\[ \sum_{n=0}^{\infty} (\rho_0 e^{-KL})^{2n} = \frac{(1 - \rho_0)^2 e^{-KL}}{1 - (\rho_0 e^{-KL})^2} \]

\[ I_T = \left[ \frac{(1 - \rho_0)^2 e^{-KL}}{1 - (\rho_0 e^{-KL})^2} \right] I_0 = \tau I_0 \]

where \( \tau \) represents the transmittance of the cover plate to incident radiation \( I_0 \).

The energy transmitted through the cover plate, \( I_T \), strikes the absorber plate. Some of this energy is absorbed by the plate and some is reflected back to the cover material. Multiple reflections occur between the absorber and the cover as shown in Figure 5. Assuming the primary absorber surface absorbs a fixed fraction, \( \alpha \), of all incident radiation, and considering only normal reflectance, the total energy it absorbs becomes the sum

\[ S = \alpha I_0 + \alpha (1 - \alpha) \rho_0 I_0 + \alpha (1 - \alpha)^2 \rho_0^2 I_0 + \ldots \]

or

\[ S = \alpha I_0 \sum_{n=0}^{\infty} (1 - \alpha)^n \rho_0^n \]

Since \( \alpha \) and \( \rho_0 \) are less than one, then the quantity \( (1 - \alpha)\rho_0 \) is less than one. Therefore, the above sum can be expanded to yield
Figure 5. Absorption of Solar Radiation by Primary Absorber
Another quantity required for the analysis is the energy absorbed by the cover plate. Consider Figure 4. Employing only normal reflectance, the amount of energy absorbed in the cover plate due to multiple reflections within the cover is the sum

$$S_1 = (1-\rho_o)I_o - (1-\rho_o)e^{-KL}I_o + (1-\rho_o)\rho_o e^{-KL}I_o - (1-\rho_o)\rho_o e^{-2KL}I_o + (1-\rho_o)\rho_o^2 e^{-2KL} - (1-\rho_o)\rho_o^2 e^{-3KL}I_o + (1-\rho_o)\rho_o^3 e^{-3KL}I_o - (1-\rho_o)\rho_o^3 e^{-4KL}I_o + \ldots$$

or

$$S_1 = (1-\rho_o)I_o \sum_{n=0}^{\infty} \rho_o^n e^{-nKL} - (1-\rho_o)I_o \sum_{n=0}^{\infty} \rho_o^n e^{-(n+1)KL}$$

Factoring yields

$$S_1 = (1-\rho_o)I_o \sum_{n=0}^{\infty} \left[ \rho_o^n (e^{-nKL} - e^{-(n+1)KL}) \right]$$

This can be simplified to

$$S_1 = (1-\rho_o) (1-e^{-KL})I_o \sum_{n=0}^{\infty} (\rho_o e^{-KL})^n$$
The above sum can be expanded to yield

\[ S_1 = \left[ \frac{(1 - \rho_o)(1 - e^{-KL})}{(1 - \rho_o e^{-KL})} \right] \]

\[ I_o = \alpha g I_o \]

where \( \alpha \) represents the "absorptance" of the cover plate due to the single input \( I_o \).

Consider the energy reflected from the collector plate (refer to Figure 5) as an additional input. For normal reflectance, this additional input is the sum

\[ S'_2 = (1 - \alpha)\tau I_o - (1 - \alpha)\tau \rho_o I_o + (1 - \alpha)^2 \tau \rho_o I_o - \]

\[ (1 - \alpha)^2 \tau \rho_o^2 I_o + (1 - \alpha)^3 \tau \rho_o^3 I_o - (1 - \alpha)^3 \tau \rho_o^3 I_o + ... \]

or

\[ S'_2 = \left[ (1 - \alpha)\tau I_o \sum_{n=0}^{\infty} (1 - \alpha)^n \rho_o^n \right] - \]

\[ \left[ \tau I_o \sum_{n=0}^{\infty} (1 - \alpha)^n \rho_o^n \right] + \tau I_o \]

This can be factored to yield

\[ S'_2 = \left[ (1 - \alpha) \tau I_o - \tau I_o \right] \left[ \sum_{n=0}^{\infty} (1 - \alpha)^n \rho_o^n \right] + \tau I_o \]

The above sum can be expanded to yield
This represents an energy input to the cover plate in addition to the direct input $I_o$. As such, the cover plate will absorb a fraction, $a_g$, of this additional energy, so that

$$S_2 = a_g S'_2$$

where $a_g$ is defined in equation II.14.

The total energy absorbed by the cover plate is the sum of equations II.17 and II.14.

$$S_g = S_1 + S_2$$

Making the appropriate substitutions for $S_1$ and $S_2$ and simplifying yields

$$S_g = \frac{(1 - \rho_o)(1 - e^{-KL})}{1 - \rho_o e^{-KL}} \left[ 1 + \tau - \frac{\tau a}{1 - \rho_o (1 - \alpha)} \right] I_o$$

Equations II.12 and II.19 represent the energy absorbed by the solar collector, that is, the energy absorbed by the cover plate and the absorber plate. The preceding analysis is based on the assumption that the radiation data input is for the collector surface. By assuming the incident angle to remain zero for this type
of data, we have approximated the energy absorbed by the collector to be a fixed fraction of the incident energy.
COLLECTOR MODEL

The simulation of a thermosyphon solar energy system usually includes some kind of model for the collector. Performance of this model will depend on the rate of useful energy removal which, in turn, depends on the flow rate of the circulating fluid. As opposed to pumped systems, thermosyphon system analyses usually involve unknown flow rates. As will be discussed later, the solution to the flow problem is heavily dependent on the temperature distribution of the circulating fluid. Therefore, a collector model which yields such a temperature distribution through its tubes is needed. In the following analysis, a capacitance model is developed to simulate collector performance. The particular collector used for this analysis is the common tube over sheet type shown in Figure 6.

An expression for the fluid temperature distribution can be derived from a consideration of an element of the absorber plate and fluid. If the capacitance of the fluid, tube and plate are lumped into a single value, an energy balance on a tube element (refer to Figure 7) yields:

\[ \text{Energy in} = \text{Energy out} + \text{Energy stored} \]
Figure 6. Tube Over Sheet Construction
Figure 7. Energy Balance on Tube Element
III.1 \[ F'\text{SW} \, dx = F'U_L (T - T_a)W \, dx + mc_a \frac{\partial T}{\partial t} \, dx + mc_a \frac{\partial T}{\partial t} \, dx \]

where

- \( F' \) = collector efficiency factor
- \( S \) = radiation absorbed on collector plate
- \( W \) = average fin length associated with each tube
- \( U_L \) = loss coefficient from plate to its surroundings
- \( m \) = mass of absorber plate per unit length
- \( \dot{m} \) = circulating fluid flow rate
- \( c_a \) = lumped capacitance value of plate + tube + fluid
- \( x \) = length in flow direction
- \( c_p \) = specific heat of fluid
- \( T_a \) = ambient temperature
- \( T \) = local fluid temperature

Letting \( W = \frac{\text{collector area}}{\text{total exposed tube length}} = \frac{A_c}{L} \)

then \( W \) becomes an averaged value of fin length associated with the tube. We can rearrange the lumped capacity value, \( mc_a \) as

\[ mc_a = \frac{\text{total absorber mass}}{L} \cdot \frac{A_c C_a}{\text{total absorber mass}} \]

or

\[ mc_a = WC_a \]

where \( C_a \) = lumped capacitance of absorber and fluid per unit area
Rewriting and condensing equation III.1 gives the equation reported by Klein:\textsuperscript{13}

\[ \frac{mc}{\rho} \frac{\partial T}{\partial x} + WC \frac{\partial T}{\partial t} = F'W \left[ S - U_L(T - T_a) \right] \]

The collector efficiency factor, $F'$, represents the heat transfer resistance from the collector plate to ambient air. The development of this factor can be found in the literature. Bliss has calculated several efficiency factors for flat plate collectors.\textsuperscript{14} For the tube over sheet case

\[ F' = \frac{1}{\frac{BU_L}{\pi d h_f} + \frac{d_0}{B} \left( \frac{1}{U_c} + \frac{B}{(B-d_0)F} \right)} \]

where $B$, $d$, $d_0$ are shown in Figure 8

$h_f = \text{film coefficient on inside of tube}$

$F = \text{fin efficiency factor defined by}$

\[ F = \frac{\tanh \left[ \left( \frac{U_L}{K \delta} \right)^{1/2} \frac{B - d_0}{2} \right]}{\left[ \left( \frac{U_L}{K \delta} \right)^{1/2} \frac{B - d_0}{2} \right]} \]

where $K = \text{conductivity of collector sheet}$

$\delta = \text{thickness of collector sheet}$
It should be noted that this efficiency factor is based on a collector having parallel tubes with unidirectional flow. In this report, the collector is assumed to have a single tube of the sinusoidal or serpentine type. Abdel-Khalik\(^{15}\) has investigated heat removal factors for serpentine tube collectors. The heat removal factor, \(F_R\), is a quantity relating actual useful energy gain of a collector to the useful gain if the whole collector were at the fluid inlet temperature. Although Abdel-Khalik does not deal with the collector efficiency factor, \(F'\), it can be related to \(F_R\) with much difficulty. However, for this report, it is assumed that the collector efficiency factor for the parallel tube collector is equal to that for the serpentine collector.

Equation III.2 can be rewritten as

\[
\text{III.5} \quad a \frac{\partial^2 T}{\partial t^2} + b \frac{\partial T}{\partial x} + cT = g
\]

where

\[
\begin{align*}
a & = WC_a \\
b & = mc_p \\
c & = F'WU_L \\
g & = F'WS + F'WU_LT_a
\end{align*}
\]

Equation III.5 is a first order non-homogeneous partial differential equation with constant coefficients. The general solution of equation III.5, as developed in Appendix A, is
III.6 \[ T(x,t) = e^{\frac{-c}{a}t} \phi(ax - bt) + \frac{g}{c} \]

To determine the arbitrary function \( \phi \), an initial or boundary condition must be specified. A reasonable initial condition is

\[ T(x,t = 0) = T_{old}(x) \]

where \( T(x,t = 0) \) = the temperature distribution for the beginning of the next time interval

\( T_{old}(x) \) = the temperature distribution at the end of the previous time interval

Substitution of this condition into equation III.6 (refer to Appendix A) yields

III.7 \[ T(x,t) = e^{\frac{-c}{a}t} \left[ T_{old}(x - \frac{b}{a}t) - \frac{g}{c} \right] + \frac{g}{c} \]

for \( 0 \leq t \leq \Delta t \)

Replacing the constants with the appropriate physical parameters gives

III.8 \[ T(x,t) = e^{\frac{-F'U_L}{C_a}t} \left[ T_{old}(x - \frac{mC_P}{W_C a}t) - \frac{S}{U_L} - T_a \right] + \frac{S}{U_L} + T_a \]

for \( 0 \leq t \leq \Delta t \)
This solution assumes constant coefficients. For a given collector, \( W, C_a, \) and \( c_p, \) will be constant. However, over an extended period of time, a real thermosyphon system collector would experience variations in ambient temperature \( (T_a), \) absorbed radiation \( (S), \) loss coefficient \( (U_L), \) collector efficiency \( (F'), \) and mass flow rate \( (\dot{m}). \) This being the case, equation III.8 can be considered valid only over a relatively short time period \( \Delta t \) during which these coefficients may be treated as constants. At the end of that time period, new values of the coefficients are applied and the solution proceeds in a stepwise fashion. The new values required for \( T_a \) and \( S \) come from input data. The mass flow rate is determined from the overall system model (see Chapter V). This leaves the calculation of loss coefficient and collector efficiency factor. To obtain these two quantities in a manner consistent with the transient collector temperature distribution solution, a transient four node collector model is developed in the following paragraphs. The model yields a set of differential equations which can be solved for the various node temperatures. These temperatures lead to the loss coefficient which, in turn, leads to the collector efficiency factor.

The thermal capacitance of a single cover flat plate collector is assumed to be lumped into four nodes as shown in Figure 6. An energy balance can be written for each node. For the cover plate (node 1) an energy balance yields (refer to Figure 9)
Figure 9. Energy Balance on Cover Plate
III.9 \[ Q_{s1} = Q_g + Q_{r21} + Q_{c21} - Q_{rla} - Q_{cla} \]

where

- \( Q_g \) = radiation absorbed by the cover plate
- \( Q_{r21} \) = net radiated energy from absorber plate to cover
- \( Q_{c21} \) = convected energy from absorber plate to cover
- \( Q_{rla} \) = radiated energy from cover to ambient
- \( Q_{cla} \) = convected energy from cover to ambient

The radiation absorbed by the cover plate \( Q_g \) is

III.10 \[ Q_g = A_1 S_g \]

where

- \( S_g \) = energy per unit area absorbed by cover plate from Chapter II
- \( A_1 \) = surface area of node 1

From basic heat transfer considerations

III.11 \[ Q_{rla} = A_1 h_{rla} (T_1 - T_a) \]

III.12 \[ h_{rla} = \varepsilon_1 \sigma (T_1^2 + T_a^2)(T_1 + T_a) \]

where

- \( \varepsilon_1 \) = emissivity of node 1
- \( \sigma \) = Boltzmann's constant
- \( T_1 \) = temperature of node 1
\[ T_a = \text{ambient temperature} \]

Similarly,

\[ Q_{r21} = A_{r21} h_{r21} (T_2 - T_1) \]

\[ h_{r21} = \frac{\sigma F_{2-1} (T_2^2 + T_1^2)(T_2 + T_1)}{\left(\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_1} - 1\right)} \]

where \( \varepsilon_2 \) = emissivity of node 2

\( T_2 \) = temperature of node 2

\( F_{2-1} \) = configuration factor from node 2 to node 1

If it is assumed that the cover plate and absorber plate are infinite planes, the configuration factor is one. Otherwise, \( F_{2-1} \) can be calculated from reference 16.

