This thesis is concerned with simulating two forms of energy retrieval. Standard computer programs are routinely used to simulate such energy systems. However, the accuracy of these simulations is not well documented. The main objective of the work presented here is to evaluate the validity of this type of simulation for the system considered.

The two forms of energy retrieval are (i) the collection of solar energy to provide heated water, and (ii) the utilization of waste heat from a 250 MVA transformer cooling oil circuit to heat a refrigerant in a heat pump circuit. The system is installed and operating at the Ross Substation near Vancouver, Washington.

The system provides space conditioning and operates in two modes, cooling and heating. In cooling, a solar-collector/absorption-cooling system provides the primary cooling with a heat pump as a backup. In heating, the heat pump provides the primary heating with a solar-
collector/storage-tank system as a supplement, and electrical resistance heating as a backup. Both heating and cooling modes of operation are simulated and the results are compared with measured data from the operating system.

Simulations of this system using basic parameters provided from design specifications gave results that compared poorly with results obtained from measured data. Calculations based on the measured performance indicated that many of these basic parameters were in error. Using parameters obtained from the measured data, simulations gave results which were close to measured values. The areas in which these parameter errors occurred are in (i) component performance of the solar collector and heat pump models, (ii) component control of the absorption cooler, and (iii) building heat gains. The conclusion reached, based on the simulation work done, is that accurate simulations can be obtained, but in order to achieve them, detailed knowledge of the system's operating characteristics is required.
Simulation of a Solar Heating and Cooling System with Experimental Evaluation

by

William Clifford Layton

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He would like to express his thanks to Dr. Milton B. Larson and Steve Fox, whose efforts assisted in the accomplishment of this project.

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<td>Area</td>
</tr>
<tr>
<td>$A_x$</td>
<td>Cross-sectional area</td>
</tr>
<tr>
<td>CAP</td>
<td>Thermal capacitance of the attic</td>
</tr>
<tr>
<td>Capn</td>
<td>Thermal capacitance of n equal volume segments in the collector</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Specific heat</td>
</tr>
<tr>
<td>d</td>
<td>Diameter; in reference to collector, the tube spacing</td>
</tr>
<tr>
<td>F</td>
<td>Fraction between 1 and 0</td>
</tr>
<tr>
<td>$F'$</td>
<td>Collector geometric efficiency</td>
</tr>
<tr>
<td>$f(\Omega)$</td>
<td>Function which accounts for the beam radiation reflected to the absorber tube</td>
</tr>
<tr>
<td>FR</td>
<td>Collector efficiency factor</td>
</tr>
<tr>
<td>$F_S$</td>
<td>Fraction of window that is shaded</td>
</tr>
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<td>$F_w$</td>
<td>Fraction of wall that is window</td>
</tr>
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<td>$G(\Omega)$</td>
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<td>h</td>
<td>Heat transfer coefficient</td>
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<td>H</td>
<td>Instantaneous total solar radiation per unit area on horizontal surface</td>
</tr>
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<td>$H_b$</td>
<td>Instantaneous beam radiation per unit area on a horizontal surface</td>
</tr>
<tr>
<td>$H_d$</td>
<td>Instantaneous diffuse radiation per unit area on a horizontal surface</td>
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<td>$H_0$</td>
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<td>$H_{BT}$</td>
<td>Instantaneous beam radiation per unit area on a tilted surface</td>
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H_{DT} \quad \text{Instantaneous diffuse radiation per unit area on a tilted surface}

H_{T} \quad \text{Instantaneous total radiation per unit area on a tilted surface}

L \quad \text{Length of collector tube}

\dot{m} \quad \text{Mass flowrate}

N \quad \text{Number of equal volume segments in collector}

P \quad \text{Perimeter}

P_{c} \quad \text{Perimeter of absorber tube based on absorber tube diameter}

Q_{\text{Gen}} \quad \text{Rate of internal energy generation}

Q_{\text{Infl}} \quad \text{Rate of energy transfer into room due to infiltration}

Q_{\text{Room}} \quad \text{Rate of energy flow from the attic to the control room}

Q_{\text{SHG}} \quad \text{Rate of solar heat gains}

Q_{\text{Walls}} \quad \text{Rate of heat transfer through walls}

R \quad \text{Radius}

R \quad \text{Rate of Air Changes per hour}

R_{b} \quad \text{Ratio of beam radiation incident on a tilted surface to that on a horizontal surface}

R_{d} \quad \text{Ratio of diffuse radiation incident on a tilted surface to that on a horizontal surface}

R_{r} \quad \text{Ratio of reflected radiation incident on a tilted surface to the total radiation or a horizontal surface}

S_{\text{eff}} \quad \text{Effective radiation incident on the absorber tube}

SST \quad \text{Saturated suction temperature (heat pump)}

t \quad \text{time}

T \quad \text{Temperature}

T_{\text{ent}} \quad \text{Temperature of fluid entering the evaporator (heat pump)}

T_{\text{ewt}} \quad \text{Temperature of the water entering the condenser (heat pump)}

T_{\text{hxo}} \quad \text{Temperature of the collector water exiting the dump heat exchanger}
\( T_N \) Temperature of the bottom segment in the storage tank

\( T_{\text{Room}} \) Control room temperature

\( U_A \) Overall heat transfer coefficient

\( U_L \) Overall loss coefficient of the collector tube assembly based on the outside absorber tube diameter

\( V_{\text{Vol}} \) Volume

**Greek Symbols**

\( \alpha \) Absorptivity

\( \Gamma_B \) Beam enhancement factor

\( \Gamma_d \) Diffuse enhancement factor

\( \varepsilon \) Infrared emissivity of exterior wall

\( \varepsilon_3 \) Infrared emissivity of absorber tube

\( \varepsilon_i \) Infrared emissivity of glass cover tube

\( \rho \) Density or reflectance in connection with solar radiation

\( \sigma \) Stefan-Boltzman constant

\( \tau \) Transmittance

\( \Omega \) The projection of the sun's beam radiation measured from vertical in a plane normal to the longitude axes of the collector tubes when the tubes are parallel to the collector array backing surface.

**Subscripts**

\( a \) Ambient

\( c \) Outside surface of cover tube

\( c_i \) Collector, in

\( c_o \) Collector, out

\( i \) Inside surface of cover tube

\( 1 \) Fluid within delivery tube (solar collector)

\( 2 \) Fluid within annulus (solar collector)

\( 3_i \) Inside surface of absorber tube

\( 3_o \) Outside surface of absorber tube
I. INTRODUCTION

The Prototype Energy Retrieval System (PER) was designed by the Bonneville Power Administration (BPA) to test and evaluate two types of energy retrieval systems. The first type of energy retrieval is collection of solar energy to provide hot water for an absorption cooler and for space heating. The second type of energy retrieval utilizes waste heat from a 250 MVA transformer to heat a refrigerant in a heat pump circuit. Figure 1 shows the block diagram for the PER system.

The two basic modes of operation are cooling and heating. In the cooling mode, the primary source of cooling is the solar-collector/absorption-cooler subsystem. In the event the primary source is unable to fulfill the cooling load, the heat pump is operated in a cooling mode as a backup system. In the heating mode, the heat pump is the primary source with the solar-collector/storage-tank as a supplement, and electric resistance heating as a backup.

Standard computer programs are routinely used to simulate such energy systems. However, the accuracy of these simulations is not well documented. The computer program used in the simulation work presented here is TRNSYS [1], version 8.2. TRNSYS, developed by the Solar Energy Laboratory, University of Wisconsin-Madison, was designed to solve transient system simulations involving solar collectors.
Figure 1. Prototype energy retrieval system block diagram.
The results of the TRNSYS simulation of the PER system are compared with measured values for a period of several days to determine the accuracy of the simulation.

To insure the validity of the simulation work, weather data as collected by the PER data collection system was processed for use in the simulation program. In addition to collecting weather data, the data collection system records component performance data which are used in evaluating the system's performance. Appendix A outlines the data collection system and the method used to process the weather data.

Chapter II of this thesis presents computer models of specific components for addition to those in TRNSYS for computer simulation. In addition, the simulation results, and comparisons of these results with the performance of the actual PER system are presented. Chapter III contains the conclusions of this thesis.
II. COMPUTER MODELING AND SIMULATION

Simulation of a system as complex as the PER system with TRNSYS reduces the complexity of simulation by modeling it as a set of interconnected components. This technique takes advantage of the fact that most systems are in essence a collection of separate components, characteristic fixed parameters, and time dependent forcing functions. This method is also known as modular simulation.

The lack of suitable subroutines in the available TRNSYS package, to model some of the components in the PER system, necessitated formulation of mathematical models in the form of Fortran subroutines to adopt for use in TRNSYS. These include models for (i) the solar collector, (ii) the absorption cooler, and (iii) the heat pump. The complexity of the conditioned space at the control house also required that load models in addition to those supplied with TRNSYS be formulated to simulate the heating and cooling requirements of the space. Unless specified otherwise, SI units are utilized in this thesis.

This chapter consists of two main sections: (i) model development in which the assumptions and formulation of models adapted to TRNSYS are presented, and (ii) the computer simulation of the PER system and comparison of these results with the actual PER system. The computer code for the Fortran subroutines developed are found in Appendix B.
The load model developed is designed to model the control house at the Ross Substation in Vancouver, Washington. The floor plan of the control house is as shown in Figure 2.

The control room is the only room conditioned by the PER system, as such, it is the only area of interest. Previous simulations [2] have assumed the heat transfer through the north wall of the control room to be proportional to the difference between the room temperature and the ambient temperature. It is felt that this method overestimates the actual heat transfer to such an extent that assuming zero heat transfer would give less error. If it is assumed that the north wall of the control room is insulated, the remaining space is represented as three separate areas: attic, control room and basement. Figure 3 shows the control room model.

In modeling the load for the control room, the heat transfer from the attic, walls, and basement are required. Subroutines supplied with TRNSYS are used to calculate the heat transfer from the walls and basement. Heat transfer from the attic is a special case and requires an additional model to calculate its input.

Walls and Roof. The heat transfer resulting from conduction heat gains is calculated by the transfer function method as developed in the ASHRAE Handbook of Fundamentals [3]. In general, the heat flux $q_o$, into or out of a room is calculated by:
Figure 2. Configuration of the Ross Control House of Vancouver, Wash.
Figure 3. Conditioned space model.
\[ q^n_o = \sum_{n=0}^{\infty} b_n (T_{sa,n} - T_{Room}) \sum_{n=1} d_n q^n_n \] (1)

where \( b_n \) are transfer function coefficients of temperature terms

\( d_n \) are transfer function coefficients of heat flux terms

\( T_{sa,n} \) is the sol-air temperature of the wall or roof surface at time \( n \)

\( q^n_n \) are the values of the heat fluxes at time \( n \)

The coefficients, \( b_n \) and \( d_n \) are obtained from the ASHRAE Handbook of Fundamentals [3]. The total conduction is then:

\[ \dot{Q}_{Walls} = q^n_o (1 - F_w) A \] (2)

Windows. Solar heat gains are calculated by:

\[ \dot{Q}_{SHG} = F_w (1 - F_s) \tau A h_T \] (3)

where \( A \) is the total area of the wall. Conduction heat gains through the windows are calculated by:

\[ \dot{Q}_{windows, cond} = U F_w A (T_a - T_{Room}) \] (4)

Attic. The attic is modeled as a single node space whose temperature is \( T_{Attic} \). From the control volume of the attic as shown in Figure 4, an energy balance gives:

\[ \text{Cap} \frac{dT_{Attic}}{dt} = \dot{Q}_{Roof} + \dot{Q}_{Walls} - \dot{Q}_{Room} + \dot{Q}_{Infl} \] (5)

The heat flux from the roof and walls, \( \dot{Q}_{Roof} \) and \( \dot{Q}_{Walls} \) respectively, are calculated by the previously considered transfer function method. The heat flux to the room is given by:

\[ \dot{Q}_{Room} = UA (T_{Attic} - T_{Room}) \] (6)
Figure 4. Energy flows into and out of the attic
The infiltration loss is calculated by:

\[ Q_{\text{Infl}} = \text{Rate} \times \text{Vol} \times 1.2185(T_a - T_{\text{Attic}}) \]  

(7)

Room Model. The control room is modeled as a single node space in a manner similar to that of the attic. The TRNSYS subroutine calculates the heat gains from the basement, infiltration, and internal generation from occupants, lights, appliances, etc. Certain heat gains such as solar and electrical heat gains do not immediately become part of the cooling load, but are time distributed. Such time lags are accounted for in the TRNSYS model by the ASHRAE transfer function method. Table 1 lists specific areas and heat transfer data for the building components.

Collector Performance Model

The PER system utilizes the Owens-Illinois tubular collector. The unique features of this type of collector are its large capacitance and the use of a vacuum seal to reduce heat losses. Figure 5 shows a typical cross section of a single tube. As TRNSYS does not contain a model to simulate this type of collector, formulation of a suitable model was required.

Assumptions Made in Modeling the Collector Performance. To account for the high capacitance of the collector and to simplify the calculations, the following assumptions are made:

1. The thermal capacitance of the glass tubes is neglected as small when compared with the heat capacity of the water.

2. The temperature difference across the glass absorber tube is negligible \((T_{30} = T_{3i} = T_3)\).
Table 1. Area and heat transfer data for building components of control room.

<table>
<thead>
<tr>
<th></th>
<th>S-Wall</th>
<th>W-Wall</th>
<th>E-Wall</th>
<th>Roof</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area of wall (m²)</td>
<td>34.92</td>
<td>38.10</td>
<td>38.10</td>
<td>213.4</td>
</tr>
<tr>
<td>Area of windows (m²)</td>
<td>13.91</td>
<td>23.19</td>
<td>23.19</td>
<td>0</td>
</tr>
<tr>
<td>α</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
<td>0.8</td>
</tr>
<tr>
<td>E</td>
<td>0.92</td>
<td>0.92</td>
<td>0.92</td>
<td>0.9</td>
</tr>
</tbody>
</table>

UA of building: 2494.5 KJ/hr °C
Capacitance of Control Room: 1900 KJ/°C
Floor Area: 189.05 m²  Volume: 691.47 m³
Ventilation Rate: 0.49 Volume Changes/hr

= 5.66 m³/min

bₙ and dₙ coefficients:

<table>
<thead>
<tr>
<th>Roof:</th>
<th>n=0</th>
<th>n=1</th>
<th>n=2</th>
<th>n=3</th>
<th>n=4</th>
<th>n=5</th>
</tr>
</thead>
<tbody>
<tr>
<td>bₙ</td>
<td>1.3x10⁻³</td>
<td>2.93x10⁻²</td>
<td>3.36x10⁻²</td>
<td>4.0x10⁻³</td>
<td></td>
<td></td>
</tr>
<tr>
<td>dₙ</td>
<td>1</td>
<td>-0.8134</td>
<td>0.137</td>
<td>-3.1x10⁻³</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Walls:

| bₙ            | 1.45x10⁻⁴| 7.73x10⁻³| 1.867x10⁻²| 5.98x10⁻³| 2.59x10⁻⁴| 1.0x10⁻⁶|
| dₙ            | 1        | -1.29    | 0.4079   | -3.331x10⁻²| 5.78x10⁻⁴|         |
Figure 5. Cross section of Owens-Illinois collector tube. (Owens-Illinois Sunpack® Instalation Manual)
3. Axial conduction is neglected.

4. All heat transfer coefficients and physical properties are independent of length.

5. Absorption of solar radiation by the cover tube is negligible.

6. Solar radiation absorbed by the outer surface of the absorber tube is uniformly distributed around the tube perimeter.

7. The loss coefficient, $U_L$ is a function of $T_3$, the absorber tube mean temperature, and $T_a$, the ambient temperature [4]. The loss coefficient is given by:

$$
U_L = \frac{(T_3^2 + T_a^2)(T_3 + T_a)}{\frac{1}{R_i} - \frac{1 - \varepsilon_3}{R_{30}} - \frac{1 - \varepsilon_i}{R_{30}} + \frac{1}{F_{i-3}}} \left(\frac{R_i}{R_{30}}\right)
$$

where $F_{i-3}$, the view factor from the inside cover tube surface to the absorber tube, is equal to:

$$
F_{i-3} = \frac{R_i}{R_{30}}
$$

8. The delivery tube is insulated [5]. This assumption holds for $\dot{m} > 4.5$ kg/hr per collector tube.

**Energy Balance on Annulus.** Equations (9) and (10) are derived from the energy balance on the control volumes involving the absorber tube and annulus respectively as shown in Figure 6:

$$
\alpha T \left(S_{eff} \rho c - U_L P_{30} (T_3 - T_a) - h_{3i} (T_3 - T_2) \right) = 0
$$

$$
\rho A \frac{dT_2}{dt} = - \dot{m} c_p \frac{dT_2}{dx} + h_3 P_{3i} (T_3 - T_2)
$$
Figure 6. Control volume of collector tube.
Solving equation (9) for $T_3$ and substituting into equation (10), yields

$$\rho A_x C_p \frac{dT_2}{dt} = -\dot{m}C_p \frac{dT_2}{dx} + \alpha \tau P_c' S_{\text{eff}} - U_L P_{3o} F' (T_2 - T_a) \quad (11)$$

where $F' = \frac{h_3 P_{3i}}{h_3 P_{3i} + U_L P_{3o}}$

By dividing the tube into $N$ segments, $\frac{dT_2}{dx}$ can be approximated by:

$$\frac{dT_2}{dx} = \frac{T_2(n) - T_2(n - 1)}{L/N} \quad (12)$$

where each segment is considered at a constant temperature, $T_2(n)$.