The convection term can be written as

\[ Q_{c_{1a}} = A_{c_{1a}} h_w (T_1 - T_a) \]

where the wind convection coefficient \( h_w \) is given by

\[ h_w = 1.0 + 0.3 \times \text{(windspeed in mph)} \]

Similarly,

\[ Q_{c_{21}} = A_{c_{21}} h_{c_{21}} (T_2 - T_1) \]

where the convection coefficient \( h_{c_{21}} \) between node 2 and node 1 is
III.18 \[ h_{c21} = C(T_2 - T_1)^{1/4} \]

where \( C \) = coefficient dependent on slope of collector

The energy stored in node 1 is

III.19 \[ Q_{s1} = (mc)_1 \frac{dT_1}{dt} \]

where \( (mc)_1 \) = capacitance of node 1

Rewriting equation III.9

III.20 \[ (mc)_1 \frac{dT_1}{dt} = A_1 g + A_2 h_{r21}(T_2 - T_1) + A_1 h_{c21}(T_2 - T_1) - A_1 h_{rla}(T_1 - T_a) - A_1 h_{w1}(T_1 - T_a) \]

Assuming all nodes will have a nominal common surface area \( A_c \), equation III.20 can be written as

III.21 \[ \frac{dT_1}{dt} = - \frac{A_c}{(mc)_1} (h_{r21} + h_{c21} + h_{rla} + h_w)T_1 + \frac{A_c}{(mc)_1} (h_{r21} + h_{c21})T_2 + \frac{A_c}{(mc)_1} (S_g + h_{rla}T_a + h_wT_a) \]

Similar analyses which can be carried out on the remaining three nodes are presented in detail in Appendix B. For convenience, only the results are presented here. For node 2 (absorber plate, tubes and fluid),

III.22 \[ \frac{dT_2}{dt} = \frac{A_c}{(mc)_2} (h_{c21} + h_{r2})T_1 - \frac{A_c}{(mc)_2} (h_{c21} + h_{r2} + \frac{2K_3}{\Delta x_3})T_2 \]
\[
+ \frac{A_c}{(mc)_2} \left( \frac{2K_3}{\Delta x_3} \right) T_3 + \frac{A_c}{(mc)_2} \left( S + \frac{mc_p}{A_c} T_{in} - \frac{mc_p}{A_c} T_{out} \right)
\]

where

\[ h_{r2} = \frac{h_{r21}}{F_{2-1}} \] (refer to equation III.14)

\[(mc)_2 = \text{capacitance of node 2}\]

\[ K_3 = \text{conductivity of node 3 (back insulation)}\]

\[ \Delta x_3 = \text{thickness of node 3}\]

\[ T_3 = \text{temperature of node 3}\]

\[ \dot{m} = \text{mass flow}\]

\[ c_p = \text{specific heat of fluid}\]

\[ T_{in} = \text{collector inlet temperature}\]

\[ T_{out} = \text{collector outlet temperature}\]

\[ S = \text{absorbed solar energy (refer to Chapter II)}\]

For node 3, the back insulation,

\[
\text{III.23} \quad \frac{dT_3}{dt} = \frac{A_c}{(mc)_3} \left( \frac{2K_3}{\Delta x_3} \right) T_2 - \frac{A_c}{(mc)_3} \left( \frac{4K_3}{\Delta x_3} \right) T_3 + \frac{A_c}{(mc)_3} \left( \frac{2K_3}{\Delta x_3} \right) T_4
\]

where

\[ T_4 = \text{temperature of node 4}\]

\[(mc)_3 = \text{capacitance of node 3}\]

For node 4, the pan or container,

\[
\text{III.24} \quad \frac{dT_4}{dt} = \frac{2A_c K_3}{\Delta x_3 (mc)_4} T_3 - \frac{A_c}{(mc)_4} \left( \frac{2K_3}{\Delta x_3} \right) T_3 + \frac{A_c}{(mc)_4} \left( \frac{2K_3}{\Delta x_3} + h_{r4a} + h_w \right) T_4 + \]

\[ \frac{A_c}{(mc)_4} \left( h_{r4a} + h_w \right) T_a \]
where the radiation coefficient $h_{r4a}$ is

\[ h_{r4a} = \varepsilon_4 \sigma (T_4^2 + T_a^2) (T_4 + T_a) \]

Equations III.21, III.22, III.23 and III.24 constitute a set of differential equations. Rewriting them as a condensed set of equations:

\[
\frac{dT_1}{dt} = a_{11}T_1 + a_{12}T_2 + b_1 \\
\frac{dT_2}{dt} = a_{21}T_1 + a_{22}T_2 + a_{23}T_3 + b_2 \\
\frac{dT_3}{dt} = a_{32}T_2 + a_{33}T_3 + a_{34}T_4 \\
\frac{dT_4}{dt} = a_{43}T_3 + a_{44}T_4 + b_4
\]

In matrix form

\[
\begin{bmatrix}
\cdot \\
T_1 \\
\cdot \\
T_2 \\
\cdot \\
T_3 \\
\cdot \\
T_4
\end{bmatrix} = \begin{bmatrix}
a_{11} & a_{12} & 0 & 0 \\
a_{21} & a_{22} & a_{23} & 0 \\
0 & a_{32} & a_{33} & a_{34} \\
0 & 0 & a_{43} & a_{44}
\end{bmatrix} \begin{bmatrix}
T_1 \\
T_2 \\
T_3 \\
T_4
\end{bmatrix} + \begin{bmatrix}
b_1 \\
b_2 \\
0 \\
b_4
\end{bmatrix}
\]

If these equations can be solved for the node temperatures, the energy lost from the absorber plate can be calculated and thus
a value for the collector loss coefficient, $U_L$, can be assigned.

There are several solution methods available. For these particular equations, stability can be a problem if an explicit solution is used. Therefore, an implicit solution such as the Crank-Nicholson technique is suggested. If more accuracy in the solution is desired and computer time is not a major concern, a matrix solution can be used. Both of the above solution techniques are presented in Appendix C.

Having these node temperatures in hand for a given time interval, a corresponding loss coefficient may be developed. Consider the energy leaving the absorber plate. The total loss consists of losses through the top, edge and back:

$$Q_{\text{LOSS}} = Q_{\text{TOP}} + Q_{\text{EDGE}} + Q_{\text{BACK}}$$

The top loss consists of radiative and convective terms which have been developed previously.

$$Q_{\text{TOP}} = A \frac{h_{\text{cl}2}}{c} (T_2 - T_1) + A \frac{h_{\text{r}2}}{c} (T_2 - T_1)$$

The only back loss term is a conductive one:

$$Q_{\text{BACK}} = \frac{K_A c}{(\Delta x_3/2)} (T_2 - T_3)$$

The following empirical relation for edge loss can be employed:\textsuperscript{19}
III.30 \[ Q_{\text{EDGE}} = (0.08) \text{ (perimeter)} \text{(depth)} (T_2 - T_a) \]

The loss coefficient used in equation III.2 may be expressed in terms of the heat losses as

\[
\text{III.31 } U_L = \frac{Q_{\text{TOP}} + Q_{\text{BACK}} + Q_{\text{EDGE}}}{A_c (T_2 - T_a)}
\]

Having the loss coefficient for one interval, the collector efficiency factor can be calculated from equation III.3. All quantities are now known for the evaluation of the local fluid temperature at any point in the collector (equation III.8).

In summary, the primary result to be obtained from this model is the temperature distribution through the collector. However, this model has also yielded other quantities which may be useful in future comparisons with data presented in the literature, such as:

1. a loss coefficient for the collector
2. mean temperature for the absorber plate \((T_2)\)
IV

TANK MODEL

In addition to the solar collector, the storage tank constitutes a major portion of the thermosyphon system. In choosing a model to simulate tank performance, it must be recalled that the solution to the mass flow problem depends on the temperature distribution around the flow loop. Therefore, as was the case with the collector, a model yielding a temperature distribution through the storage tank is needed. In the following analysis, a capacitance model is developed to simulate tank performance. For generality, the model will allow for fluid drawoff.

If the storage tank is divided into a number of sections and the temperature of each section is determined, a suitable temperature distribution will have been obtained. In order to perform such an analysis, some simplifying assumptions will be made:

1. each section of the storage tank is well mixed
2. there are no heat exchangers or boosters in the tank
3. there are no controllers in the system to limit flow
4. fluid enters and leaves the tank as shown in Figure 10

Consider an arbitrary section of the stratified tank shown in Figure 11. The following energy balance can be written:
Figure 10. Stratified Storage Tank
Figure 11. Energy Balance on $i^{th}$ Section of Storage Tank
\[ IV.1 \quad (mc)_i \frac{d}{dt} (T_{S_i}) = \dot{m}_c (T_{S_{i-1}} - T_{S_i}) - \dot{m}_L c (T_{S_i} - T_{S_{i+1}}) - U_T A_i (T_{S_i} - T_a) \quad i = 2, n-1 \]

where
- \( t \) = time
- \( T_{S_i} \) = temperature of section \( i \) of storage tank
- \( (mc)_i \) = capacitance of section \( i \)
- \( \dot{m} \) = flow rate of circulating fluid
- \( c_p \) = specific heat of fluid
- \( \dot{m}_L \) = flow rate of fluid removed (drawoff)
- \( U_T \) = tank loss coefficient
- \( A_i \) = circumferential area of section \( i \)
- \( T_a \) = ambient temperature
- \( n \) = number of sections desired

For \( i \) equal to 1 (top section), equation IV.1 applies, but \( T_{S_{i-1}} \) is replaced by the collector outlet temperature. For \( i \) equal to \( n \) (bottom section), equation IV.1 applies, but \( T_{S_{i+1}} \) is replaced by the temperature of the makeup fluid.

To obtain a value for the tank loss coefficient, one can employ a steady state conduction analysis. By treating the tank as a simple insulated cylinder and neglecting the effect of convective films on the insulation,

\[ IV.2 \quad U_T = \frac{K_t}{\ln \left( \frac{r_t + \Delta x}{r_t} \right)} \]
where \( K_t \) = conductivity of tank insulation
\( r_t \) = tank radius
\( \Delta x \) = thickness of insulation

After writing \( n \) equations, a set of differential equations similar in form to the collector node equations (chapter III) is obtained. Several methods of solving this set of equations are given in Appendix C. If a value for the flow rate, \( m \), can be assigned, these tank node temperatures can be found for any given time interval. In the following chapter, the problem of assigning a value for the flow rate will be addressed.
SYSTEM FLOW RATE

One of the problems in analyzing a thermosyphon system is the unknown flow rate of the circulating fluid. This problem will be addressed in this chapter.

Up to now, we have assumed the flow rate is known for a given time interval. Based on this assumed flow, expressions for a corresponding temperature distribution were determined for the storage tank and the collector. If some simplifying assumptions are made, one can use the results of Chapters III and IV to completely define the temperature around the flow loop. That is

1. the pipes connecting the collector and tank are well insulated
2. conduction effects along the pipe at the collector inlet and outlet are negligible
3. there is no back-syphoning or reverse flow

Under these assumptions, the temperature profile through the connecting pipes is a constant. Once flow has begun, the fluid temperature in the lower pipe (refer to Figure 1) becomes that of the tank bottom. Similarly, the fluid temperature in the upper pipe is that of the collector outlet. Now that the temperature
around the loop is specified, an analysis yielding the flow rate can be developed.

The temperature distribution around the flow loop has the general shape shown in Figure 12. Since density, in general, decreases with temperature, a corresponding density or specific gravity distribution would appear as shown in Figure 13. It is these density variations which induce thermosyphon flow or so called natural circulation. The pressure head which causes flow, herein referred to as the thermosyphon head, is proportional to the area inside the specific gravity versus position diagram. In particular,

\[ V.1 \quad H_T = \int \gamma \, dy \]

where \( H_T \) = thermosyphon head
\( \gamma \) = specific gravity
\( y \) = vertical position in system

Expanding equation V.1 into sums, the integral becomes

\[ V.2 \quad H_T = \sum_{i=1}^{j} \gamma_i y_i \text{TANK} + \sum_{i=1}^{k} \gamma_i y_i \text{INLET} - \sum_{i=1}^{l} \gamma_i y_i \text{COLLECTOR} - \sum_{i=1}^{m} \gamma_i y_i \text{OUTLET} \]

where \( \gamma_i \) = average specific gravity of \( i^{th} \) section
\( y_i \) = vertical distance of \( i^{th} \) section
Figure 12. Temperature Distribution as a Function of Position in the System
Figure 13. Specific Gravity as a Function of Position in the System
j, k, l, m = number of sections into which the tank, inlet, collector and outlet are divided respectively.

Obviously, as j, k, l, and m approach infinity, equation V.2 becomes exactly equal to the thermosyphon head.

Since the temperature distribution in the pipes is assumed to be constant, it is not necessary to break them up into several sections. This is not the case for the storage tank and the collector where the temperature (and thus the specific gravity) is not a constant.

To evaluate equation V.2, the specific gravity is needed for each section around the flow loop. A temperature distribution has already been developed for the loop. Therefore, if the specific gravity is written as a function of temperature, the specific gravity at each section can be evaluated. An estimate of specific gravity is

\[ \gamma = A T^n + B T^{n-1} + \ldots + Z \]

where \( A, B, C, \ldots \) = constants

For instance, if water is used as the circulating fluid, a second order polynomial representation yields

\[ \gamma = A T^2 + B T + C \]

where

\[ A = -1.116 \times 10^{-6} \]
\[ B = 1.057 \times 10^{-3} \]
\[ C = .7515 \]
T is in degrees Rankine

Substituting equation V.3 into V.2

\[ H_T = \left[ \sum_{i=1}^{i} (AT_i^n + BT_i^{n-1} + \ldots + Z) \Delta y_i \right]_{\text{tank}} + \]

\[ (AT_{\text{inlet}}^n + BT_{\text{inlet}}^{n-1} + \ldots + Z) \cdot (\Delta y)_{\text{inlet}} \]

\[ - \left[ \sum_{i=1}^{i} (AT_i^n + BT_i^{n-1} + \ldots + Z) (\Delta y_i) \right]_{\text{collector}} - \]

\[ (AT_{\text{outlet}}^n + BT_{\text{outlet}}^{n-1} + \ldots + Z) (\Delta y)_{\text{outlet}} \]

Assuming the flow rate to be quasi-steady, the thermosyphon head must equal the friction head which resists flow. That is,

\[ V.6 \quad H_T = H_f \]

where \( H_f = \) total friction head of flow loop

The total friction head is the sum of the individual friction heads in each section or length of pipe in the flow loop. Following Close, the Darcy Weisbach equation for the friction head in the \( i^{th} \) arbitrary section is

\[ V.7 \quad h_i = \frac{f_i}{2} \frac{l_i^2 u_i^2}{2g} + \frac{k_i}{2} \frac{l_i^2 u_i^2}{2g} \]

where \( f_i = \) friction factor in \( i^{th} \) section
\( l_i = \) length of \( i^{th} \) section
\[ u_i = \text{flow velocity in } i^{th} \text{ section} \]
\[ g = \text{gravitational constant} \]
\[ d = \text{inside tube diameter} \]
\[ k_i = \text{loss coefficient for bends, tees, etc., for } i^{th} \text{ section} \]

The term \( \frac{k_i u_i^2}{2g} \) takes into account losses due to bends, tees, valves, and the like which are in the flow circuit. It is zero for straight sections.