If the approximation for $\frac{dT_2}{dx}$ in equation (12) is substituted into equation (11), then the result is:

$$\rho \frac{LA_x C_p}{N} \frac{dT_2}{dt} = \dot{m}C_p (T_2(n - 1) - T_2(n)) + \alpha \tau P_c \frac{L}{N} F' S_{\text{eff}}$$

$$- U_L P_{3o} \frac{L}{N} F' (T_2(n) - T_a) \quad (13)$$

Now let $C_{np} = \rho LA_x C_p/N$. Equation (13) becomes:

$$\frac{dT_2(n)}{dt} = \dot{m}C_p (T_2(n - 1) - T_2(n)) + \alpha \tau P_c \frac{L}{N} F' S_{\text{eff}} \quad (14)$$

Equation (14) is solved numerically by use of a first-order predictor-corrector algorithm using Euler's Method for the predicting step and the trapezoid rule for the correcting step. All differential equations in TRNSYS are solved by this method. The collector tube dimensions and surface properties that were used in solving equation (14) are shown in Table 2.
Table 2. Collector tube dimensions and surface properties.

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cover Tube Inside Diameter</td>
<td>49 mm</td>
</tr>
<tr>
<td>Absorber Tube Outside Diameter</td>
<td>43 mm</td>
</tr>
<tr>
<td>Absorber Tube Inside Diameter</td>
<td>39 mm</td>
</tr>
<tr>
<td>Delivery Tube Outside Diameter</td>
<td>23 mm</td>
</tr>
<tr>
<td>Collector Tube Length</td>
<td>1.067 m</td>
</tr>
<tr>
<td>Selective Surface Properties</td>
<td>$\alpha = 0.85; \varepsilon = 0.07$</td>
</tr>
<tr>
<td>Cover Tube Transmittance</td>
<td>$\tau = 0.92$</td>
</tr>
</tbody>
</table>

Insolation on Tubes in an Array. The term $S_{\text{eff}}$ is defined as the effective insolation on the absorber tube based on the cross sectional area of the absorber tube. In terms of $H_{\text{BT}}$ and $H_{\text{DT}}$, $S_{\text{eff}}$ has been determined [6] as:

$$S_{\text{eff}} = \left[ \frac{G(\Omega)}{\cos(\Omega)} + \rho \frac{d}{d_0} f(\Omega) \right] H_{\text{BT}} + \Gamma d H_{\text{DT}}$$  \hspace{1cm} (15)

The factor $G(\Omega)$ is a shading factor equal to unity for $|\Omega| > |\Omega_S|$, $G(\Omega)$ is approximated by:

$$G(\Omega) = \frac{d}{d_c} \cos(\Omega)$$  \hspace{1cm} (16)

Assuming that shading is determined by the cover tube rather than the absorber tube, the shading angle, $\Omega_S$, is given by:

$$\Omega_S = \cos^{-1} \left( \frac{d_c}{d} \right)$$  \hspace{1cm} (17)

In equation (15), the quantity $f(\Omega)$ accounts for the beam radiation which is ultimately reflected to the absorber tube.
For a cylindrical reflector as installed in the PER system, \( f(\Omega) \) is given by:

\[
f(\Omega) = \begin{cases} 
1 - \frac{d_{30}/d}{\cos(\Omega)} & |\Omega| \leq |\Omega_s| \\
0 & |\Omega| > |\Omega_s| 
\end{cases}
\]  

(18)

providing all rays incident on the reflector are intercepted by the absorber tube. In general this is not the case and evaluation of \( f(\Omega) \) is carried out by ray tracing techniques. Comparison of the two methods is shown in Figure 7. Three separate tube spacings are presented; the one of concern is the set of curves for \( d = 2d_c \) (curve set [2]).

The diffuse enhancement factor, \( \Gamma_d \), is given by:

\[
\Gamma_d = \int_{0}^{\pi/2} \cos(\Omega) \, \Gamma_B \, d\Omega
\]  

(19)

From equation (15), the beam enhancement factor, \( \Gamma_B \), is given by:

\[
\Gamma_B = \frac{G(\Omega)}{\cos(\Omega)} + \rho \frac{d}{d_{30}} \, f(\Omega)
\]  

(20)

Calculation of \( \Gamma_d \) is obtained by numerical integration of equation (19) with \( \Gamma_B \) obtained from equation (20). The value of \( \Gamma_d \) calculated in this manner is 2.04. The specular reflectance was taken as \( \rho = 0.85 \), which is a typical value for the reflectors that have been tested by Owens-Illinois.

**Heat Pump Model**

The heat pump in the PER system is designed to utilize waste heat from a 250 MVA transformer cooling oil circuit for space heating (Figure 8). During the cooling season, the heat pump supplements solar
Figure 7. $f(\Omega)$ factors for the cylindrical reflector as determined by ray trace and equation 19. (Beekley and Mather [4].)
cooling when required. Figure 9 shows the schematic diagram of the heat pump and the direction of refrigerant flow for both cooling and heating modes.

The heat pump is basically a combination of a compressor, condenser, evaporator, and refrigerant flow controls as required. Determination of the performance of the components as a system involves elimination of the internal variables by the process of simultaneously solving a series of equations, each describing the performance of an individual component. As these equations may be very complex, a graphical technique is preferred. Just such a method is outlined in the ASHRAE Handbook of Fundamentals and Equipment [7]. The method described is called component balancing. Details of this method can be found in Appendix C.

Component balancing performance characteristics for the PER system heat pump, in the cooling and heating modes, are shown in Figures 10 and 11 respectively. In each mode, the performance is predicted on the basis of two inputs being specified: (i) the temperature of the water entering the condenser and (ii) the temperature of the air (or water in the cooling mode) entering the evaporator. Calculation of the performance curves for the heat pump in the cooling mode is detailed in Appendix C.

Absorption Cooler

Although TRNSYS in its present form has a component model of an absorption cooler, the model is unsuitable for simulation of the PER
A. Conceptual top view of transformer and present heat exchange system.

B. Conceptual side view of heat exchange system.

Figure 8. Schematic representation of transformer and heat exchange system.
HEAT PUMP REFRigerATION SYSTEM

Figure 9. Heat pump refrigeration system.
Figure 10. Heat pump performance in cooling mode.
Figure 11. Heat pump performance in heating mode.
system absorption cooler because it contains an auxiliary cooling system in conjunction with an energy/degree-day cooling load. Since an auxiliary cooling system was not desired nor was the energy/degree-day method of load calculation adequate, the performance test results obtained by Carrier on the absorption cooler, prior to shipping and installation in the PER system, were used to develop a suitable model for the absorption cooler. The performance curves are shown in Figure 12.

The absorption cooler as supplied by Carrier (Figure 13) for the PER system is designed to supply 15 tons cooling capacity with 180°F solar heated water, 85°F condensing water, and 45°F leaving cooling water. For a constant evaporator temperature, the performance curves for the absorption cooler are a function of the solar heated water and condensing water temperatures. If the condensing water temperature is assumed constant, the performance is a function of the entering hot water temperature only.

The condensing water temperature at the test site is approximately constant (65°F ± 5°F), but test data for condensing temperatures less than 75°F were unavailable. Since the performance of an absorption unit after a period of "off time" is reduced as a function of the off time [8], the performance of the absorption unit was described as:

\[ \text{Cooling capacity (tons)} = 0.3(T_{hx0}) - 34.41 \]  

(21)

which is an approximation of the test curve for condensing water at 75°F. The performance predicted by equation (21) will be somewhat less than the actual steady state performance at the test site; however, it should represent an average performance of the absorption unit.
Figure 12. Absorption cooler performance as tested by Carrier Corp.
Figure 13. Absorption cooler flow diagram.
Controller

The purpose of a controller is, as its name implies, to control the system components. The application of a controller in a solar energy system is usually limited to generating a control function, $\gamma_0$, which assumes values of 0 or 1, to turn off or on respectively, a specific component.

The value of a control function is determined by the value of the difference between two inputs, and the magnitude of each input. In a solar energy system, the inputs are generally temperatures. As such, define:

- $\Delta T_1$ - upper dead band temperature difference
- $\Delta T_2$ - lower dead band temperature difference
- $T_1$ - upper input temperature
- $T_2$ - lower input temperature
- $\gamma_i$ - input control function
- $\gamma_0$ - output control function

Comparison of $(T_1 - T_2)$ to $\Delta T_1$ or $\Delta T_2$ is determined by the value of $\gamma$ from the previous time-step, in this case $\gamma_i$. In this manner, a hysteresis effect is possible. Mathematically, the relations are expressed as:

- $\gamma_i = 1$ and $\Delta T_2 \leq (T_1 - T_2)$, $\gamma_0 = 1$
- $\gamma_i = 1$ and $\Delta T_2 > (T_1 - T_2)$, $\gamma_0 = 0$
- $\gamma_i = 0$ and $\Delta T_1 \leq (T_1 - T_2)$, $\gamma_0 = 1$
- $\gamma_i = 0$ and $\Delta T_1 > (T_1 - T_2)$, $\gamma_0 = 0$
Hysteresis in simulation work not only adds stability to the program but also adds a measure of reality. Hysteresis provides stability from time step to time step. To provide stability from iteration to iteration within a given time step, a user can "stick" the controller after a given number of calls (NSTK) to the controller. The control functions may be in either the "on" or the "off" position for NSTK calls to the unit. After the NSTK call, the control functions remain in the same mode, for the remainder of the time step. It is possible that a component will be "stuck" in the wrong mode for the time step which consequently results in an error. These errors, however, tend to cancel out over a period of many time steps.

The controller in the PER system operates in 22 different modes as shown in Table 3. The first 13 modes represent operation of the collector and absorption subsystems. The objective of these modes is to fill the storage with chilled water from the absorption unit in the summer, to fill the storage with hot water in the winter, and to dump surplus heat as required. Modes 14 through 22 represent the operating modes associated with space conditioning. This involves circulating hot or cold water from storage to the fan coil unit or from the heat pump to the fan coil unit. Figure 14 represents the control logic diagram for the system operating in the cooling mode. The mode in which the system is operating can be determined from the circled numbers in the diagram.
Table 3. Modes of operation for PER system.

<table>
<thead>
<tr>
<th>Mode No.</th>
<th>Mode Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Stagnation ($T_{co} &gt; T_{ci} + 100$)</td>
</tr>
<tr>
<td>2</td>
<td>OFF (Insufficient Insolation ($T_{co} &lt; T_{ci}$))</td>
</tr>
<tr>
<td>3</td>
<td>Absorption Cooling ($T_{co} &lt; T_{ci}$) ($T_{hxo} &gt; 190^0 F$)</td>
</tr>
<tr>
<td>4</td>
<td>Absorption Cooling ($T_{co} &gt; T_{ci}$) ($T_{hxo} &gt; 190^0 F$)</td>
</tr>
<tr>
<td>5</td>
<td>Increasing Solar ($T_{co} &gt; T_{ci}$) ($T_{hxo} &lt; 190^0 F$)</td>
</tr>
<tr>
<td>6</td>
<td>Absorption Cooling and Dump ($T_{co} &gt; T_{ci}$) ($T_{co} &gt; 210^0 F$)</td>
</tr>
<tr>
<td>7</td>
<td>Dump ($T_{co} &gt; T_{ci}$, $T_{co} &gt; 210^0 F$, $T_{hxo} &lt; 190^0 F$)</td>
</tr>
<tr>
<td>8</td>
<td>Stagnation ($T_{co} &gt; T_{ci} + 100$)</td>
</tr>
<tr>
<td>9</td>
<td>OFF (Insufficient Insolation ($T_{co} &lt; T_{ci}$))</td>
</tr>
<tr>
<td>10</td>
<td>Dump ($T_{co} &gt; T_{ci}$, $T_N &gt; 160$)</td>
</tr>
<tr>
<td>11</td>
<td>Heat Storage ($T_{co} &gt; T_{ci}$, $T_N &lt; 160$)</td>
</tr>
<tr>
<td>12</td>
<td>Dump ($T_{co} &gt; T_{ci}$, $T_{co} &gt; 210$, $T_N &gt; 160$)</td>
</tr>
<tr>
<td>13</td>
<td>Heat Storage ($T_{co} &gt; T_{ci}$, $T_{co} &gt; 210$, $T_N &lt; 160$)</td>
</tr>
<tr>
<td>14</td>
<td>Fan Only</td>
</tr>
<tr>
<td>15</td>
<td>OFF</td>
</tr>
<tr>
<td>16</td>
<td>Heating (Satisfied)</td>
</tr>
<tr>
<td>17</td>
<td>Heating (Heat Pump)</td>
</tr>
<tr>
<td>18</td>
<td>Heating (Solar and Heat Pump Assist)</td>
</tr>
<tr>
<td>19</td>
<td>Heating (Solar)</td>
</tr>
<tr>
<td>20</td>
<td>Cooling (Satisfied)</td>
</tr>
<tr>
<td>21</td>
<td>Cooling (Heat Pump)</td>
</tr>
<tr>
<td>22</td>
<td>Cooling (Solar)</td>
</tr>
</tbody>
</table>
Figure 14. Control logic for cooling mode operation.
Computer Simulation

Actual weather data collected by the PER data collection system were used as input data for the computer simulation. The data collection system and the method used to process the weather data are described in Appendix A. In general, the weather related data include the (i) incident radiation, (ii) ambient temperature, (iii) wind speed, and (iv) temperature of the air exiting the oil-air heat exchanger on the transformer. In addition, data regarding internal generation in the conditioned space are required. Figure 15 shows time dependent internal heat gains within the control room. The data are from BPA specifications and are entered into TRNSYS via time-dependent forcing functions. A constant internal heat gain arising from the switching gear load is also specified as 3kW.

The schematic diagram for the TRNSYS model of the PER system in the cooling mode is shown in Figure 16. Modifications to the cooling system diagram to allow modeling the PER system in the heating mode are shown in Figure 17.

Cooling Mode Simulation

The period of simulation for the cooling mode is from 2400 August 2, 1978 to 2400 August 13, 1978. However, insufficient data collection by the PER system precluded simulation for days seven and eight. Thus there is a lack of data points for this interval. The data set for day six was also incomplete, lacking data in the evening from 1800 to 2400. Data on day eight started at 1800 so the data set for day six
Figure 15. Time dependent heat gains within the control room.
Figure 16. Schematic diagram of PER system in cooling mode for Transys simulation.
Figure 17. Modifications to Figure 16 to model PER system in heating mode.
was supplemented with data from day eight during the hours of 1800 to 2400. In this manner, continuity within the data set was maintained.

Two separate simulation models of the PER system were run with TRNSYS. The initial simulations (data points indicated by "TRNSYS") were run with parameters based on manufacturer specifications and BPA estimates. A complete list of the computer input for the initial simulation is shown in Appendix D. As will become apparent, some of these parameters were in error and were altered to determine their influence on the simulated performance. A final model with the modifications was run for the same period as the original for comparison. Data points reflecting the modified simulation are indicated by "Modified TRNSYS".

Initial Simulation. The results of the initial simulation (before modification) are shown in Figures 18 to 23. This section discusses these results.

Of particular interest was the performance of the solar collector. The simulated and measured daily performance characteristics of the collector are shown in Figures 18 and 19 while Figure 20 shows hourly performance for August 3, 1978. Related to the collector performance is the absorption cooler performance shown in Figure 21.

From Figure 18, the measured PER collector efficiency is seen to be 10 to 15 percentage points below the efficiency observed for the TRNSYS simulated collector model. A similar difference in performance is noted in Figure 19 where the daily collected energy is compared. Possible reasons for this discrepancy between the TRNSYS values and the measure values on a daily basis are:
Figure 18. Daily collector efficiency.
Figure 19. Daily collected energy.
1. Dirt on the reflectors and/or cover tubes, (This has been observed to reduce efficiency by 5. to 10 percent.)

2. Defective collector tubes; (The total number of defective tubes is thought to be small and therefore so is the influence.)

3. Unbalanced flow among the tube banks; (This could cause possible poor performance in some of the tube banks as well as an error in measurement.)

Figure 20 presents the hourly collected energy for the TRNSYS simulation and for the PER measurement system, with the solar radiation that was incident on the tube banks illustrated for reference. The simulated performance is seen to follow the same form as the incident radiation but reduced in magnitude. The measured values seem to follow no exact pattern, with several spikes in the performance curve. The difference in the shape of the performance curves is attributed to the method of calculating the instantaneous value of \( \dot{Q} \).

The PERDATA calculation of the collected energy is the integral of the instantaneous value of \( \dot{Q} \), as calculated from equation (22):

\[
\dot{Q} = \dot{m}C_p(T_{co} - T_{ci})
\]

The TRNSYS simulation calculates the collected energy in two ways. The first method is the same as used in program PERDATA. The second method calculates \( Q \) from equation (23):

\[Q = \int \dot{Q} \, dt\]

\[Q = \int \dot{m}C_p(T_{co} - T_{ci}) \, dt\]

This information was obtained from a conversation with Val Diaga of Owens-Illinois. He also related that other sites that have installed Owens-Illinois collectors are experiencing daily efficiencies of 50-55 percent; values that correspond closely to the daily efficiencies as calculated by the simulation.
\[ \dot{Q} = A_{\text{eff}} \frac{2R}{d} \frac{3 \alpha}{\pi} FR (\alpha \tau_{\text{eff}} - \pi U_L) \] (23)

As discussed previously, the collected energy is the integrated value of \( \dot{Q} \). Only the results of the second method are shown in Figure 20 for the simulation.