Assuming thermosyphon flow is laminar, the friction factor is given as

\[ V.8 \quad f = \frac{64}{Re} = \frac{64v}{ud} \]

where \( Re = \) Reynolds number
\[ \nu = \text{kinematic viscosity} \]

Substituting V.8 into V.7

\[ V.9 \quad h_i = \frac{32\nu_i u_i}{gd^2} + \frac{k_i u_i^2}{2g} \]

The flow rate is expressed as

\[ V.10 \quad \dot{m} = \rho_i A_i u_i \]

where \( \rho_i = \text{density of fluid in } i^{th} \text{ section} \)
\[ A_i = \text{cross sectional area of pipe} \]
\[ u_i = \text{velocity of fluid in } i^{th} \text{ section} \]
The cross sectional area of the pipe is just

\[ V.11 \quad A_i = \frac{\pi}{4} \, d_i^2 \]

Equation V.11 becomes

\[ V.12 \quad \dot{m} = \frac{\rho_i u_i \pi d_i}{4} \]

Solving for velocity \( u_i \)

\[ V.13 \quad u_i = \frac{4 \dot{m}}{\rho_i \pi d_i} \]

Substituting V.13 into V.9 yields

\[ V.14 \quad h_i = \frac{128 V.11 \dot{m}^i}{\rho_i \pi gd_i^4} + \frac{8 k_i \dot{m}^2}{\rho_i^2 \pi^2 gd_i^4} \]

The total friction head of the flow circuit is then

\[ V.15 \quad H_f = \sum_{i=1}^{n} h_i = \sum_{i=1}^{n} \left\{ \frac{128 V.11 \dot{m}^i}{\rho_i \pi gd_i^4} + \frac{8 k_i \dot{m}^2}{\rho_i^2 \pi^2 gd_i^4} \right\} \]

where \( n \) = total number of sections of pipe in the circuit

The summation does not include the tank sections because there is negligible friction head in the tank.

Assuming the pipe diameter remains a constant around the flow circuit, equation V.15 can be written as

\[ V.16 \quad H_f = \frac{128 \dot{m}}{\pi gd^4} \left[ \sum_{i=1}^{n} \frac{V.11 \dot{m}^i}{\rho_i} \right] + \frac{8 \dot{m}^2}{\pi gd^4} \left[ \sum_{i=1}^{n} \frac{k_i}{\rho_i^2} \right] \]
Recall that the thermosyphon head, $H_T$, is equal to the friction head, $H_f$, so that

$$V.17 \quad \frac{8m^2}{\pi gd^4} \sum_{i=1}^{n} \frac{k_i}{\rho_i^2} + \frac{128m}{\pi gd^4} \sum_{i=1}^{n} \frac{v_{ii}}{\rho_i} = H_T$$

This expression can be put in the form,

$$V.18 \quad \cdot2 + bm + c = 0$$

where

$$a = \frac{8}{\pi^2 gd^4} \sum_{i=1}^{n} \frac{k_i}{\rho_i^2}$$

$$b = \frac{128}{\pi gd^4} \sum_{i=1}^{n} \frac{v_{ii}}{\rho_i}$$

$$c = -H_T$$

The constants $a$ and $b$ contain the density $\rho_i$ for each section. This can be converted to specific gravity for each section $\gamma_i$ by

$$V.19 \quad \rho_i = \gamma_i \rho_s$$

where $\rho_s = \text{standard value of density}$

so that

$$a = \frac{8}{\pi^2 gd^4 \rho_s^2} \sum_{i=1}^{n} \frac{k_i}{\gamma_i^2}$$

$$b = \frac{128}{\pi gd^4 \rho_s} \sum_{i=1}^{n} \frac{v_{ii}}{\gamma_i}$$
The b term contains a viscosity term for each section. Since temperature is known for each section, it is most convenient to express viscosity as a function of temperature just as was done previously with specific gravity. For example, if water is the circulating fluid, the following representation yields accurate values for the temperature ranges of interest:\(^*\)

\[
v = 0.00672 \cdot 10^x \frac{1 \text{lbm}}{\text{ft-sec}}
\]

where 
\[
x = \frac{1301}{998.333 + 8.1855(T-20) + 0.00585(T-20)}^2 - 3.30233
\]

for \(0^\circ \leq T \leq 20^\circ \text{C}\)

and 
\[
v = (0.00672)(1.002)10^y \frac{1 \text{lbm}}{\text{ft-sec}}
\]

where 
\[
y = \frac{1.3272(20-T) - 0.001053(T-20)^2}{T + 105}
\]

for \(20^\circ \leq T \leq 100^\circ \text{C}\)

Solving equation V.18 for the flow rate,

\[
\dot{m} = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}
\]

It should be noted that a and b are always positive and c is negative (assuming no reverse flow). Therefore, the quantity under the radical is positive. The two solutions for \(\dot{m}\) are both real.
with one positive and the other negative. The positive solution is chosen to be the desired mass flow rate. Thus,

\[
\dot{m} = \frac{-b + \sqrt{b^2 - 4ac}}{2a}
\]

V.23

It is difficult to give the negative solution a physical interpretation. It does not represent an existing reverse flow condition because the thermosyphon head, \( H_T \), is positive. If the thermosyphon head was negative, a reverse flow could theoretically exist. However, as this is not the case, no interpretation is given the negative solution.

To summarize, the flow rate is dependent on the temperature distribution around the flow circuit. One can relate the specific gravity to this distribution and integrate this specific gravity around the flow loop to obtain the thermosyphon head. Then, using the Darcy Weisbach relation, the friction head is related to the flow rate. Setting the friction head equal to the thermosyphon head, a solution for the flow rate is obtained. Obviously, this must be done in an iterative fashion since neither the flow rate nor the temperature distribution is known initially.
The constituents of a thermosyphon system are the solar collector, the storage tank and the connecting piping. The behavior of each component has been described in terms of a set of equations in Chapters III through V. As a result of the complexity and interdependence of these equations, an iterative solution technique using the digital computer is necessary to determine numerical results.

As discussed in previous chapters, the circulating fluid flow rate must be known before calculating the temperature distribution through the system. This distribution, on the other hand, determines the flow rate. After estimating a flow rate, a distribution can be generated. This leads to a better estimate of the flow rate and the process can be repeated. This is the procedure incorporated by the computer model, the TSP.

Functionally, the TSP may be broken down into three basic components. The first of these, calculation of the absorbed solar energy, is independent of the others. The other two components, namely the system temperature distribution and flow rate, are calculated iteratively as discussed earlier. A simplified flow chart
The TSP requires an extensive list of input variables describing the system and the environmental operating conditions. One such input is the experimental radiation values for the tilted collector surface. Under the assumptions set forth in Chapter II, this form of incident energy input eliminates the need for calculating incident angles associated with obtaining incident beam energy normal to the collector. Having the normal radiation data, the energy absorbed by the cover plate and the absorber plate can be calculated directly.

To establish the temperature distribution through the system at the end of some time interval \( t = n\Delta t \), the initial \( t=0 \) temperature distribution and the mean flow rate for the interval must be known. By using this previous flow rate as an estimate for the new flow rate, a final temperature distribution can be generated. This distribution is used to calculate a new flow rate value which, in turn, generates a new distribution. This continues until convergence when the calculated flow rate changes little from one solution to the next.

The determination of a temperature distribution through the system can be broken down into three basic parts, the calculation of the distribution through the collector, storage tank, and piping. The behavior of the collector is described as a set of four ordinary differential equations and a partial differential equa-
Figure 14. Simplified Flowchart of TSP
tion. The partial differential equation requires values for collector efficiency and loss coefficient which can only be assigned after solving the four ordinary differential equations.

Appendix C describes methods for solving the set of differential equations. To simplify the solution, the TSP approximates the coefficients in the set of ordinary differential equations as constants for any one time interval. From experience, it was found that the simpler predictor-corrector, an explicit numerical solution, was not stable for the collector node equations. However, the Crank-Nicholson technique, an implicit solution, was found to be in agreement with the exact solution. Although the exact solution requires several times the computation time needed by the Crank-Nicholson method, both are available in the TSP. The exact solution is composed of an infinite sum as described in Appendix C. This infinite sum must be truncated for use on a digital computer and this truncation is specified by the TSP user. It should be noted, however, that larger time steps require more terms in the sum to guarantee convergence to the correct solution.

Having solved the set of ordinary differential equations, the partial differential equation can be solved directly to yield the temperature distribution through the collector. The solution to this partial differential equation is described in Appendix A.

The behavior of the storage tank has been described as a set of n ordinary differential equations in Chapter IV. For ease of programming as well as solution accuracy, the tank was chosen to
have four sections \((n = 4)\). The total capacitance associated with each of these equations is usually much larger than that associated with the collector's differential equations. As a result, the tank equations are much less affected by sudden changes such as load drawoff allowing the predictor-corrector method to remain stable. Therefore, in an effort to save computation time, the Adams-Moulton predictor-corrector technique is used to obtain a solution. The solution to the tank equations yields directly a temperature distribution through the storage tank.

In its present configuration, the TSP neglects pipe losses. Therefore, the temperature distribution in the piping is considered to be a constant. That is, the inlet pipe contains fluid at the tank bottom temperature, and the outlet pipe contains fluid at the collector outlet temperature.

The determination of the circulating fluid flow rate is straightforward after obtaining a temperature distribution. A numerical integration of the specific gravity around the flow loop yields the thermosyphon head. Setting this head equal to the friction head through the loop results in a solution for the flow rate.

A more complete understanding of the computer program may be obtained from the detailed flowchart presented in Appendix E.

The thermosyphon predictor program is intended to estimate performance characteristics and aid in design procedures. For this purpose, the output of the TSP contains the following parameters for the end of each time step:
1. Time
2. Flow rate through the collector ($\dot{m}$)
3. Load flow removed from the storage tank ($\dot{m}_L$)
4. Inlet temperature to the collector
5. Outlet temperature from the collector
6. Storage tank temperatures
7. Mean tank temperature
8. Insolation on the collector surface ($I_o$)

In addition to the above, the output contains the following parameters averaged over each time step:

1. Temperature rise across the collector ($\Delta T_{avg}$)
2. Useful energy gain from the collector ($Q_u$)
3. Collector efficiency ($\eta$)

Daily totals for the insolation, useful energy gain, collector efficiency, $\eta_{day}$, and a daily system efficiency, $\eta_s$, is also given.

The useful energy gain from the collector is described as

$$Q_u = \dot{m}_a c_p \Delta T_{avg} \Delta \tau$$

where

$\dot{m}_a =$ flow rate through the collector averaged over the time interval

$c_p =$ specific heat of fluid

$\Delta T_{avg} =$ average temperature rise across the collector

$\Delta \tau =$ time step
The collector efficiency, \( \eta \), is simply the useful gain for the period, \( Q_u \), divided by the insolation, \( I_o \) over the period.

\[
\eta = \frac{Q_u}{I_o}
\]

Daily totals for useful energy gain, \((Q_u)_{day}\) and insolation, \((I_o)_{day}\) can be obtained by summing the individual time interval totals. The daily collector efficiency is then

\[
\eta_{day} = \frac{(Q_u)_{day}}{(I_o)_{day}}
\]

A daily system efficiency is described as

\[
\eta_s = \frac{\Delta E_{TANK} + E_{LOAD}}{(I_o)_{day}}
\]

where

\[
\Delta E_{TANK} = \text{energy increase of the storage tank over the day}
\]

\[
E_{LOAD} = \text{energy supplied by the storage tank to the load}
\]

The energy increase of the storage tank, \( \Delta E_{TANK} \), is

\[
\Delta E_{TANK} = M_T C_p \Delta T_{TANK}
\]

where

\[
M_T = \text{mass of fluid in the storage tank}
\]

\[
C_p = \text{specific heat of fluid}
\]

\[
\Delta T_{TANK} = \text{rise in mean tank temperature for the day}
\]
The energy supplied to the load, $E_{LOAD}$, is

$$ E_{LOAD} = M_L C_P \Delta T_{LOAD} $$

where

- $M_L$ = mass of fluid supplied to the load
- $\Delta T_{LOAD}$ = temperature difference between hot fluid supplied to the load and cold fluid returned to the storage tank

The preceding parameters constitute the output of the TSP. The accuracy of these predicted values will be discussed in the following chapter.
VII

MODEL VALIDATION

The following discussion presents a validation of the thermosyphon model, the TSP. This is accomplished by

1. investigating the face validity of the model
2. comparing the TSP to a model presently accepted in the literature
3. comparing experimental and predicted results

The face validity of the model can be accomplished by establishing the TSP solutions to be reasonable, non-oscillatory, and stable. It was decided that the most direct method to approach this problem was to provide the TSP with hypothetical data and observe the output closely. This test consisted of simulating the experimental system described later in this chapter and in Appendix D. Briefly, the system was composed of an 80 gallon storage tank connected to a collector with approximately 48 square feet of absorber surface.

An ideal eight hour day was simulated. The incident radiation for this day was described by

\[ R = 300 \sin \left( \frac{\pi t}{8} \right) \frac{\text{BTU}}{\text{hr-ft}^2} \quad t = 0-8 \text{ hr} \]
The ambient temperature was described by

\[ T_a = 75 + 15 \sin \left( \frac{\pi t}{8} \right) \quad (^\circ F) \quad t = 0-8 \text{ hr} \]

In addition, there was no fluid removed from the storage tank for this simulation.

Given these ideal conditions, the system operation should be continuous smooth functions. Also, the flow rate and collector outlet temperature could be expected to be sinusoidal in nature following the radiation.

Figures 15-19 present radiation, collector inlet and outlet temperatures, flow rate, collector efficiency, and storage tank temperatures versus time. As these figures indicate, the TSP prediction exhibits no oscillations or instabilities. In addition, the results appear reasonable when compared to the sinusoidal insolation input.

A non-ideal day with cloud cover and load flow is simulated later in this chapter. This simulation is validated by experimental data. However, it should be noted here for the purpose of face validity no instabilities or oscillations were exhibited in the TSP predictions.

A second method of verifying a model is comparing it to other previously verified models. A thermosyphon system model presented by D. J. Close has become widely accepted. Close’s model assumes the ideal cases of no drawoff from the tank and the sun is not obscured by cloud or haze. Briefly, the model consists of two differential equations. Employing analytical expressions for insola-
Figure 15. Radiation versus Time for Ideal Day
Figure 16. Collector Inlet and Outlet Temperatures versus Time for Ideal Day
Figure 17. Collector Flow Rate versus Time for Ideal Day
Figure 18. Collector Efficiency versus Time for Ideal Day
Figure 19. Storage Tank Temperatures versus Time for Ideal Day
tion and ambient temperature in energy balances for the absorber and tank, Close relates a mean system temperature and time in a single differential equation. The equation is solved to yield the mean system temperature as a function of time. The second differential equation, when solved, yields an expression for the flow rate and mean system temperature as a function of time. To obtain this equation, Close assumes a linear temperature distribution through the tank and absorber. Using a parabolic relationship between temperature and specific gravity, the thermosyphon head, $H_T$, can be expressed as a function of geometry and mean system temperature. An expression for the friction head, $H_f$, is derived from a direct application of the Darcy Weisbach equation. Equating the friction head and the thermosyphon head, an ordinary differential equation involving the flow rate is obtained.