The difference in the hourly value of the collected energy as calculated by the two methods is the result of (i) the fluid residence time in the collector, and (ii) the operation of the absorption cooler. The collected energy, when calculated by equation (22), is dependent upon the difference in temperature of the fluid (water) leaving and entering the collector. Not considered in this calculation is the energy which has gone into increasing the temperature of the water and collector tube. A collector which has no load will have an outlet temperature equal to the inlet temperature. The collected energy as calculated in equation (22) for this situation will be zero. The collected energy as calculated by equation (23) is influenced primarily by physical properties of the collector and the incident radiation. Thus, the collected energy calculated by equation (23) will be non-zero for the same collector mentioned above.

In the cooling mode, energy is removed from the collector loop when the absorption unit is operating, but when the absorption unit is off, the collector is in a closed loop with no load. Thus, when the absorption unit is operating, the temperature difference across the collector is relatively large and the collected energy is large [as calculated by equation (22)], but when the absorption unit is off the collected energy as calculated by equation (22) is almost zero. Comparison of the collector and absorption cooler performances for the PER...
Figure 20. Collected energy as calculated by PERDATA and that calculated by the TRNSYS simulation with the total incident radiation at the same angle.
system, shown in Figures 20 and 21 respectively, confirm the influence of the absorption cooler operation on the measured solar collector performance. The "spikes" in the collector performance correspond to the same "spikes" at the same time in the absorption cooler performance.

The conditioned space temperatures from the TRNSYS simulation and the measured values are compared in Figure 22. The simulation space temperature is observed to be 3-4°C cooler than the measured value. Since the room temperature as calculated by the simulation is lower than the measured temperature, the cooling supplied by the simulation should be greater than the cooling supplied the actual system. In Figure 23 the cooling load supplied by the simulation is compared with the measured values. The cooling supplied as calculated by the initial simulation is not greater than the measured value, but is observed to be significantly less. This discrepancy indicates that the heat gains in the TRNSYS load model are too small.

The initial simulation showed major discrepancies in the predicted performance of the PER system. In particular, the performances of the solar collector and the load model were found to be in error. To improve the simulation of the PER system, additional simulation work was required.

**Modified Simulation.** In the "modified simulation" parameter adjustments were made for the solar collector, load, and controller models. This section discusses how the adjustments were made and the results of the modified simulation.
Figure 21. Hourly absorption cooler performance.
Figure 22. Room temperature from PER system and the TRNSYS simulation of the PER system with the ambient temperature for comparison.
Figure 23. Hourly cooling supplied.
Due to the difference in collector performance between the measured and simulated performance, methods to account for this difference were considered. From the list of possible discrepancies noted in the previous section, the most logical discrepancy, plus the easiest to simulate, was the first: "Dirt on the reflectors and/or cover tubes." From equation (14) the product $\alpha \tau$, and from equation (15), $\rho$, are the two terms which could account for the reduced performance caused by dirt. Simulation runs were made, altering the values for these terms until the performance of the collector in the simulation approached that of the PER system for August 2, 1978, the first day of the simulation period. The original and modified values of the two terms are shown in Table 4. In Figures 18 and 19, the performance indicated by "Modified TRNSYS" is the result of this modification.

In Figure 18, the daily efficiency of the collector is presented, and in Figure 19, the daily collected energy. Both Figures show a marked improvement in performance prediction of the actual system by the modified simulation. The curves representing the modified simulation tend to follow the same form as the curves from the initial simulation, but are reduced in magnitude. Although the modification seems to be valid in light of the comparison with the measured values, the remaining discrepancy indicates that some factors have not been considered. A further discussion into this discrepancy is considered below.

The hourly performance of the collector is shown in Figure 24 for August 3, and in Figure 25 for August 5. The performance from August 5 is included to illustrate that the performance as noted for August 3 was typical. For both days it is noted that there is a significant
Figure 24. Collected energy as calculated by PERDATA and that calculated by the TRNSYS simulation with the total incident radiation at the same angle.
Figure 25. Collected energy as calculated by PERDATA and that calculated by the TRNSYS simulation with the total incident radiation at the same angle.
difference in the performance on the collector as predicted by the simulation (when equation [22] is used to calculate $\dot{Q}$) and as observed in the actual system during the afternoon. The difference are the spikes during the afternoon for the simulated collector performance compared to the absence of spikes for the measured performance. As discussed previously, these spikes are the result of the absorption cooler operation. Comparison of the absorption cooler performance for the modified simulation with measured values is presented in Figure 26. In Figure 26 it is seen that the performance of the absorption cooler is quite different between the simulation and actual system, particularly in the late afternoon. Data from the actual system show that the collector output temperature is quite high, yet the absorption unit is not on. An apparent error in the PER system absorption controller results in the collected energy continuing to raise the collector fluid in temperature during the afternoon, with this energy then being lost by heat transfer to the ambient during the night.

The apparent low output of the absorption cooler in the "TRNSYS" simulation relative to the PER system data shown in Figure 21 would indicate that a modification to the model is required. However, the low output of the TRNSYS simulation of the absorption cooler is the result of the storage tank having reached its lower cutoff temperature, and the absorption cooler shut off. The storage tank is quickly cooled to the cutoff temperature because very little space cooling is required in the initial TRNSYS simulation (see Figure 23). The small amount of cooling in the TRNSYS simulation is apparently the result of discrepancies in the load model (considered later).
Figure 26. Hourly absorption cooler performance.
The validity of the absorption cooler model as used in the simulation is difficult to determine since its performance is based on several external variables. These include: (i) the temperature of the water exiting the solar collector, (ii) the tank temperature, and (iii) the control of the absorption cooler. Since the control of the actual absorption cooler is apparently in error, no modifications were made to the simulation model.

To improve the load model of the control room, parameters that influence the load were investigated to determine their affect. These parameters include: (i) the ventilation rate (the rate at which room air is exchanged with the ambient), (ii) the capacitance of the control room, (iii) the controller parameters, and (iv) the internal load.

The ventilation rate and room capacitance were examined by altering the original values as given in Table 1 in simulation runs. Neither parameter affected the performance of the simulated system to a large extent when kept within reasonable limits. Thus these parameters were not changed for subsequent runs.

Examination of the hourly summaries\textsuperscript{2} indicated that the controller which determines whether cooling is from the storage tank or from the heat pump, was not operating as specified. Original specifications set the maximum temperature within the storage tank at which space cooling could be provided by the tank alone at 9°C. The data however, show that this maximum temperature is approximately 16°C. As a result, the modified simulation incorporated 16°C as this maximum tank temperature.

\textsuperscript{2}PERDATA output as described in Appendix A.
The change from 9° C to 16° C for this maximum temperature influences the cooling supplied in two ways: (i) overall cooling from the tank is increased, and (ii) the rate of cooling is decreased at increased tank temperatures.

The final parameter of interest is the internal energy generation from the switching gear. It was found from TRNSYS simulation, with (i) the collector performance reduced to approximately the value observed in the PER system and (ii) the controller operating as discussed in the previous paragraph, that there was still a large gap between the PER system cooling and the TRNSYS simulation cooling. The main parameter that could account for this large gap is the internal energy generation. Evidently, the value of this parameter as given by BPA specifications is too low. With the value of this parameter increased from 3 kW to 11 kW, the TRNSYS simulation compared reasonably well with measured values. Verification of this can be seen in Figure 27 during the period of 11 and 12 August in which the insolation was comparatively small. With low insolation, the heat gains resulting from the switching gear become the largest source of heat generation. Table 4 shows the original and modified parameters used in the "TRNSYS" and "Modified TRNSYS" simulations respectively. These are the values used for all results presented here. Only those parameters which were modified are shown.

The rather large cooling load during the evening (Figure 23) and the coinciding high room temperature (Figure 22) for the modified TRNSYS simulation are thought to be the result of the absorptivity of the wall's outside surface. Since the concrete walls have a large heat
capacity, a reduction in the absorptivity would reduce the radiation absorbed, thus reducing the heat stored in the walls. The expected result is a reduced cooling load during the evening.