The two differential equations, when solved, are sufficient to describe the absorber inlet and outlet temperatures and the flow rate. The solutions to these equations presented by Close are easily applied to the digital computer for a given thermosyphon system and operating conditions. Consequently, it was decided to use the Close model to simulate the previously described thermosyphon system (refer to Appendix D for detailed description) under the same ideal operation conditions. That is, the radiation is described by

$$ R = 300 \sin \left( \frac{\pi t}{8} \right) \frac{\text{BTU}}{\text{hr-ft}^2} \quad t = 0-8 \text{ hr} $$
and the ambient temperature is described by

\[ T_a = 75 + 15 \sin\left(\frac{\pi t}{8}\right) \quad (^\circ F) \quad t = 0-8 \text{ hr} \]

The results of this simulation are compared to the TSP predictions in Figures 20 and 21. As indicated in Figure 20, the flow rate curves exhibit the same shape and are within 23% of one another at mid-day. The collector inlet and outlet temperatures as predicted by the TSP and the Close model are presented in Figure 21. Except for the initial dip in the inlet temperature present in the Close model, the shapes of the curves are the same. The initial dip is due to the nature of Close's solutions and does not reflect the actual case. In addition, the inlet temperature rises much more rapidly than that predicted by the TSP. This is a result of Close's assumption that the mean tank temperature and mean absorber temperature are equal. As the storage tank size and collector size increase, this assumption becomes less valid. It is believed that this caused the increasing difference between the inlet temperatures predicted by the two models. The outlet temperature predicted by the TSP agrees with the Close model. For the first seven hours, the models are within 20% of one another.

In conclusion, the TSP predicted a much slower rise in inlet temperature than the Close model. However, it is believed that the large storage tank and collector used in this simulation caused a violation of an assumption used in the Close model. This, in turn,
Figure 20. Flow Rate versus Time from TSP and Close model
Figure 21. Collector Inlet and Outlet Temperatures versus Time from Close and TSP.
resulted in the apparent overprediction of the inlet temperature. Nevertheless, the TSP predicted similar curves for the collector outlet temperature and the system flow rate.

The TSP model verification is finalized by a comparison of actual experimental data and predicted values. For this purpose, a thermosyphon unit located at Florida Technological University in Orlando, Florida, was tested under cloudy weather conditions. This test was taken with the mobile STAR (Solar Testing and Recording) unit developed at Florida Technological University under the direction of Bruce Nimmo. The STAR unit has been used to successfully monitor in situ domestic solar hot water systems.

Figure 22 shows the unit tested. Briefly, the unit consisted of an 80 gallon storage tank connected to a collector with an absorbing area of approximately 48 square feet. Further details on the thermosyphon system is presented in Appendix D.

The test setup is presented in Figure 23. Copper-Constantan thermocouples were equally spaced inside the storage tank. All thermocouple signals were referenced and amplified by an Omega Omni-Amp amplifier. The amplifier was calibrated such that the output voltage corresponded directly to the temperature in Centigrade. To determine the thermocouple's non-linearity, all thermocouples and accompanying amplifiers were previously calibrated. As a result of this calibration, the following fifth order temperature correction was derived.
Figure 22. Thermosyphon Unit Tested
Indicates visual reading or recording

T<sub>1</sub> Thermocouple
F<sub>1/2</sub> Flowmeter (collector/drawoff)
P Pyranometer
W<sub>1/2</sub> Wind indicator (speed/direction)

Figure 23. Test Setup
VII.5 \[ T_c = AT^5 + BT^4 + CT^3 + DT^2 + ET + F \]

where 
- \[ T_c \] = corrected temperature (°C)
- \[ T \] = measured temperature (°C)
- \[ A = 8.10586 \times 10^{-10} \]
- \[ B = -2.34205 \times 10^{-7} \]
- \[ C = 2.72754 \times 10^{-5} \]
- \[ D = -2.50217 \times 10^{-3} \]
- \[ E = 1.13378 \]
- \[ F = -1.66236 \times 10^{-2} \]

The uncorrected temperature readings were manually recorded.

Insolation was measured by an Epply pyranometer mounted at an angle of 34 degrees to the horizontal, that is, the same angle of inclination as the absorber. The pyranometer readings were continuously recorded with a Speedomax chart recorder.

One of the major difficulties of monitoring a thermosyphon unit is measuring the slow flow rates. For this test, the flow rate through the collector was obtained with a Barco 220 venturi flowmeter. Even for high thermosyphon flow rates, the pressure differential output from the venturi fell below the calibration curve provided by the manufacturer. Therefore, a calibration was performed for flow rates down to 1.5 gallons per hour.

The pressure differential across the venturi was measured with a Validyne pressure transducer with a one psi diaphragm. The
accompanying Validyne amplifier was adjusted so that a 0.5 psi differential corresponded to a full scale output of 5.0 volts. All voltage readings were recorded manually.

Two nine hour tests on the aforementioned thermosyphon unit were performed under the non-ideal condition of cloudy weather. The raw data for both days is contained in Appendix D. The temperature entries have not been corrected according to equation VII.5.

One of the two tests consisted of drawoff from the storage tank. In order to fully examine its capability, the TSP was used to simulate the thermosyphon system performance for this particular day. Tables 1 and 2 present the condensed data for this test. The temperatures have been corrected according to equation VII.5 and converted to degrees Farenheit. Insolation and load flow for the test are plotted as a function of time in Figures 24 and 25.

Figures 26-30 present measured and predicted values for the mass flow, absorber inlet and outlet temperatures, and mean tank temperature. Although the TSP yields temperatures of the tank at four equally spaced points and the experimental data is taken at three equally spaced points in the tank, both are plotted. This is done to compare trends and test the validity of the model.

As indicated from Figures 26 and 27, the tank is not as well mixed as the TSP predicts. That is, the TSP yields an approximately linear temperature distribution in the storage tank. However, the actual stratification exhibited an un-mixed condition. This
Table 1. Collector Data for September 29, 1976

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<th>$T_{\text{outlet}}$ ($^\circ\text{F}$)</th>
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Figure 24. Average Insolation Rate versus Time of Day for 9/29/76
Figure 25. Load Flow versus Time of Day for 9/29/76

Load Flow (GPM) vs. Time of Day
Figure 26. Measured Storage Tank Temperatures versus Time of Day for 9/29/76
Figure 27. Predicted Storage Tank Temperatures versus Time for 9/29/76
Figure 28. Mean Tank Temperature versus Time of Day for 9/29/76
Figure 29. Collector Flow Rate versus Time of Day for 9/29/76
Figure 30. Collector Inlet and Outlet Temperatures versus Time of Day for 9/29/76
indicates the fluid in the tank to move in "slugs" when not disturbed by load flow, etc. This idea can be supported by comparing information derived from Figures 26 and 29. Knowing the distance between the thermocouples, one can estimate the time it will take the warm fluid to reach each thermocouple by integrating under the flow rate curve. One can then compare this time to the time the thermocouple actually sensed the warmer fluid (refer to Figure 26). For example, the first thermocouple in the tank is located eight inches below the tank top. This distance represents 13 gallons of fluid. An integration under the flow rate curve (Figure 29) indicates that a total of 13 gallons had flowed through the system by 10:30 am. From Figure 26 this is about the same time the first thermocouple senses hot fluid.

The second thermocouple is located 16 inches below the first thermocouple. This distance corresponds to 26 gallons of fluid in the tank. Another integration under the flow rate curve indicates that an additional 26 gallons had passed through the system by 1:10 pm. Therefore, if the fluid did move in "slugs," the middle thermocouple should have sensed the first warm fluid at this time. Figure 26 shows this to be the case.

Under the assumption the fluid moved in slugs, the last thermocouple in the storage tank would not sense warmer fluid until the end of the test. However, Figure 26 indicates a temperature rise at approximately 1:40 pm. This is due to a load imposed on
the tank. This represents a disturbance to the system which would tend to mix the tank. In addition, the return fluid for this particular load removal was much warmer than the fluid in the bottom of the tank.

The TSP appears to predict minimum tank temperatures after each load removal one time interval later than the actual case. This is due solely to the nature of the predictor-corrector solution used to obtain the tank node temperatures.

Regardless of its poor prediction of the actual storage tank stratification, the TSP accurately predicts the mean tank temperature as shown in Figure 28. The average error of the TSP in this case is 1.5 percent. The maximum error (6.5 percent) occurred after intervals involving load flow. Nevertheless, the TSP exhibited the correct shape for these load flow intervals.

Figure 29 presents a comparison between the predicted and measured flow rate. The TSP is in close agreement with the actual values. The error is greatest in the early morning when the flow is getting started. This is due, in part, by the error in the flow monitoring device (pressure transducer) at the extremely low flows of 1.5 gallons per hour and less. Between the hours of 9 am and 5 pm, the average error is 6.4 percent. This error is increased to 9.6 percent if the first hour is included.

The collector inlet and outlet temperatures are presented in Figure 30. As with the system flow rate, the TSP yielded accurate
values. The TSP underpredicted the inlet temperature somewhat. This is attributed, in part, to insufficient insulation of the inlet pipe. A small portion of the pipe was subject to direct radiation and higher ambient temperatures. The largest error (9 percent) occurs after the first load flow test when the actual inlet temperature rose unexpectedly. It was later determined that the return water line had been exposed to direct radiation causing a portion of the return fluid to be at a significantly higher temperature than the tank bottom. Since this return fluid entered the tank well below the lowest thermocouple, the thermocouple on the collector inlet saw a much larger rise in temperature than did the one located in the tank. Nonetheless, the average error between predicted and measured inlet temperature is 3 percent.

The predicted outlet temperatures followed the measured values within 9 percent for the nine hour test. The average error in the TSP prediction is 2 percent. This close agreement is attributed to the rather sophisticated collector model. J. B. Pearce has also successfully employed this model in a simulation of pumped solar hot water systems. He also found the collector model to be accurate for non-ideal days.

In conclusion, the TSP accurately predicted collector inlet and outlet temperatures, system flow rate, and mean tank temperature for a test involving cloudy weather and drawoff from the storage tank. It did not exhibit instability in the solutions and presented a somewhat more realistic simulation than the Close
model. Therefore, the TSP can be considered valid for normal weather conditions if the need for the storage tank temperature stratification is not essential to the user.
VIII

CONCLUSIONS AND RECOMMENDATIONS

The computer model presented in the preceding chapters met its intended goal. That is, it represents an accurate model of a thermosyphon system subject to fluid drawn from the tank and cloudy weather conditions. The model is able to predict the system performance within 10 percent.

Although the mean tank temperature is accurately predictable, the tank stratification is not. Therefore, it is recommended that this be more closely investigated. If possible, a tank model which utilizes a continuous non-linear temperature distribution should be investigated.

The model can be used to evaluate design alternatives for thermosyphon units. The effects of collector geometry (tube size, tube spacing, tube configuration, etc.) can easily be determined as a guide to obtaining maximum performance.

This model can be used to investigate the effectiveness of adding a pump to the system. Obviously, a pumped system is more efficient than a similar thermosyphon system since the latter usually operates at a much higher temperature and thus has a greater energy loss to the environment. However, a pumped system
generally requires power, a controller, and more maintenance. It should be possible with this model and the pumped system model (referred to as PSP) developed by Pearce to determine the cost effectiveness of adding a pump to a solar hot water system.
Appendix A

SOLUTION TO THE FIRST ORDER PARTIAL DIFFERENTIAL EQUATION

The solution to

\[ a \frac{\partial T}{\partial t} + b \frac{\partial T}{\partial x} + c T = g \]

where \( a, b, c, \) and \( g \) are arbitrary constants \(( \neq 0 )\) is the sum of

of the particular solution and the complimentary solution. The

complimentary solution is obtained by solving equation A.1 with \( g \)

set equal to zero:

\[ a \frac{\partial T}{\partial t} + b \frac{\partial T}{\partial x} + c T = 0 \]

Lagrange solved the more general case of the homogenous first

order partial differential equation

\[ f(x,t,T) \frac{\partial T}{\partial t} + g(x,t,T) \frac{\partial T}{\partial x} + h(x,t,T) = 0 \]

by reducing the problem to an equivalent set of ordinary

differential equations (the Lagrange system) of the form

\[ \frac{dt}{f(x,t,T)} = \frac{dt}{g(x,t,T)} = \frac{dt}{-h(x,t,T)} \]
For the case at hand, equation A.2 will have the Lagrange system

\[ \frac{dt}{a} = \frac{dx}{b} = \frac{1}{c} \frac{dT}{T} \]

From equation A.5,

\[ \frac{dt}{a} = \frac{dx}{b} \]

\[ \int_{0}^{x} adx = \int_{0}^{t} bdt + K_1 \]

which yields,

\[ K_1 = ax - bt \]

Again from equation A.5,

\[ \frac{dt}{a} = \frac{1}{c} \frac{dT}{T} \]

\[ -\frac{c}{a} \int_{0}^{t} dt + K_2 = \int_{0}^{t} \frac{dT}{T} \]

yielding

\[ -\frac{c}{a} t + \ln(K_2) = \ln(T) \]

which simplifies to

\[ T = K_3 e^{-\frac{c}{a} t} \]
Equations A.8 and A.12 are combined to give the general solution to A.2 as

\[ T = e^{-\frac{C}{a} t} \left[ \phi (ax - bt) \right] \]

where the function \( \phi (ax - bt) \) is to be determined from initial conditions or boundary conditions. Equation A.13 constitutes the complimentary solution to A.1.

The particular solution can be had rather easily by noting that \( g \) is a constant. Therefore, assuming the particular solution to be a constant \( K_4 \), the total solution becomes

\[ T (x,t) = e^{-\frac{C}{a} t} \left[ \phi (ax - bt) \right] + K_4 \]

Substituting this back into A.1 yields \( K_4 \)

\[ K_4 = \frac{g}{c} \]

so that the total solution to A.1 becomes

\[ T (x,t) = e^{-\frac{C}{a} t} \left[ \phi (ax - bt) \right] + \frac{g}{c} \]

If this equation is to be solved for various time intervals, it would be desirable to choose an initial condition which allows for time continuity of the solution. Such a condition would be

\[ T_{\text{new}} (x,t = 0) = T_{\text{old}} (x,t = \Delta t) \]
where

\[ T_{\text{new}}(x, t = 0) = \text{the temperature distribution for the beginning of the next time interval} \]

\[ T_{\text{old}}(x, t = \Delta t) = \text{the temperature distribution at the end of the last time interval} \]

Substituting this condition into equation A.16 yields

A.17 \[ T_{\text{old}}(x) = \phi(ax) + \frac{q}{c} \]

Solving for the unknown function \( \phi \)

A.18 \[ \phi(ax) = T_{\text{old}}(x) - \frac{q}{c} \]

Let \( z = ax \). Then

A.19 \[ \phi(z) = T_{\text{old}} \left( \frac{z}{a} \right) - \frac{q}{c} \]

Therefore,

A.20 \[ \phi(ax - bt) = T_{\text{old}} \left( \frac{ax - bt}{a} \right) - \frac{q}{c} \]

Then the total solution becomes

A.20 \[ T = e^{-\frac{c}{a}t} \left[ T_{\text{old}} \left( x - \frac{bt}{a} \right) - \frac{q}{c} \right] + \frac{q}{c} \]

\[ 0 \leq t \leq \Delta t \]
Appendix B

COLLECTOR NODE EQUATIONS

The collector model used in this paper consists of four nodes, the cover plate, the absorber and fluid, the back insulation, and the container (refer to Figure 31). An energy balance written on each node will yield a set of differential equations describing mean temperatures of these nodes.