The heat pump performance on a daily basis is shown in Figure 28 for the simulation period. Since the heat pump was not modified, it is interesting to note the difference in the amount of cooling supplied between the two simulation models. The initial simulation shows more cooling supplied by the heat pump than the modified simulation does. The reason for this is that higher tank temperatures are allowed in the modified model and thus there is greater tank cooling and consequently less heat pump cooling. To explain the comparatively large amount of heat pump cooling by the PER system relative to the simulation, an examination of the absorption cooling by the system is required. A previous discussion has pointed out the error in control of the absorption cooler. The amount of absorption cooling lost by the system because of this error, can be estimated from Figure 29. Figure 29 shows the absorption cooling for the initial and modified simulation, and the actual system. As seen in Figure 29, the amount of absorption cooling by the actual system is significantly less than by the modified simulation. The difference between the simulated and actual values represents the additional amount of absorption cooling obtainable by the actual system. The reduced amount of absorption cooling by the actual system results in the increased reliance on the heat pump to provide cooling.
Table 4. Original and modified parameters in the TRNSYS simulation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Original Value</th>
<th>Modified Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sigma$</td>
<td>Specular reflectance in collector model</td>
<td>0.85</td>
<td>0.7</td>
</tr>
<tr>
<td>$\alpha T$</td>
<td>Product of cover tube transmittance and selective surface absorbtivity in collector model</td>
<td>0.782</td>
<td>0.7</td>
</tr>
<tr>
<td>Tcool</td>
<td>Parameter in controller (see Appendix A) which represents the temperature of the storage tank, above which cooling is initiated by the heat pump.</td>
<td>8.89°C</td>
<td>14.5°C</td>
</tr>
<tr>
<td>$\dot{Q}_{\text{Gen}}$</td>
<td>Internal energy generation in control room</td>
<td>10,800 $\frac{kJ}{hr}$</td>
<td>40,000 $\frac{kJ}{hr}$</td>
</tr>
</tbody>
</table>

The validity of the modified simulation is reinforced by examination of the amount of energy that is wasted through the dump heat exchanger. Energy is wasted through the dump heat exchanger when the fluid from the collector approaches its boiling temperature. When solar energy is available, and the absorption cooler is shut off, this temperature is reached quickly. In the TRNSYS simulation, almost 58% of the collected energy was wasted, while the actual system wasted only 2.7%. In the modified simulation, with no changes in the absorption model, only 0.3% was wasted in the dump heat exchanger. These results are for the entire simulation period. Since the energy wasted is dependent upon the performances of the absorption cooler and the solar collector, the proximity of the wasted energy for the modified simulation and actual system indicates that the modified simulation is much improved over the initial simulation.
Figure 27. Total cooling supplied by the heat pump and storage tank.
Figure 28. Daily heat pump cooling delivered.
Figure 29. Daily absorption cooling.
Heating Mode Simulation

The period of simulation for the heating mode is from 0600 January 22, 1979 to 2400 February 1, 1979. The results of this simulation are presented in graphical form and are shown in Figures 30 to 37.

The lack of data collection by the PER system for a 24 hour period starting 1000 January 29 and ending 1000 January 30 required combining the data for these two days to keep continuity in the data file. The performance values for these two days are recorded on January 30 in the graphs for both the measured and simulated values. Only one model of the PER system was simulated in the heating mode. The primary reason for only one simulation model is that the actual system has an auxiliary electrical resistance heating unit whose power input is not measured at this time, making it impossible for an accurate comparison of the simulated and actual performance of the PER system in the heating mode. However, certain performance characteristics are possible to point out in spite of this problem. Component performance comparisons were made for the solar collector and heat pump models. Parameter modifications used in the solar collector and load models from the previous simulation work are used in the present simulation work as well.

The collector performance is shown in Figures 30, 31 and 32. The hourly performance as shown in Figure 30 reveals an unexpected result for the performance calculations based on equation (22) for both the simulated and actual performances of the collector. For these, significant energy collection is indicated until 2400. The conditions which make this possible are (i) the collector has a large capacitance and (ii) the fluid temperature out of the collector is greater than that of
Figure 30. Collected energy as calculated by PERDATA and that calculated by the TRNSYS simulation with the total incident radiation at the same angle.
Figure 31. Daily collector efficiency.
Figure 32. Daily collected energy.
the storage tank. The period during which conditions (i) and (ii) are met is from the late afternoon till 2400. The incident radiation for this period is zero; however, the fluid temperature is higher than the tank temperature as a result of the collected energy during the day. This indicated collected energy does not really represent an energy gain by the collector fluid but rather, an energy transfer from the collector fluid to the storage tank. The indicated collection occurs until 2400 because of poor heat transfer coupling between the collector loop and the storage tank. This poor coupling results in a time lag between when energy is collected in the collector and when it is transferred to the storage tank.

From calculations based on measured data, the effectiveness of the tank heater was determined to be only about 0.1. Calculations based on the NTU method for calculating the effectiveness indicate the effectiveness should have been about 0.5. The low value of the actual effectiveness is undoubtedly the cause of the time lag for the system to reach temperature equilibrium. No doubt if the effectiveness is improved, the system performance will improve.

There is still some discrepancy in the simulation of the PER collector. The actual system loses energy during the evening at a more rapid rate then the simulation (Figure 30). A cause for the increased rate may be a relatively large heat loss in the collector headers.

3 The effectiveness as calculated from measured data (0.1) was used in the simulation. The reasons 0.1 was used instead of the theoretical value, was to compare the simulated collector performance with the measured performance.
and the piping to and from the collector. This loss is not accounted for in the simulation. Another reason may be the capacitance value used in the simulation. A decrease in the capacitance would move the simulation values to the left, and increase the rate of energy loss. A similar effect would be observed if the effectiveness used in the simulation (0.1) was increased; however, an increase in the effectiveness is not considered to be valid since most values calculated from measured data were 0.1 or below.

The heat loss in the morning from the collector for the PER system is the result of the collector remaining on for the entire simulation period. The reason the collector remained on was to reduce the possibility of a freeze-up in the collector system. Mather [9], has determined that the low loss coefficient in the collector would prevent freeze-up even in severe conditions for several hours. The largest heat loss will be from the headers and pipes. As such, they will be the first areas to pose a threat of freeze-up. To prevent freeze-up in these areas, the collector fluid should be circulated when required. A continuous circulation of fluid is not necessary and reduces the over-all efficiency of the system. Monitoring the fluid temperature in the critical areas and basing the circulation of the collector fluid on these temperatures will reduce wasted energy.

In Figure 33 the simulated and actual room temperature are compared. The same effect is seen here as was seen in the cooling mode simulation, in which the room temperature is high compared to the actual room temperature during the evening. The cause may be the same as discussed previously: the absorptivity of the outside walls being too high,
Figure 33. Room temperature from PER system and the TRNSYS simulation of the PER system with the ambient temperature for comparison.
resulting in a large quantity of heat stored in the walls. An additional reason may be that the switching gear load is time dependent. With a reduced switching gear load during the afternoon and evening, the room temperature would drop to within the thermostat setting in the room.

From PERDATA summaries and simulation printouts for the period 0700 to 1100 January 31, it was noted that the heat pump was in operation for both the simulation and actual system. In Figure 34, the heat pump performance is shown for this time period and an obvious discrepancy between the predicted and actual performance is observed. The performance of the PER system heat pump is approximately half that as calculated by the simulation. The probable explanation for this difference is that the distance between the air-refrigerant heat exchanger and compressor is about 75 ft. This distance results in a rather significant heat loss from the refrigerant to the environment. In addition, a sizable pressure drop could result from friction losses in the long line. The problem of slugging of the compressor, encountered with the actual system indicated that the long line was a problem. A heating coil was added to the accumulator (shown in Figure 9) to prevent two-phase flow into the compressor. Daily heat pump performance is shown in Figure 35.

The daily heating supplied as measured by the PER system is shown in Figure 36. A comparison with the simulated values indicates a major discrepancy. From January 25 through the 29, the PER system indicates that no space heating was supplied. In contrast, the simulation provided heating for all four days. As previous discussion indicated,
Figure 34. Hourly heating supplied.
Figure 35. Daily heating supplied by the heat pump.
Figure 36. Total heating supplied by the heat pump and storage tank.
the electrical resistance heating was not measured. A major cause for this discrepancy is undoubtedly the unmeasured heating.

The portion of heat supplied by solar heating is shown in Figure 37. The overall low heating supplied by solar heating is caused, in part, by the combination of cold days and low insolation days prior to January 31 (see Figure 34).

In summary, the results of the heating mode simulation were utilized to compare component performance of the PER system solar collector and heat pump. The simulation of the collector was fairly accurate in predicting the shape of the actual performance curve. The low effectiveness of the tank heater inhibited energy transfer from the collector loop to the storage tank. The low effectiveness could also account for the difference in the predicted performance of the collector relative to the measured performance.

The heat pump model was not as accurate for simulation in the heating mode as in the cooling mode. The primary reason for the inaccuracy was because of the influence of the pipe length on the refrigerant's thermodynamic state. The affect of the pipe length was not considered in the simulation model.
Figure 37. Daily heating supplied by the storage tank.
III. CONCLUSIONS

Computer simulation of the PER system was accomplished with TRNSYS for two modes of operation, cooling and heating. To properly simulate the PER system, several changes and additions to TRNSYS subroutines were made. The models either changed from those available with TRNSYS or developed from basic information for this simulation work are: (i) the control room load, (ii) the solar collector, (iii) the heat pump, (iv) the absorption cooler, and (v) the controller.

The results of the initial simulation of the PER system in the cooling mode compared poorly with the measured values. The inaccuracy of the initial simulation was due in part to errors in the basic parameters used in the simulation. To improve the simulation of the PER system, adjustments were made on the simulation parameters. The parameter modifications were based on calculations using measured values and empirical methods. The most significant results of the modifications pointed out that (i) the controller of the PER system was not operating as specified, (ii) the estimated switching gear load of 10,800 kJ/hr is in error and the load is approximately 40,000 kJ/hr, and (iii) the observed collector performance is about 10 to 15 percentage points less than predicted by simulation. Simulations using the modified parameters gave results which compared fairly close to measured values.

Simulation of the PER system in the heating mode was accomplished using a computer model of the PER system similar to that used in the cooling mode. Comparison of the simulation and measured performance
quantities was not attempted because of the optional electrical resistance heating in the control room. Component performance comparisons were made for the solar-collector/storage-tank system and the heat pump.

The simulation of the collector was fairly accurate in predicting the shape of the actual performance curve. The simulation corroborated the measured results, which showed that the energy transfer between the collector loop and the storage tank is inhibited by the low effectiveness (approximately 0.1) of the tank heater.

The heat pump model was not accurate in predicting the performance of the actual heat pump in the heating mode. The primary reason for the inaccuracy was the influence of the refrigerant piping on the refrigerant's thermodynamic state. The influence of the refrigerant piping on the refrigerant was not accounted for in the heat pump model.

The results of the simulation work completed indicate that accurate simulations can be obtained. However, to obtain these results, detailed knowledge of the system's operating characteristics is required.
BIBLIOGRAPHY


APPENDIX A

I. PER Data Collection System
II. Weather Data Input
I. PER Data Collection System

To evaluate the PER system, an extensive data collection system was utilized. The system collects data on 65 channels with a computer. The channels and channel descriptions are shown in Table A-1. The data is sampled at one minute intervals during the day (0600-1800) and at 15 minute intervals during the night (1800-0600). These data are stored on magnetic tape, with about two weeks of data per tape. The data collected are processed by a Fortran program PERDATA, which calculates the performance for each of the system's major components on an hourly and daily basis. The hourly output for program PERDATA is shown in Figure A-1.

II. Weather Data Input

For valid comparison between the TRNSYS simulation and the PER system, the weather data as measured by the PER data collection system was used in the simulations. Channels from the PER data collection system containing the data used in the simulations are described below:

<table>
<thead>
<tr>
<th>Channel</th>
<th>Channel Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>7</td>
<td>Wind speed</td>
</tr>
<tr>
<td>9</td>
<td>Ambient Temperature</td>
</tr>
<tr>
<td>10</td>
<td>Total Insolation</td>
</tr>
<tr>
<td>15 or 16</td>
<td>Air Temperature Entering Heat Pump Evaporator (heating only)</td>
</tr>
</tbody>
</table>

Program PERDATA was developed by Professor M. B. Larson of the Mechanical Engineering Department at Oregon State University. Modifications and updating of PERDATA was performed by Steve Fox, an undergraduate at Oregon State University.
Table A-1. Channel and channel description for PER data collection system.

<table>
<thead>
<tr>
<th>Channel No.</th>
<th>Channel Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>Calibration +5.000 v</td>
</tr>
<tr>
<td>02</td>
<td>Calibration 0.000 v</td>
</tr>
<tr>
<td>03</td>
<td>Fan Coil Htr &amp; Clr Pump Power</td>
</tr>
<tr>
<td>04</td>
<td>Heat Pump Comp Htr Power Watts</td>
</tr>
<tr>
<td>05</td>
<td>Absorption Pump &amp; Water Clr Pump</td>
</tr>
<tr>
<td>06</td>
<td>Wind Dir - Azimuth Degree</td>
</tr>
<tr>
<td>07</td>
<td>Wind Spd - Meter/Sec</td>
</tr>
<tr>
<td>08</td>
<td>Collector Pump Power Watts</td>
</tr>
<tr>
<td>09</td>
<td>Outdoor Temp - Thermocouple</td>
</tr>
<tr>
<td>10</td>
<td>Insolation, Total - Pyranometer</td>
</tr>
<tr>
<td>11</td>
<td>Insolation, Norm. Incld - Pycheliometer</td>
</tr>
<tr>
<td>12</td>
<td>Supply Water Temp - City Water</td>
</tr>
<tr>
<td>13</td>
<td>Collector - Ent Water Temp</td>
</tr>
<tr>
<td>14</td>
<td>Collector - Diff Temp</td>
</tr>
<tr>
<td>15</td>
<td>Air Temp Ent O.D. Evap # 1</td>
</tr>
<tr>
<td>16</td>
<td>Air Temp Ent O.D. Evap # 2</td>
</tr>
<tr>
<td>17</td>
<td>Space Return Air Temp</td>
</tr>
<tr>
<td>18</td>
<td>Supply Water Temp - Fan Coll</td>
</tr>
<tr>
<td>19</td>
<td>Return Water Temp - Fan Coll</td>
</tr>
<tr>
<td>20</td>
<td>Fan Coll - Diff Temp</td>
</tr>
<tr>
<td>21</td>
<td>Absorp Evap - Ent Temp Water</td>
</tr>
<tr>
<td>22</td>
<td>Absorp Evap - Diff Temp Water</td>
</tr>
<tr>
<td>23</td>
<td>Storage Tank Leaving Water Temp</td>
</tr>
<tr>
<td>24</td>
<td>Storage Tank - Diff Temp Water</td>
</tr>
<tr>
<td>25</td>
<td>Heat Pump - Disch (Hot Gas) Temp</td>
</tr>
<tr>
<td>26</td>
<td>Heat Pump - Liquid Ref Temp</td>
</tr>
<tr>
<td>27</td>
<td>Heat Pump - Suction (Return Gas) Temp</td>
</tr>
<tr>
<td>28</td>
<td>Heat Pump - Sat Suction Pressure</td>
</tr>
<tr>
<td>29</td>
<td>Cooler - Leaving Water Temp</td>
</tr>
<tr>
<td>30</td>
<td>Cooler - Diff Temp, Water</td>
</tr>
<tr>
<td>31</td>
<td>Dump Heat Exch - Leaving Water Temp</td>
</tr>
<tr>
<td>32</td>
<td>Dump Heat Exch - Diff Temp, Water</td>
</tr>
<tr>
<td>33</td>
<td>Absorp Regen - Leaving Water Temp</td>
</tr>
<tr>
<td>34</td>
<td>Absorp Regen - Diff Temp Water</td>
</tr>
<tr>
<td>35</td>
<td>Collector Water Flow Rate</td>
</tr>
<tr>
<td>36</td>
<td>Absorption Mach Evap Water Flow</td>
</tr>
<tr>
<td>37</td>
<td>Fan Coll Water Flow Rate</td>
</tr>
<tr>
<td>38</td>
<td>Heat Pump Heflg Flow Rate</td>
</tr>
<tr>
<td>39</td>
<td>Return Air - Rel Humidity</td>
</tr>
<tr>
<td>40</td>
<td>Absorption Absorb-Cond-diff Temp</td>
</tr>
<tr>
<td>41</td>
<td>Absorption Absorb-Cond-Flow Rate</td>
</tr>
<tr>
<td>42</td>
<td>Heat Pump Condenser - Flow Rate</td>
</tr>
<tr>
<td>43</td>
<td>Heat Pump Condenser - Diff Temp</td>
</tr>
<tr>
<td>44</td>
<td>Storage Tank - Diff Temp</td>
</tr>
<tr>
<td>45</td>
<td>Storage Tank Outlet Temp</td>
</tr>
<tr>
<td>46</td>
<td>Storage Tank Bottom Temp</td>
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</table>
Table A-1. Cont.