Consider the cover plate, node one, as shown in Figure 32. An energy balance yields

\[ B.1 \quad Q_{s1} = Q_g + Q_{r21} + Q_{c21} - Q_{rla} - Q_{cla} \]

where

- \( Q_{s1} \) = energy stored in cover plate
- \( Q_g \) = radiation absorbed by cover plate
- \( Q_{r21} \) = radiated energy from absorber to cover
- \( Q_{c21} \) = convected energy from absorber to cover
- \( Q_{rla} \) = radiated energy from cover to ambient
- \( Q_{cla} \) = convected energy from cover to ambient

The radiation absorbed by the cover plate \( Q_g \) is

\[ B.2 \quad Q_g = A_1 S_g \]

where \( S_g \) = energy per unit area absorbed by cover plate (refer to Chapter II)
Figure 31. Four Node Collector Model
Figure 32. Energy Balance on Cover Plate
where $A_1 = \text{area of node 1}$

From basic heat transfer considerations,

\[ B.3 \quad Q_{rla} = A_1 h_{rla} (T_1 - T_a) \]

where

\[ h_{rla} = \varepsilon_1 \sigma (T_1^2 + T_a^2) (T_1 + T_a) \]

\[ \varepsilon_1 = \text{emissivity of node 1} \]

\[ \sigma = \text{Boltzman's constant} \]

\[ A_1 = \text{area of node 1} \]

\[ T_1 = \text{temperature of node 1} \]

\[ T_a = \text{ambient temperature} \]

Similarly,

\[ B.4 \quad Q_{r21} = A_2 h_{r21} (T_2 - T_1) \]

where

\[ h_{r21} = \frac{\sigma F_{2-1} (T_2^2 + T_1^2) (T_2 + T_1)}{\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_1} - 1} \]

\[ \varepsilon_2 = \text{emissivity of node 2} \]

\[ T_2 = \text{temperature of node 2} \]

\[ F_{2-1} = \text{configuration from node 2 to node 1} \]

If it is assumed that the cover plate and absorber plate are infinite planes, the configuration factor is one. Otherwise,
$F_{2-1}$ can be calculated from reference 16.

The convection terms can be written as

$$Q_{c_{1a}} = A_1 h_w (T_1 - T_a)$$

where the wind convection coefficient $h_w$ is given by

$$h_w = 1.0 + 0.3 \times \text{windspeed in mph}$$

Similarly,

$$Q_{c_{21}} = A_1 h_{c21} (T_2 - T_1)$$

where the convection coefficient $h_{c21}$ between node 2 and node 1 is given by

$$h_{c21} = C \cdot (T_2 - T_1)^{1/4}$$

where $C$ = coefficient dependent on slope of collector

The energy stored in node 1 is

$$Q_{s1} = (mc)_1 \frac{dT_1}{dt}$$

where $(mc)_1 = \text{capacitance of node 1}$

Rewriting equation B.1,
Assuming all nodes will have a nominal common surface area $A_c$, equation B.10 can be written as

\[
\frac{dT_1}{dt} = \frac{A_c}{(mc)_1} (h_{r21} + h_{c21} + h_{r1a} + h_w) T_1 + \frac{A_c}{(mc)_1} (h_{r21} + h_{c21}) T_2 + \frac{A_c}{(mc)_1} (S_g + h_{r1a} T_a) + h_w T_a
\]

The second node is the absorber plate, tubes and fluid as shown in Figure 33. An energy balance yields

\[
Q_{s2} = Q_p - Q_{c21} - Q_r - Q_{x23} - Q_{\text{REMOVED}}
\]

where

- $Q_{s2}$ = energy stored in node 2
- $Q_p$ = absorbed solar energy
- $Q_{c21}$ = energy convected from absorber to cover
- $Q_r$ = net energy radiated from absorber
Figure 33. Energy Balance on Absorber Plate, Tubes, and Fluid
\[ Q_{K23} = \text{energy conducted from absorber through insulation} \]

\[ Q_{\text{REMOVED}} = \text{energy removed via moving fluid} \]

From equation B.7,

\[ B.13 \quad Q_{c2l} = A c h_{c2l} (T_2 - T_1) \]

Assuming the absorber radiates primarily to the cover, basic heat transfer considerations yield

\[ B.14 \quad Q_r = A c h_{r2} (T_2 - T_1) \]

where the radiation coefficient is

\[ h_{r2} = \frac{\sigma (T_2^2 + T_1^2) (T_2 + T_1)}{\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_1} - 1} \]

For a linear temperature drop through the insulation the energy conducted from the absorber through the insulation is

\[ B.15 \quad Q_{K23} = \frac{K_3 A c}{\Delta x_3} (T_2 - T_3) \left( \frac{\Delta x_3}{2} \right) \]

where

\[ K_3 = \text{conductivity of insulation} \]

\[ \Delta x_3 = \text{thickness of insulation} \]

The energy removed from the absorber via the circulating
fluid is

\[ Q_{\text{REMOVED}} = \dot{m} c_p (T_{\text{OUT}} - T_{\text{IN}}) \]

where
\( \dot{m} = \) flow rate
\( c_p = \) constant specific heat
\( T_{\text{OUT}} = \) outlet temperature of collector
\( T_{\text{IN}} = \) inlet temperature of collector

The absorbed solar radiation is given in Chapter II as

\[ Q_P = A_S \]

The energy stored in the absorber plate is

\[ Q_{s2} = (mc)^2 \frac{dT_2}{dt} \]

Substituting equations B.13 through B.18 into B.12

\[ (mc)^2 \frac{dT_2}{dt} = A_S c - A_{hc2l} (T_2 - T_1) - A_{hc2r} (T_2 - T_1) \]
\[ - \frac{K A}{\Delta x_3} (T_2 - T_3) - \dot{m} c_p (T_{\text{OUT}} - T_{\text{IN}}) \]

This can be condensed to

\[ \frac{dT_2}{dt} = \frac{A_c}{(mc)^2} (h_{c2l} + h_{c2r}) T_1 - \frac{A_c}{(mc)^2} (h_{c2l} + h_{c2r}) \]
\[
\frac{(2K_3)}{\Delta x_3} T_2 + \frac{A_c}{(mc)^2} \left( \frac{2K_3}{\Delta x_3} \right) T_3 + \frac{A_c}{(mc)^2} (S + \frac{mc}{A_2} T_{IN} - \frac{mc}{A_2} T_{OUT})
\]

The back insulation is the third node. An energy balance on node 3 (refer to Figure 34) yields

\[Q_{s3} = Q_{k23} - Q_{k34}\]  \hspace{1cm} (B.21)

where
\[Q_{s3} = \text{energy stored in insulation}\]
\[Q_{k23} = \text{energy conducted into node 3 from absorber}\]
\[Q_{k34} = \text{energy conducted out of node 3 to the container}\]

From equation B.15

\[Q_{k23} = \frac{K_3 A_c}{\Delta x_3} \left( T_2 - T_3 \right) \]  \hspace{1cm} (B.22)

The energy conducted to the container is

\[Q_{k34} = \frac{K_3 A_c}{\Delta x_3} \left( T_3 - T_4 \right) \]  \hspace{1cm} (B.23)

The stored energy is

\[Q_{s3} = (mc)^3 \frac{dT_3}{dt} \]  \hspace{1cm} (B.24)
Figure 34. Energy Balance on Back Insulation
Combining equations B.22, 23, 24

\[ \text{B.25} \quad \frac{dT_3}{dt} = \frac{K_3 A_c}{(mc)_3} \left( T_2 - T_3 \right) - \frac{K_3 A_c}{(\Delta x_3)^2} \left( T_3 - T_4 \right) \]

which can be condensed to

\[ \text{B.26} \quad \frac{dT_3}{dt} = \frac{A_c}{(mc)_3} \left( \frac{2K_3}{\Delta x_3} \right) T_2 - \frac{A_c}{(mc)_3} \left( \frac{4K_3}{\Delta x_3} \right) T_3 + \frac{A_c}{(mc)_3} \left( \frac{2K_3}{\Delta x_3} \right) T_4 \]

The last node to consider is the container or pan. An energy balance on this fourth node (refer to Figure 35)

\[ \text{B.27} \quad Q_{s4} = Q_{k34} - Q_{c4a} - Q_{r4a} \]

where

- \( Q_{s4} \) = energy stored in node 4
- \( Q_{k34} \) = energy conducted from node 3 to 4
- \( Q_{r4a} \) = energy radiated from node 4 to ambient

From equation B.23

\[ \text{B.28} \quad Q_{k34} = \frac{K_3 A_c}{(\Delta x_3)^2} (T_3 - T_4) \]

The energy radiated from node 4 is

\[ \text{B.29} \quad Q_{r4a} = A_r h_{r4a} (T_4 - T_a) \]
Figure 35. Energy Balance on Container
where the radiation coefficient is just

\[ h_{r4a} = \sigma \varepsilon_4 (T_4^2 + \bar{T}_a^2)(T_4 + \bar{T}_a) \]

The energy convected from node 4 is

\[ B.30 \quad Q_{c4a} = A_c w (T_4 - \bar{T}_a) \]

The energy stored in node 4 is

\[ B.31 \quad Q_{s4} = (mc)_4 \frac{dT_4}{dt} \]

Substituting equations B.28 - B.31 into B.27

\[ B.32 \quad (mc)_4 \frac{dT_4}{dt} = \frac{K_3 A_c}{\Delta x_3}(T_3 - T_4) - A_c w (T_4 - \bar{T}_a) - A_c h_{r4a} (T_4 - \bar{T}_a) \]

This can be condensed to

\[ B.33 \quad \frac{dT_4}{dt} = \frac{A_c}{(mc)_4} \left( \frac{2K_3}{\Delta x_3} \right) T_3 - \frac{A_c}{(mc)_4} \left( \frac{2K_3}{\Delta x_3} \right) T_4 + h_w + h_{r4a} \]

\[ \cdot \bar{T}_a + \frac{A_c}{(mc)_4} h_{w4a} \bar{T}_a + h_{r4a} \bar{T}_a \]

Equations B.11, B.20, B.26 and B.33 constitute a set of linear differential equations describing the mean temperatures of the collector's four nodes.
Appendix C

SOLUTION OF A SET OF FOUR
ORDINARY DIFFERENTIAL EQUATIONS

The general set of four simultaneous ordinary linear differential equations which occurs in this paper has the matrix form

\[
\begin{bmatrix}
\dot{T}_1 \\
\dot{T}_2 \\
\dot{T}_3 \\
\dot{T}_4
\end{bmatrix} =
\begin{bmatrix}
a_{11} & a_{12} & a_{13} & a_{14} \\
a_{21} & a_{22} & a_{23} & a_{24} \\
a_{31} & a_{32} & a_{33} & a_{34} \\
a_{41} & a_{42} & a_{43} & a_{44}
\end{bmatrix}
\begin{bmatrix}
T_1 \\
T_2 \\
T_3 \\
T_4
\end{bmatrix} +
\begin{bmatrix}
b_1 \\
b_2 \\
b_3 \\
b_4
\end{bmatrix}
\]

or in condensed form

\[
\begin{bmatrix}
\dot{T}(t)
\end{bmatrix} =
\begin{bmatrix}
A
\end{bmatrix}
\begin{bmatrix}
T(t)
\end{bmatrix} +
\begin{bmatrix}
B
\end{bmatrix}
\]

The initial conditions are at \( t = t_o \)

\[
\begin{bmatrix}
T_1(t_o) \\
T_2(t_o) \\
T_3(t_o) \\
T_4(t_o)
\end{bmatrix} =
\begin{bmatrix}
c_1 \\
c_2 \\
c_3 \\
c_4
\end{bmatrix}
\]

or

\[
\begin{bmatrix}
T_o
\end{bmatrix} =
\begin{bmatrix}
c
\end{bmatrix}
\]
To simplify the solution of equation C.1, it is assumed that $a_{ij}$ and $b_i$ are constants for the time interval under consideration.

An exact solution to this set of coupled differential equations is possible using the method of Laplace transforms. In theory, this approach can be employed successfully if the initial conditions are known. In practice, however, the initial transformation, the ensuing matrix manipulation and the inverse Laplace transformations become overwhelming for this system size. Alternatively, the equations can be uncoupled using the method of determinants and solved independently. However, the resulting equations are no less difficult to solve than the original coupled set.

Bronson develops a matrix approach to the solution of the initial value problem

$$C.5 \quad [\dot{T}(t)] = [A] [T(t)] + [f(t)]$$

with

$$[T(t_o)] = [C]$$

where the matrix $[A]$ is a constant coefficient matrix and $[C]$ is the initial condition matrix. Equation C.5 is identical to C.2 if $[f(t)]$ is just a constant matrix.

The result of Bronson's analysis yields the exact solution

$$C.6 \quad [T(t)] = e^{[A](t-t_o)} \begin{bmatrix} C \end{bmatrix} + e^{[A]t} \int_{t_o}^{t} \left\{ e^{-[A]s} \right\} ds$$

$$[f(s)] ds$$
If the time axis is adjusted so that the initial condition of any time interval occurs at time \( t \) equal to zero then equation C.6 becomes

\[
\begin{align*}
\mathbf{T}(t) &= e^{\mathbf{A}t} \cdot \mathbf{C} + e^{\mathbf{A}t} \int_0^t e^{-\mathbf{A}s} \{f(s)\} ds.
\end{align*}
\]

The matrix \( e^{\mathbf{A}t} \cdot \mathbf{C} \) constitutes the transient portion while the integrated term constitutes the steady state solution. Equation C.7 now allows the calculation of the four node temperatures, \( T_i \), at the end of some time interval \( (t = n\Delta t) \) when the matrix \( \mathbf{A} \) can be considered constant for the interval.

To employ equation C.7, the calculation of the matrix \( e^{\mathbf{A}t} \) is required. This term may be expressed as

\[
\begin{align*}
e^{\mathbf{A}t} &= a_3\{\mathbf{A}t\}^3 + a_2\{\mathbf{A}t\}^2 + a_1\{\mathbf{A}t\} + a_0\mathbf{I},
\end{align*}
\]

where \( \mathbf{I} \) = identity matrix

and \( a_i \) is found by solving the equations

\[
\begin{align*}
e^{\lambda_1} &= a_3\lambda_1^3 + a_2\lambda_1^2 + a_1\lambda_1 + a_0, \\
e^{\lambda_2} &= a_3\lambda_2^3 + a_2\lambda_2^2 + a_1\lambda_2 + a_0.
\end{align*}
\]
\[ e^{\lambda t} = a_3 \lambda_3^3 + a_2 \lambda_2^2 + a_1 \lambda_1 + a_0 \]

\[ e^{\lambda t} = a_3 \lambda_3^4 + a_2 \lambda_2^2 + a_1 \lambda_1 + a_0 \]

where \( \lambda_i \) = eigenvalues of the matrix \([A]_t\)

It becomes apparent that the calculations to obtain \( e^{[A]_t} \) are time consuming. This is due mainly to the problem of obtaining the eigenvalues for the four by four matrix \([A]\).

An alternate method for obtaining the matrix \( e^{[A]_t} \) is use of the definition

\[ e^{[A]_t} = \sum_{K=0}^{\infty} \frac{([A]_t)^K}{K!} = [I] + \frac{[A]_t}{1!} + \frac{([A]_t)^2}{2!} + \ldots \]

where \([I]\) is the identity matrix. The error of taking a finite sum is at the most the size of the first neglected term. Therefore, the number of terms needed for a desired accuracy depends largely on the time step \( (t = \Delta t) \) chosen.

Using the results of equations C.8 or C.10, the exact solution to the set of coupled differential equations can be calculated from equation C.7. It should be recalled that the solution is for a constant \([A]\) matrix. If \([A]\) is not constant but is a function of the dependent variables \( T_i \), a first guess at \([A]\) can be made. Using this matrix, a solution for \([T]\) can be found. Then using \([T]\), a new \([A]\) matrix can be calculated which leads to a new solution for the \([T]\) matrix. This technique can be carried out until there
is little change in the $[T]$ matrix.

In many cases with differential equations, exact solutions become prohibitive with respect to the time required for a solution. The alternative is a numerical approach. Perhaps the easiest numerical solution technique which could be used to solve equation C.1 is the common predictor-corrector method. There are several different routines, many of which have higher order predictors and correctors. For purposes of illustration, the Adams-Moulton predictor-corrector of order 3 is described here. The predictor is

$$C.11 \quad T_{n+1}^p = T_n + \frac{\Delta t}{24} \left[ 55 \dot{T}_n - 59 \dot{T}_{n-1} + 37 \dot{T}_{n-2} - 9 \dot{T}_{n-3} \right]$$

where

- $n$ = present time
- $p$ = predicted value
- $\Delta t$ = time step
- $T$ = dependent variable to be predicted
- $\dot{T}$ = time derivative of dependent variable

For the system of four equations, there will be four predicted values. The next step is to calculate the $(n + 1)$ time derivatives of the dependent variables using the predicted values. That is,

$$C.12 \quad \dot{T}_{n+1} = f \left( T_n^p + 1 \right)$$

where $f$ is the differential equation.
Now the corrector is

$$T_{n+1}^c = T_n + \frac{\Delta t}{24} \left[ 9T_{n+1}^* + 19T_n^* - 5T_{n-1}^* + T_{n-2}^* \right]$$

where the superscript $c$ denotes the corrected value. One can accept this corrected value or use it to go through the routine again until some convergence occurs between the predicted and corrected values.

For a set of coupled differential equations, it is difficult to determine stability and error criteria by other than a trial basis. However, for a single first-order equation, Gerald gives the convergence criteria as

$$\Delta t < \frac{24/9}{|\dot{T}|}$$

$$D \cdot 10^N < \frac{24/9}{\Delta t|\dot{T}|}$$

and the accuracy criteria as

$$D \cdot 10^N < 14$$

where

$D = \text{difference between corrected and predicted value}$

$N = N^{th} \text{ decimal place}$

$\dot{T} = \text{value of derivative}$

The error of this method is on the order of $(\Delta t)^5$.

Although the predictor-corrector can be readily applied, it is an explicit method and is subject to oscillations which can
easily cause instability. The alternative is an implicit method which is not subject to instability. The Crank-Nicholson technique is such a method. This technique predicts the new values by using the averages of the time derivatives at the old time, \( t \), and the new time, \( t + \Delta t \). That is,

\[
[T]^{(V+1)} = [T]^{(V)} + \frac{1}{2} \left( [T]^{(V)} + [T]^{(V+1)} \right) \Delta t
\]

where

- \([T] =\) dependent variable matrix (equation C.2)
- \([\dot{T}] =\) time derivative of
- \(\Delta t =\) time step
- \(V = \) \(v^{th}\) time period

From the differential equations describing the system (equation C.1)

\[
\]

\[
[T]^{(V+1)} = [A]^{(V+1)}[T]^{(V+1)} + [B]^{(V+1)}
\]

Substitution of these two equations into equation C.14 yields after suitable manipulation

\[
\left\{ [I] - \frac{1}{2}[A]^{(V+1)}\Delta t \right\} [T]^{(V+1)} = [T]^{(V)} + \frac{1}{2} \left\{ [A]^{(V)} \right\} . \\
[T]^{(V)}\Delta t + \frac{1}{2} \left\{ [B]^{(V)} + [B]^{(V+1)} \right\} \Delta t
\]

An iterative process can be used to obtain \([T]^{(V+1)}\). That is, assume \([A]^{(V+1)}\) and \([B]^{(V+1)}\) are approximately equal to \([A]^{(V)}\) and
\[ [B]^{(V)} \] respectively, and solve equation C.17 for \([T]^{(V+1)}\). This process can be repeated until a convergence criteria is satisfied.
Appendix D

EXPERIMENTAL DATA

The following presents the raw data from the tests performed on the thermosyphon system described in Chapter VII (see Figure 22). Also included is specific information on the characteristics of the system.

Solar Collector Characteristics

<table>
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<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>collector slope</td>
<td>34°</td>
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<tr>
<td>surface area</td>
<td>48 sq. ft.</td>
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<td>glass cover thickness</td>
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<td>glass emissivity</td>
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<td>absorber plate thickness</td>
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<td>absorptance</td>
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<tr>
<td>absorber emissivity</td>
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<tr>
<td>tube to plate bond</td>
<td>continuous solder</td>
</tr>
<tr>
<td>tube configuration</td>
<td>2 parallel sinusoids (10 passes)</td>
</tr>
<tr>
<td>inside tube diameter</td>
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<tr>
<td>outside tube diameter</td>
<td>.875 in.</td>
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<tr>
<td>back insulation thickness</td>
<td>.5 in.</td>
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</table>
insulation conductivity  \( \cdot03 \text{ BTU/hr-ft } °\text{F} \)
collector container  galvanized steel
container emissivity  \( \cdot1 \)
collector perimeter  32 ft.
collector depth  3 in.

**Storage Tank Characteristics**

tank capacity  80 gal.
height  4 ft.
diameter  1.84 ft.
insulation thickness  0.5 ft.
insulation conductivity  \( \cdot028 \text{ BTU/hr-ft } °\text{F} \)

More system geometry is given in Figure 36.
The following is the instrumentation used to obtain data recorded September 28, 1976, and September 29, 1976, on a thermosyphon system at Florida Technological University.
Figure 36. Geometry of System

Storage Tank

Collector

3.4 ft

2.2 ft

4.5 ft

8.0 ft

2.1 ft

0.5 ft
Insolation

Eppley pyranometer with Leeds and Northrup Speedomax recorder

Flow Rate

Barco 220 venturi with Validyne pressure transducer/amplifier (flow is given by:

\[ 10.53 \times 10^{0.56578\log(\Delta P)} \text{ in GPH} \]

\[ \Delta P = 2.77 \times \text{voltage reading} \]

Temperature (uncorrected)

Copper-Constantan thermocouples with Omega Omni-Amp millivolt amplifier/reference junction

Load Flow

Carlon Flow Detector

Wind Speed

Texas Electronics Wind Direction and Velocity Monitor and Recording System

Voltage Signals

Data Precision digital multimeter
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Appendix E

TSP FLOWCHART

The following represents a detailed flowchart for the TSP. Also included is a complete TSP listing and sample output.
Set Old Tank Derivatives to Zero

Collector Temperature Distribution Initialized

Collector Coordinate System Established

Calculate Volume, Mass of Storage Tank Fluid

Calculate Capacitance, Area of Each Tank Section

B
Daily Total of Loading Set to Zero

Daily Totals of Insolation, Useful Gain Set to Zero

Collector Flow Rate Set to Zero

Save Initial Mean Tank Temperature

Save Initial Flow Rate
Calculate Fraction of Insolation Absorbed by Collector

Calculate Energy Absorbed by Cover and Plate

Old Collector Node Temps. Set Equal to Present Values

Old Tank Node Temps. Set Equal to Present Values

Set Last Calculated Flow Rate Equal to New Flow Rate
Use Matrix Solution?

Yes

Set up Coefficient Matrix

Use Matrix Solution to Calculate Collector Node Temps.

Recalculate Coefficients

Use Crank-Nicholson to Recalculate Collector Node Temps.

Calculate Collector Loss Coefficient

Calculate Collector Efficiency Factor

No

Calculate Coefficients Using Old Collector Node Temps.

Use Crank-Nicholson to Calculate Collector Node Temps.
Calculate Collector Temperature Distribution

Predict New Tank Node Temperatures

Calculate Derivatives of Tank Temperatures

Calculate Corrected Tank Node Temperatures

Calculate New Flow Rate
Flow Rate Converged?

Yes

Calculate Derivatives of Tank Temperatures

Reset Tank Derivative Matrix

Increase Time by One Interval

Calculate Mean Tank Temperature

No

F

K

G
Calculate Temperature Rise Across Collector

Calculate Collector Useful Energy Gain

Calculate Energy Load on Storage Tank

Calculate Collector Efficiency

Calculate Daily Totals to Present

G ——> H
Set Old Flow Rate Equal to Last Known Flow Rate

Print Interval Data

Set Collector Temp. Distribution Equal to Old Values

Last Interval?

Yes
Calculate Daily Efficiency

No

I

J

H
Calculate System Efficiency

Print Daily Totals and Efficiency

Print System Efficiency

Stop
THIS PROGRAM IS A SIMULATION FOR A THERMOSYPHON SYSTEM. IT EMPLOYS TRANSIENT MODELS FOR THE COLLECTOR AND STORAGE TANK. PIPE LOSSES ARE NEGLECTED. IT IS ASSUMED THAT THE SYSTEM IS CONTROLLED ONLY TO THE EXTENT THAT REVERSE FLOW DOES NOT OCCUR.

IN ITS PRESENT CONFIGURATION, THIS PROGRAM REQUIRES THE COLLECTOR TO BE A SINGLE COVER, FLAT PLATE TYPE WITH A TUBE OVER SHEET CONFIGURATION. THE TANK IS ASSUMED NOT TO CONTAIN AUXILIARY HEATING ELEMENTS OR HEAT EXCHANGERS. THE PIPING IS ASSUMED TO BE SUCH THAT THE TOP FOURTH OF THE TANK RECEIVES FLUID FROM THE COLLECTOR AND DELIVERS FLUID TO THE LOAD. THE BOTTOM FOURTH OF THE TANK RECEIVES MAKEUP FLUID AND DELIVERS FLUID TO THE COLLECTOR.

THE PROGRAM MUST BE INITIALIZED AT SOME TIME EARLY IN THE DAY WHEN IT CAN BE ASSUMED THAT THE COLLECTOR IS AT AMBIENT TEMPERATURE AND THE TANK HAS BEEN IN STEADY STATE FOR FOUR PREVIOUS TIME STEPS.

ALL EQUATIONS INVOLVING VISCOSITY AND DENSITY OF THE WORKING FLUID IS FOR WATER. THEREFORE, THIS SIMULATION IS FOR A SYSTEM UTILIZING ONLY WATER.

THE FOLLOWING INPUT VARIABLES ARE REQUIRED:

COLLECTOR CHARACTERISTICS

SLOPE = COLLECTOR SLOPE (RADIANS)
T = THICKNESS OF COVER PLATE ON COLLECTOR
EC = EXTINCTION COEFFICIENT OF COVER PLATE
REFIDX = REFRACTIVE INDEX OF COVER PLATE
UC = BOND CONDUCTANCE
W = AVERAGE FIN WIDTH FOR COLLECTOR TUBE
B = DISTANCE BETWEEN TUBE CENTERS
DI = INSIDE TUBE DIAMETER
LP = HALF THE DISTANCE BETWEEN BONDS
DE = OUTSIDE TUBE DIAMETER
PER = PERIMETER OF COLLECTOR
DEP = DEPTH OF COLLECTOR
MCINS = THERMAL CAPACITANCE OF BACK INSULATION
MCPAN = THERMAL CAPACITANCE OF CONTAINER
CA = OVERALL COLLECTOR CAPACITANCE PER UNIT AREA
EMISP = EMISSIVITY OF ABSORBER PLATE
EMISG = EMISSIVITY OF COVER PLATE
EMISPN = EMISSIVITY OF CONTAINER
VF = VIEW FACTOR FROM ABSORBER TO COVER PLATE
CP = SPECIFIC HEAT OF WORKING FLUID
AC = NOMINAL SURFACE AREA OF COLLECTOR
CV = CONVECTION COEFFICIENT DEPENDENT ON SLOPE OF COLLECTOR
KINS = THERMAL CONDUCTIVITY OF BACK INSULATION IN COLLECTOR
XINS = THICKNESS OF BACK INSULATION
TUBLEN = TOTAL LENGTH OF TUBE IN COLLECTOR
DELTA = THICKNESS OF ABSORBER SHEET
K = THERMAL CONDUCTIVITY OF ABSORBER
ALPHA = ABSORTIVITY OF ABSORBER
THE FOLLOWING INFORMATION MUST BE ACCURATE. AS A CHECK, DOES
(# OF BENDS IN COLLECTOR)\times (H + HC) = DISTANCE BETWEEN INLET AND
OUTLET AT COLLECTOR?

UP = DISTANCE FROM TANK BOTTOM TO COLLECTOR INLET PIPE
DOWN = DISTANCE FROM TANK TOP TO COLLECTOR OUTLET PIPE
BE = CIRCUMFERENTIAL LENGTH OF A BEND IN THE COLLECTOR TUBES
TFB = DISTANCE FROM INLET TO FIRST BEND IN COLLECTOR TUBE
TLB = TUBE LENGTH BETWEEN BENDS IN COLLECTOR
LIN = LENGTH OF INLET PIPE FROM TANK TO COLLECTOR
LOUT = LENGTH OF OUTLET PIPE FROM TANK TO COLLECTOR
DYIN = VERTICAL LENGTH OF INLET PIPE
DYOUT = VERTICAL LENGTH OF OUTLET PIPE
H = VERTICAL DISTANCE ACROSS BEND IN COLLECTOR
MC = VERTICAL DISTANCE FROM THE END OF ONE BEND TO THE
BEGINNING OF A SUCCEEDING BEND
HEIGT = HEIGHT OF TANK
SLOPE = SLOPE OF COLLECTOR (RADIANS)

STORAGE TANK CHARACTERISTICS

US = OVERALL LOSS COEFFICIENT FOR STORAGE TANK
DIAM = DIAMETER OF TANK

TEST DATA

TIME = TIME OF TEST
PERIOD = LENGTH OF TIME INTERVALS (MINUTES)
INTRVL = TOTAL NUMBER OF TIME INTERVALS
RADTN = VALUE OF INTEGRATED NORMAL SOLAR RADIATION
TAMB = AMBIENT TEMPERATURE
VEL = WIND VELOCITY
MOL = LOAD FLOW RATE
TLOAD = TEMPERATURE OF RETURN FLUID
TST(N) = INITIAL TEMPERATURES OF FOUR TANK NODES
TCO(N) = INITIAL TEMPERATURES OF FOUR COLLECTOR NODES

MISCELLANEOUS

SIGMA = ROLTZMAN'S CONSTANT
SECTNS = DESIRED NUMBER OF SECTIONS COLLECTOR TUBE IS TO BE DIVIDED
MATRIX = 1 FOR THE MATRIX SOLUTION TO THE DIFFERENTIAL
EQUATION, 0 FOR THE CRANK-NICHOLSON
TERMS = DESIRED NUMBER OF TERMS FOR THE MATRIX SOLUTION
SHADE = FRACTION OF RADIATION REDUCED FROM SHADING
FK = FRICTION FACTOR ASSOCIATED WITH BENDS, TEES, ETC.
IGATE = # OF GATE VALVES IN INLET PIPE
ITEE = # OF TEES IN INLET PIPE
ILONGC = # OF LONG ELBOWS IN INLET PIPE
ISTC = # OF SHORT ELBOWS IN INLET PIPE
NGATE = # OF GATE VALVES IN OUTLET PIPE
NLONGC = # OF LONG ELBOWS IN OUTLET PIPE
NSTC = # OF SHORT ELBOWS IN OUTLET PIPE
EG = EQUIVALENT LENGTH OF A GATE VALVE
ET = EQUIVALENT LENGTH OF A TEE
ELC = EQUIVALENT LENGTH OF A LONG ELBOW
ESTC = EQUIVALENT LENGTH OF A SHORT ELBOW
DIMENSION RADTN(144), TAMB(144), VEL(144), MOLO(144), TCO(4), TST(4),
1 DTS(4*4), TFOLD(101), X(101), TF(101), TCOLD(4)
2 DIMENSION DERT(4), TSTOLD(4), AA(4), A(4*4), F(4*1)
3 REAL KL, LP, XINS, X, MCP, MCPAN, MDC, MDO, MCO
4 REAL MCINS, MOLO, LENGTH, MDCL, MASS, MDCST, LIN, LOUT
5 INTEGER SECTNS, TIME, PERIOD, TERMS
6 READ, T, EC, REFIDX, UC, B, D, DI, LP, OE, PER, DEP,
1 MCP, MCPAN, MCINS, CA, EMISP, EMISG, EMISPAN, AC, CV, XINS
7 READ, TUBLEN, K, DELTA, ALPHA, US, DIAM, CP, SIGMA,
1 PERIOD, TIME, INTRVL, SECTNS, VF, SHADE
8 READ, UP, DOWN, BE, TFB, TLB, LIN, LOUT, D YIN, DYOUT, H, MC, HEIGHT,
1 SLOPE
9 READ, FK, IGATE, ITEE, ILONGC, ISTC, NGATE, NLONGC, NSTC,
1 EG, ET, EL, ESTC
10 READ, (RADTN(I), I=1, INTRVL)
11 READ, (VEL(I), I=1, INTRVL)
12 READ, (MOLO(I), I=1, INTRVL)
13 READ, (TAMB(I), I=1, INTRVL)
14 READ, (TST(N), N=1, 4)
15 READ, TLOAD, (TCO(I), I=1, 4)
16 READ, MATRIX, TERMS
17 DATA PI, PS/3.14159, 62.0/
18 DELTAT=PERIOD/60.
19 DO 5 NN=1, 4
20 DO 5 JJ=1, 4
21 5 DTS(NN, JJ)=0.0
22 M=SECTNS+1
23 MM=M+1
24 DO 10 NN=1, MM
25 10 TFOLD(NN)=TCO(2)
26 TFEND=TCO(2)
27 LENGTH=TUBLEN/(FLOAT(SECTNS))
28 X(1)=0.0
29 DO 15 J=2, M
30 15 X(J)=X(J-1)+LENGTH
31 VOLUME=PI*DIAM*DIAM*HEIGHT/4.0
32 MASS=PS*VOLUME
33 MDCST=MDC/4.0
34 AST=PI*DIAM*HEIGHT/4.0
35 FLAG=0
36 ELSE=0.0
37 RT=0.0
38 QUT=0.0
39 DIST=0.0
40 DISTO=0.0
41 IPAGE=18
42 MDC=0.0
43 KNI=0
44 PIPE=LIN
45 TMTS=(TST(1)+TST(2)+TST(3)+TST(4))/4.
46 MDCL=MDC
47 TAI=TAMB(1)
48 IN=TAI
49 CALL PRELIM(T, EC, REFIDX, ALPHA, AP, AB)
50 I=1
51 IHOUR=TIME/100
52 IHOUR=IHOUR+100
53 20 IF=RADTN(I)
54 TA=TAMB(I)
V=VEL(I)
IF (KNT .EQ. 1) TIN=ST(4)
TINOLD=TIN
MQL=MQL(I)*8.3/DELTAT
CALL QABSOR(RI,AP,DELTAT,AB,SHADE,S,S8)
CALL COPY1(TCO,TCOOLD+4)
CALL COPY1(TST,TSTOLD+4)
MOCO=MOC
IF (MATRIX .EQ. 0) GO TO 70
CALL AMAX(MCP,MCINS,MCINS+AC,SB,BF,SIGMA,V,CV,EMISG,1.EMISP,EMISP+KINSXINS+CP,TA,TIN,TFEND+TCOLD+MOC+A+F)
CALL SOLVE(A,F,TCOLD+TERMS,DELTAT,TCO)
GO TO 80
CALL CNSOL(MCP+MCINS+MCINS+AC,SB,BF,SIGMA,V,CV,EMISG,1.EMISP,EMISP+KINSXINS+CP,TA,TIN,TFEND+TCOLD+DELTAT,TCO)
CALL ULOSS(TCO,TCOLD+TA-SIGMA+EMISG+EMISP+V+PER+DEP+AC+1.CV+KINSXINS+EMISP+TPA+UL)
CALL FPRIME(UL+K+DELTAT+L+D+TA+DEP+UC+FP)
CALL TCDIST(DELTAT+H+CA+X+MOC+CP,TA+TST+FP+TA+S+M+TF+UL+PIPE+DI+LENGTH+DIST+KNT+TIN+TFEND+TFOLD)
CALL AOPRED(DELTAT+OTS+TSTOLD+TST)
CALL DERV(TST+MCPST+AST+MOC+CP,TA+TIN+TFEND+MOL+US+TLOAD+TA+DERT)
CALL AOCORR(DELTAT+OTS+DERT+TSTOLD+TST)
CALL MFLOW(SECTN+LENGTH+AT+TST+TIN+UP+OWN+BE+TFB+TLB,LIN+LOUT+DI+OYIN+OOUT+H+HC+DI+SLOPE+FK+HEIGHT+IGATE+ITEE+ILOGC+1.5ST+NGATE+ÑLONGC+NSTC+EG+ET+ELC+ESTC+MOC)
IF (MOC-MOCO .EQ. 0.0) GO TO 60
IF (ABS((MOC-MOCO)*2./(MOC+MOCO)) .GE. .02) GO TO 30
CALL DERV(TST+MCPST+AST+MOC+CP,TA+TIN+TFEND+MOL+US+TLOAD+TA+DERT)
CALL NEWVAL(DT+DERT)
TIME=TIME+PERI00
ICHECK=TIME-IHOUR-60
IF (ICHECK .LT. 0) GO TO 50
TIME=TIME+40
IF (TIME .GE. 1300.) GO TO 55
TIME=0.0
IHOUR=0
IHOUR=IHOUR+100
TMT=(TST(1)+TST(2)+TST(3)+TST(4))/4.
GPH=MOC/8.3
RI=AC*RI
RT=RT*RI
TRISE=((TF(M)+TFOLD(M))-(TIN+TINOLD))*5
QU+.5*(MOC+MOC)+CP+TRISE+DELTAT
QU=OUT+QU
ELOAD=ELOAD+((MQL(I)*8.3*(.5*(TSTOLD(1)+TST(1)))=TLOAD))=CP
EFF=(QU/RI)*100.
IF (EFF .GE. 100.) FLAG=1.
MOCL=MOC
DIST=DIST+(4.*MOC+DELTAT/(PS+4.DI+DI*DI))
IF (DIST .GE. PIPE) KNT=1
DIST=DIST
IPAGE=IPAGE+1
IF (IPAGE .NE. 19) GO TO 57
IPAGE=2
WRITE(6,93)
WRITE(6,94)
WRITE(6,91)
CONTINUE
WRITE(6,90) TIME, MDLO(I), GPH, TIN, TF(M), TRISE
1 (TST(J), J=1,4), TMT, RI, QU, EFF
110 CALL COPY1(TF, TFOLD, M)
111 I=I+1
112 IF (I .LE. INTRVL) GO TO 2
113 EFFD=100.*QUT/RT
114 EFFS=(MASS(TMT-TMTS)*ELOAD)/RT
115 EFFS=100.*EFFS
116 WRITE(6,95)
117 WRITE(6,96) RT, QUT, EFFD
118 WRITE(6,97) EFFS
119 IF (FLAG.EQ.1.) WRITE(6,99)
120 WRITE(6,98)
121 95 FORMAT(••+28X, 'TOTALS FOR THE DAY:•'
122 96 FORMAT(••+28X, 'SYSTEM EFFICIENCY FOR THE DAY:•'
123 97 FORMAT(••+28X, 'SYSTEM EFFICIENCY FOR THE DAY:•'
124 98 FORMAT(••+28X, 'TOTALS FOR THE DAY:•'
125 99 FORMAT(••+28X, 'SYSTEM EFFICIENCY FOR THE DAY:•'
126 FORMAT(••+28X, 'TOTALS FOR THE DAY:•'
127 FORMAT(••+28X, 'TOTALS FOR THE DAY:•'
128 FORMAT(••+28X, 'TOTALS FOR THE DAY:•'
129 FORMAT(••+28X, 'TOTALS FOR THE DAY:•'
130 FORMAT(••+28X, 'TOTALS FOR THE DAY:•'
131 STOP
132 END

SUBROUTINE PRELIM(T, EC, REFIX, ALPHA, AP, AB)
133 C THIS CALCULATES THE EFFECTIVE ABSORPTANCE OF THE COVER PLATE
134 A8=(1.-REF)/0.4/EC\*\*2.
135 EKL=EXP(-EC\*T)
136 TAU=1.-REF)/0.4/EC\*EKL
137 RAY=TAU/(1.-REF)\*EKL
138 AB=1.-REF)/0.4/EC\*EKL
139 AP=RA\*AB/TAU/(1.-REF)\*EKL
140 RETURN
141 END

SUBROUTINE QA8SOR(RI, AP, DELTAT, AB, SHADE, S, SB)
143 S=RI\*AP/(1.-SHADE)/DELTAT
145 SB=RI\*AB/DELTAT
146 RETURN
147 END
SUBROUTINE COPY(A,B,LL)
DIMENSION A(LL),B(LL)
      00 10 N=1
      10 B(N)=A(N)
      RETURN
      END

SUBROUTINE ADPRED(H,DT,TOLD,T)
THIS IS THE ADAMS MOULTON PREDICTOR
DIMENSION T(4),DT(4),TOLD(4)
      DO 10 N=1,4
      10 T(N)=TOLD(N)+H*(55.*DT(N,1)-59.*DT(N,2)+37.*DT(N,3)-9.*DT(N,4))/24.
      RETURN
      END

SUBROUTINE ADCORR(H,DER,TOLD,T)
THIS IS THE ADAMS MOULTON CORRECTOR
DIMENSION T(4),DER(4),DT(4),TOLD(4)
      DO 10 N=1,4
      10 T(N)=TOLD(N)+H*(9.*DER(N)+19.*DT(N,1)-5.*DT(N,2)+DT(N,3))/24.
      RETURN
      END

SUBROUTINE NEWMAL(DT,DER)
THIS RESETS THE VALUES OF THE DERIVATIVES FOR THE NEXT
ADAMS MOULTON STEP
DIMENSION DT(4),OLD(4),DER(4)
      DO 30 N=1,4
      30 OLD(N,J)=DT(N,J)
      DO 10 N=1,4
      10 DT(N+1)=DER(N)
      DO 20 N=1,4
      20 DT(N+1)=OLD(N,J-1)
      RETURN
      END

SUBROUTINE AMAX(MCP,MCG,MCINS,MCPAN,AC,S,SB,VF,SIGMA,V,CV,1
EMISP,EMISP,EMISP,KINS,KINS,CPI,TAN,TEND,TCOLD,MDC,A,F)
THIS SETS UP THE A MATRIX FOR USE IN SOLVING THE
DIFFERENTIAL EQUATIONS
DIMENSION TCOLD(4),A(4,4),F(4,1)
REAL MDC,KINS,MCP,MCG,MCINS,MCPAN
      00 10 I=1
      10 DR=459.67
      DO 10 I=1,4
      10 F(I,1)=0.0
      DO 10 J=1,4
      10 A(I,J)=0.0
      CONTINUE
      HPG=CV*ABS(TCOLD(2)-TCOLD(1))-0.25
      M1R=SIGMA*(TCOLD(1)+TCOLD(2)+2.*DR)*(TCOLD(1)+OR)*0.25
      M1R=M1R/(1.+EMISP)*(1.+EMISP)-1.
      M1R=M1R*VF-HPG
      H2=EMISP*(SIGMA*(TCOLD(1)+TA+2.*DR)*(TCOLD(1)+DR)*0.25
      CONTINUE
      HPG=CV*ABS(TCOLD(2)-TCOLD(1))-0.25
      M1R=SIGMA*(TCOLD(1)+TCOLD(2)+2.*DR)*(TCOLD(1)+OR)*0.25
      M1R=M1R/(1.+EMISP)*(1.+EMISP)-1.
      M1R=M1R*VF-HPG
      H2=EMISP*(SIGMA*(TCOLD(1)+TA+2.*DR)*(TCOLD(1)+DR)*0.25
SUBROUTINE EXCUT (A,F,TCOLD,K,DT,TCO)

DIMENSION A (4,4),F (4,1),XO (4,1),EAT (4,4),X (4,1),SUM (4,1),EATX0 (4,1),EATSUM (4,1),TCOLD (4),TCO (4)

DO 10 I=1,4
10 XO(I,1)=TCOLD(I)

CALL EXCINT (A,F,DT,K,EAT,STUM)
CALL MULT (EAT,XO,EATX0,4+4+1)
CALL MULT (EAT,STUM,EATSUM,4+4+1)
CALL ADD (EATX0,EATSUM,X4+1)
DO 20 I=1,4
20 TCO(I)=X(I,1)
RETURN
END

SUBROUTINE EXCINT (A,F,DT,K,EAT,STUM)

DIMENSION A (4,4),F (4,1),XO (4,1),EAT (4,4),X (4,1),SUM (4,1),ANF (4,1),TEMP (4,4),AF (4,1),EAT (4,4)

DO 10 I=1,4
10 DO 20 J=1,4
20 EAT(I,J)=0.0

Q(I,J)=0.0
AN(I,J)=A(I,J)
IF (I.EQ. J) Q(I,J)=1.0
CONTINUE

CALL SCALAR (DT,F,STUM,4+1)
CALL SCALAR (DT,F,STUM,4+1)
CALL ADD (Q,AT,EAT,4+4)
CALL MULT (A,F,AF,4+4+1)
COEF=DT*DT/2.
CALL SCALAR (COEF,AF,4+4+1)
CALL ADD (AF,STUM,STUM,4+1)
COEF=COEF
KK=K-2
DO 20 N=1, KK
DO 99 I=1,4
99 SUM(I)=SUM(I)+1
DO 99 II=1,4
IF (N_GT_1) COEF=DT/(N+1)
CALL SCALAR(COEFF=AN*AN+4*4)
CALL ADD(AN+AT+AT+4*4)
CALL MULT(TMP+F+ANF+4*4)
COEF=(COEFF*DT/(N+2))*((-1)**(N+1))
CALL SCALAR(COEFF=ANF*ANF+4*4)
CALL ADD(ANF+SUM+SUM+4*4)

CONTINUE
RETURN
END

SUBROUTINE SCALAR(SCALE=X+Z+L+M)
DIMENSION X(L+M),Z(L+M)
DO 10 I=1,L
DO 10 J=1,M
Z(I,J)=SCALE*X(I,J)
RETURN
END

SUBROUTINE MULT(X,Y,Z,M+N)
DIMENSION X(L,M),Y(M,N),Z(L,N)
DO 10 I=1,L
DO 10 J=1,N
Z(I,J)=X(I,J)*Y(K,J)
RETURN
END

SUBROUTINE ADD(X,Y,Z,L+M)
DIMENSION X(L,M),Y(L,M),Z(L+M)
DO 10 I=1,L
DO 10 J=1,M
Z(I,J)=X(I,J)+Y(I,J)
RETURN
END

SUBROUTINE CNSOL(MCP,MCINS,MCPS,MCINS,MCPS,MCINS,MCPS,
EMISP,EMISP,EMISP,EMISP,EMISP,EMISP,EMISP,EMISP,
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SUBROUTINE RECUR(A,F,T)
THIS SUBROUTINE CALCULATES THE MATRIX EQUATION \( A \cdot T = F \) BY THE RECURSION FORMULA
DIMENSION \( A(4,4), F(4,1), T(4,1) \), \( \beta(4), \gamma(4) \)
00 10 I=1,4
B(I)=A(I,1)
AA(I+1)=A(I+1,1)
GAM(I=AA(I+1)*GAM(I-1))/BETA(I)
DO 20 I=2,4
BETAC(I)=B(I)*\( A(I+1, I) \) - \( C(I+1) \cdot T(I+1,1) \)/BETA(I)
GAM(I)=B(I)/BETA(I)
DO 30 I=2,4
K=4-I
T(K+1)=GAM(K)-C(K)*T(K+1,1))/BETA(K)
RETURN
END

SUBROUTINE ULOSS(TCO,TCOLD,TA.SIGMA,EMISG,EMISP,EPSI,PER,DEP,AC,1.
C,KINS,XINS,EMISP,TPA,UL)
DIMENSION TCO(4),TCOLD(4),TAVE(4)
REAL KINS
DO 10 N=1,4
TAVE(N)=(TCO(N)+TCOLD(N))/2.
TPA=TAVE(2)
DIFF=(TAVE(2)-TA)
IF (DIFF LT 1.) GO TO 50
QRAD=SIGMA*(TAVE(2)+59.87)**4 - (TAVE(1)+59.87)**4
QRAD=QRAD/(1./EMISP*(1./EMISG)-1.)
QCON=ABS(TAVE(2)-TAVE(1))
IF (QCON LT .001) GO TO 40
QCONV=(QCON**25)*CV*(TAVE(2)-TAVE(1))
GO TO 45
342 40 QCONV=0.
343 45 QTOP=QRAD+QCONV
344 QREAR=2.*KINS*(TAVE(2)-TAVE(3))/XINS
345 QEDGE=.*08*PER*DEP*(TAVE(2)-TA)/AC
346 QTOTAL=QTOP+QREAR+QEDGE
347 UL=QTOTAL/ABS(TAVE(2)-TA)
348 IF (UL .GT. 0.0) GO TO 60
349 50 UL=4.*SIGMA*(TA+459.68)*3/((1./EMISP)+(1./EMISP)+1.)
350 UL=UL+(.*08*PER*DEP/AC)
351 UL=UL+(2.*KINS/XINS)
352 60 RETURN
353 END
354 SUBROUTINE FPRIME(UL, KDELTA, LP, B, OI, TPA, DE, UC, FP)
355 \ THIS CALCULATES THE FIN EFFICIENCY FACTOR
356 REAL KLP
357 IF (UL .LE. 0.0) GO TO 10
358 F= TANH(SQRT(UL/(K*DELTA))*LP)/(SQRT(UL/(K*DELTA))*LP)
359 HF= 3.636*COND(TPA)/OI
360 FP= (B/((B-DE)*F)) + (B*UL/UC)
361 FP= FP + (3.14159*O1*HF)
362 FP= FP / FP
363 GO TO 20
364 10 FP= 1.0
365 20 RETURN
366 END
367 END
368 SUBROUTINE TCDIST(DELTA, W, CA, X, MDC, CP, TA, I, TST, FP, TA, S, M, TF, UL, 
1 PIPE, OI, LENGTH, UST, KN, TIN, TFEND, TFOLD)
369 \ THIS CALCULATES THE TEMP. DISTRIBUTION IN THE COLLECTOR
370 \ REAL MDC, LENGTH, LIN
371 \ DIMENSION TST(4), X(100), TF(100), TFOLD(100)
372 \ **** FOR TWO SINUSOIDAL COLLECTORS IN PARALLEL,
373 \ CHANGE 1 IN NEXT CARD TO 2
374 MDC=MDC/1.
375 DO 260 J=1,M
376 \ VALU=X(J)-((MDC*CP*DELTA)/(W*CA))
378 IF (VALU .LT. 0.0) GO TO 10
379 IF (I .GT. 1) GO TO 10
380 TPLUS=TA
381 TMINUS=TA
382 \ GO TO 80
383 TPLUS=TFOLD(IFIX(VALU/LENGTH+2,0))
384 TMINUS=TFOLD(IFIX(VALU/LENGTH+1,00))
385 FRAC=VALU-IFIX(VALU/LENGTH+1,00)
386 VALUF=FRAC*(TPLUS-TMINUS)/LENGTH+TMINUS
387 RO=FP*UL*DELTA/CA
388 TF(J)= (EXP(RO)) *(VALU-(S/UL)-TA)+(S/UL)+TA
389 GO TO 260
10 IF (KNT .EQ. 1) GO TO 20
20 VALUF=TST(4)

IF (DIST .LT. PV) VALUF=TA
GO TO 30

TIME=X(J)*DELTAT/(X(J)-VALU)
RO=-FP*UL*TIME/CA
TF(J):(EXP(RO)*CVALUF-<SIUL>-TAJ*CS/UL)*TA
CONTINUE

MOC=MOC*1.
RETURN

SUBROUTINE DVRVT(TST, MCPST, AST, MOC, CP, TFEND, MDL, US, TLOAD, TA, DERT)

DIMENSION TST(4), DERT(4)

DERT(1)=((MOC*CP*(TFEND-TST(1)))-(MDL*CP*(TST(1)-TST(2)))-
1 (AST*US*(TST(1)-TA)))/MCPST

DO 10 J=2,3
10 DERT(J):((MOC*CP*(TST(J-1)-TST(J)))-(MDL*CP*(TST(J)-
1 TST(J+1)))-(AST*US*(TST(J)-TA)))/MCPST

DERT(4)=((MOC*CP*(TST(3)-TST(4)))-(MDL*CP*(TST(4)-TLOAD))-
1 (AST*US*(TST(4)-TA)))/MCPST
RETURN
END

SUBROUTINE MFLOW(SECTNS, LENGTH, TF, TST, TIN, UP, DOWN, BE, LP, LIN, LOUT, OYIN, OYOUT, H, X, OI, SLOPE, FK, HEIGHT, IGATE, ITEE, ILONGC, ISTC, NGATE, NLONGC, NSTC, GE, ET, ELGC, ESTC, MOD)

INTEGER SECTNS
REAL MOC, MOD
REAL TF(101), TST(4)
REAL LENGTH, L, LP, LIN
DATA PI, PS, GRAV, 3.141593, 62.82, 3.17/
DATA A, R, C, -.6965368, -4.659389E-1, -46.8964/
DATA IPASS, 1 /
DATA RO(DUMMY) = (A*(DUMMY+459.67)+B)*(DUMMY+459.67)+C
K=L/LENGTH
M=SECTNS*K
S=X/LP
SLOP=SIN(SLOPE)
HT=0.
F2S=0.
NS=1
NF=K

DO 520 J=NS, NF

DY=X/(2*LP)*LENGTH*SLOP
T=TF(J)
HT=HT+RO(T)*DY
CONTINUE

IF(J+GE, SECTNS) GO TO 600

J=K+1

CALCULATION OF HT IN CURVED SECTIONS STARTS HERE
437 550  J=J+1
438 551  DY=(H+(2*LENGTH-BE)*S)*SLOP
439 552  T=TF(J)
440 553  HT=HT-RO(T)*DY

C CALCULATE - FRICTION LOSS SUM TERM FOR CURVES
441 554  F2S=F2S+1./RO(T)**2
442 555  J=J+1
443 556  IF(J.EQ.(K+3)) EXCESS=L-LENGTH
444 557  NS=J
445 558  NF=SECTNS
446 559  GO TO 500

C CALCULATION OF HT FOR STRAIGHT SECTION
447 560  IF(J.EQ.(K+3)) EXCESS=L-LENGTH
448 561  EXCESS=EXCESS*LENGTH*BE

C TEST TO SEE IF IT IS IN A BEND
449 562  IF((J.LE.10) .AND. EXCESS.GT.LENGTH) GO TO 550
450 563  GO TO 570

C ADD TO HT THE CONTRIBUTION FROM THE TANK AND THE PIPES
451 564  HT=HT+RO(T)*DYIN
452 565  T=TF(SECTNS)
453 566  HT=HT+RO(T)*DYOUT
454 567  DY=HEIGHT/4.
455 568  TT=TST(1)
456 569  HT=HT+.5*(RO(T)*RO(T))*(DY/2.)*DOWN
460 569  DO 450 N=1,3
461 570  TT=TST(N)
462 571  HT=HT+.5*(RO(T)*RO(T)*DY)
463 572  TT=TST(4)
464 573  T=TIN
465 574  HT=HT+.5*(RO(T)*RO(T))*(DY/2.)*UP
466 575  HT=HT/PS
467 576  IF (HT.LT.0.0) GO TO 580

C BEGIN CALCULATION OF FRICTION FACTORS FO & FT
468 577  FS=0.
469 578  DO 620 J=1,SECTNS
470 579  T=TF(J)
471 580  FS=FS+VIS(T)*LENGTH/RO(T)**2
472 581  CONTINUE

C FOR TWO SINUSOIDAL COLLECTORS IN PARALLEL.
473 582  CHANG IN NEXT CARD TO 2
474 583  FTEMP=FS/1.
475 584  FS=0.0
476 585  IF(IPASS.NE.1) GO TO 630
477 586  CALL EQUIVALENCE(IGATE,ITEE,ILONGC,ISTC,NGATE,NLONGC,NSTC,
478 587  LENSITY,ELC,ESTC,EOIN,EOOUT)
479 588  IPASS=2

C INCREASE FS TO INCLUDE FRICTION IN THE INLET AND OUTLET PIPES
480 590  LIN=LIN+EOIN
481 591  LOUT=LOUT+EOOUT
482 592  IF(IPASS.NE.1) GO TO 630
483 593  FS=FS+VIS(T)*LIN/RO(T)**2
484 594  T=TF(SECTNS)
485 595  FS=FS+VIS(T)*LOUT/RO(T)**2
F1S=F1S+FTemp

*** FOR TWO SINUSOIDAL COLLECTORS IN PARALLEL,
CHANGE 1 IN NEXT CARD TO 4

F2S=F2S/1.
FO=128./((DI**4)*PI*GRAV)*F1S
FT=8.*FK/(PI*PI*GRAV*(DI**4.))*F2S
MDC=(SQRT(FO*FO**4.*FT*HT)-FO)/(2*FT)
MOC=3600.*MDC
GO TO 590
MOC=0.0
CONTINUE
RETURN

FUNCTION VIS(TEMP)
CALCULATES THE ABSOLUTE VISCOSITY OF WATER AT TEMPERATURE TEMP.
DATA ABSV20/1.0002/
TC=(TEMP-32.)/1.8
IF(TC.GE.20.) GO TO 700
T=(TC-20)
X=1301.1((.00585*T+8.1855)*T+998.333)-3.30233
VIS=(10.*X)**.000672
RETURN
X=(1.3272*(20.-TC)-.001053*(TC-20.)*.02)/(TC+105.)
VIS=(ABSV20**10.*X)**0.00672
RETURN
END

SUBROUTINE EQUIVL(IGATE,ITEE,ILONGC,ISTC,NGLATE,NLONGC,NSTC,
1 EG,ET,ELC,ESTC,EQIN,EQOUT)
THIS SUBROUTINE CALCULATES THE EQUIVALENT LENGTHS OF PIPE
CORRESPONDING TO THE ELBOWS,GATES, AND TESS IN THE INLET AND
OUTLET PIPES. IT RETURNS 2 VARIABLES, 1 FOR INLET AND 1 FOR
THE OUTLET TO BE USED IN CALCULATING THE FRICTION IN THE PIPES.
EQIN=IGATE*EG+ITEE*ET+ILONGC*ELC+ISTC*ESTC
EQOUT=NGLATE*EG+NLONGC*ELC+NSTC*ESTC
RETURN
END

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<th>COLLECTOR OUTLET (°F)</th>
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**TOTALS FOR THE DAY:** 73330.4 30286.7 41.2

**SYSTEM EFFICIENCY FOR THE DAY:** 48.7%
FOOTNOTES


12 Duffie and Beckman, Solar Energy Thermal Processes, p. 15.


14 Bliss, "The Derivation of Several 'Plate Efficiency Factors' Useful in the Design of Flat Plate Solar Heat Collectors," p. 60.


18 Ibid., p. 382.

19 Ibid., p. 383.


21 Chemical Rubber Company Handbook of Chemistry and Physics, 49th ed., s.v. "The Viscosity of Water from 0°C to 100°C."


26. Ibid., p. 382.


29. Ibid., p. 122.


32. Ibid., p. 127.


Chemical Rubber Company Handbook of Chemistry and Physics, 49th ed. S.v. "The Viscosity of Water from 0°C to 100°C."


Klein, S. A.; Duffie, J. A.; and Beckman, W. A. "Transient Considerations of Flat Plate Solar Collectors." Transactions of the American Society of Mechanical Engineering Journal of