<table>
<thead>
<tr>
<th>Channel No.</th>
<th>Channel Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>47</td>
<td>Storage Tank Sec. from Bottom Temp</td>
</tr>
<tr>
<td>48</td>
<td>Storage Tank Third from Bottom Temp</td>
</tr>
<tr>
<td>49</td>
<td>Storage Tank Top Temp</td>
</tr>
<tr>
<td>56</td>
<td>Heat/Cool - Selector Switch (SW)</td>
</tr>
<tr>
<td>57</td>
<td>Heat Pump - Heating Mode (H)</td>
</tr>
<tr>
<td>58</td>
<td>Storage Tank - Heating Mode (14)</td>
</tr>
<tr>
<td>59</td>
<td>Dump Mode - Tank or Collector Safety Control (D)</td>
</tr>
<tr>
<td>60</td>
<td>Collector - Heat or Cool (1)</td>
</tr>
<tr>
<td>61</td>
<td>Heat Pump - Pump Down Mode (PDR)</td>
</tr>
<tr>
<td>62</td>
<td>Heat Pump - Cooling Mode (C)</td>
</tr>
<tr>
<td>63</td>
<td>Storage Tank - Cooling Mode (7)</td>
</tr>
<tr>
<td>64</td>
<td>Extra</td>
</tr>
<tr>
<td>65</td>
<td>Extra</td>
</tr>
<tr>
<td>Mode of Operation</td>
<td>Energy Inputs - Total System</td>
</tr>
<tr>
<td>------------------</td>
<td>-----------------------------</td>
</tr>
<tr>
<td>Storage Tank</td>
<td>Total Collected Energy</td>
</tr>
<tr>
<td>Cooled by Absorp Mach</td>
<td>47.4278 KWH</td>
</tr>
<tr>
<td>Collector Pump</td>
<td>Fan Coil Cooling of Space</td>
</tr>
<tr>
<td>Is Running</td>
<td>8.1580 KWH</td>
</tr>
<tr>
<td>Collector Pump</td>
<td>Fan Coil during Cooling</td>
</tr>
<tr>
<td>Energy Input</td>
<td>Electrical Input</td>
</tr>
<tr>
<td>Total Energy</td>
<td>Total Energy Input</td>
</tr>
<tr>
<td>From Transformer</td>
<td>8.3932 KWH</td>
</tr>
<tr>
<td>Collector Pump</td>
<td>60.7507 KWH</td>
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<tr>
<td>Fan Coil System</td>
<td></td>
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<tr>
<td>Cooling</td>
<td></td>
</tr>
<tr>
<td>Cooling of Space</td>
<td>8.3580 KWH</td>
</tr>
<tr>
<td>Energy to Storage</td>
<td>7.9237 KWH</td>
</tr>
<tr>
<td>Energy to HP Chiller</td>
<td>0.0000 KWH</td>
</tr>
<tr>
<td>Pipe Gain</td>
<td>-0.4343 KWH</td>
</tr>
<tr>
<td>Heat Pump System</td>
<td></td>
</tr>
<tr>
<td>Cooling, COP= 0.0000</td>
<td>1.0052 KWH</td>
</tr>
<tr>
<td>Energy to Drive Comp and Circ Pump</td>
<td>0.0000 KWH</td>
</tr>
<tr>
<td>Energy Input thru Chiller</td>
<td>0.0000 KWH</td>
</tr>
<tr>
<td>Energy to Condenser - Waste</td>
<td>-0.4343 KWH</td>
</tr>
<tr>
<td>Unaccounted for - Cooling</td>
<td>1.0052 KWH</td>
</tr>
<tr>
<td>Collector System</td>
<td></td>
</tr>
<tr>
<td>Performance, EFF = 57.14 PER CENT</td>
<td></td>
</tr>
<tr>
<td>Incident Energy</td>
<td>83.1509 KWH</td>
</tr>
<tr>
<td>Collected Energy</td>
<td>47.4278 KWH</td>
</tr>
<tr>
<td>Energy to Storage</td>
<td>0.0000 KWH</td>
</tr>
<tr>
<td>Energy to Absorption Generator</td>
<td>48.3844 KWH</td>
</tr>
<tr>
<td>Energy to Dump</td>
<td>-0.7552 KWH</td>
</tr>
<tr>
<td>Pipe Loss Collector System</td>
<td>-1.7144 KWH</td>
</tr>
<tr>
<td>Absorption Machine Performance, COP= 0.6196</td>
<td></td>
</tr>
<tr>
<td>Energy for Pump</td>
<td>3.4731 KMH</td>
</tr>
<tr>
<td>Energy to Regen from Collector Loop</td>
<td>48.3844 KWM</td>
</tr>
<tr>
<td>Energy from Storage Tank</td>
<td>32.1346 KWH</td>
</tr>
<tr>
<td>Energy to Waste</td>
<td>76.6717 KWH</td>
</tr>
<tr>
<td>Unaccounted for</td>
<td>7.3256 KWH</td>
</tr>
</tbody>
</table>

**Figure A-1. PERDATA hourly printout.**
To process the weather data collected by the PER system for input to TRNSYS, the first step is to create a data file comprised of weather data obtained from the proper channels for the simulation period. Since TRNSYS requires data at constant time intervals, the next step involves interpolation of the data to obtain estimates of the values at that interval. Interpolation is not required if data exists at the time interval; however, the PER data collection system was not perfect and gaps in the data did exist at several intervals throughout the simulation period. Interpolation of data is by an approximating function, Newton's Divided Difference Interpolating Polynomial. From Figure A-2 the following terms are defined:

\[ f[X_0] = f(X_0) \]
\[ f[X_1, X_0] = \frac{f(X_1) - f(X_0)}{X_1 - X_0} \]
\[ f[X_2, X_1, X_0] = \frac{f[X_2, X_1] - f[X_1, X_0]}{X_2 - X_1} \]

The second-degree interpolating polynomial is defined as
\[ P_2(x) = f[X_0] + (x - X_0) f[X_1, X_0] + (x - X_0)(x - X_1) f[X_2, X_1, X_0] \] (A-1)
The estimated value of \( f(x) \) is found from equation (A-1) by substituting the known value of \( x \). In this case, the \( x \) values are time, and the interpolated value is one of the four channels discussed. Program 1 incorporates this method to complete step two.

The solar radiation data as collected by the PER system is measured at 30° from the horizontal (the same angle of tilt as the collectors). The radiation calculations used in TRNSYS require that
Figure A-2. The interpolating polynomial.
radiation data used as inputs to TRNSYS, be values that are horizontal measurements. Thus, the third and final step is to transform the instantaneous total radiation, measured by the PER system at 30°, to beam and diffuse radiation incident on a horizontal surface. The relation given by Liu and Jordan A-2

\[
\frac{H_d}{H} = 1.0045 + 2.6313 k_t^3 - 3.5227 k_t^2 + 0.04349 k_t
\]  

(A-2)

where \( k_t = \frac{H}{H_0} \)

relates the total radiation, \( H \), to the diffuse radiation \( H_d \).

The total radiation on a tilted surface, \( H_T \) is correlated to the beam, diffuse and total radiation on a horizontal surface by

\[
H_T = H_b R_b + H_d R_d + H R_r
\]  

(A-3)

The ratios, \( R_b \), \( R_d \), and \( R_r \) are constant for a given time step and are independent of the radiation terms.

Dividing equation (A-3) by \( H \) and substituting \( H - H_d \) for \( H_b \):

\[
\frac{H_T}{H} = \frac{H - H_d}{H} R_b + \frac{H_d}{H} + R_r
\]  

(A-4)

rearranging to:

\[
H = \frac{H_T}{R_b} + \frac{H_d}{H} (R_d - R_b) + R_r
\]  

(A-5)

Equation (A-5) can be solved by an iterative process involving equation (A-2). First assume a value of \( K_T \) (\( H_T \) can be used in place of \( H \) as it is a known quantity to estimate \( K_T \)) to calculate equation (A-2). As all terms in equation (A-5) are now known, \( H \) can be solved for. The process is repeated using the calculated value of \( H \) to

determine $K_T$ in equation (A-2). When succeeding calculations of $K_T$ converge to a specified tolerance the iteration is stopped and values of $H_b$ and $H_d$ are calculated. Program number two completes the final step.
Program 1.

```plaintext
PROGRAM QUICK(INPUT, OUTPUT, TAPE10, TAPE6)
DIMENSION TIM(5500), HT(5500), W(5500), AM(5500), THX(5500)
REAL NEXTIME
P(FX0, X0, X1, X, FX1X0, F3X) = FX0 + (X-X0) * FX1X0 + (X-X0) * (X-X1) * F3X
READ*, NEXTIME, NR, DAYS
DO 1 I=1, NR
   8 READ(10,*) TIM(I), HT(I), W(I), AM(I), THX(I), TDAY
   IF(EOF(10)) 2, 5
   IF(ABS(NEXTIME-TIM(I)-DAYS) > 13.) DAYS = DAYS + 24.
   5 TIM(I) = TIM(I) + DAYS
   IF(TIM(I) .EQ. TIM(I-1)) GO TO 8
   IF(I .LT. 3) GO TO 1
   IF(TIM(I) .LT. NEXTIME) GO TO 1
   IF(TIM(I) .NE. NEXTIME) GO TO 4
   6 WRITE(6,*) TIM(I), HT(I), W(I), AM(I), THX(I), TDAY
   NEXTIME = TIM(I) + 0.125
   GO TO 1
4 CONTINUE
   X0 = TIM(I-2) $ X1 = TIM(I-1) $ X2 = TIM(I) $ X = NEXTIME
   FX1X0 = (HT(I-1) - HT(I-2)) / (X1-X0)
   FX2X1 = (HT(I) - HT(I-1)) / (X2-X1)
   F3X = (FX2X1 - FX1X0) / (X2-X1)
   RAD = P(FX0, X0, X1, X, FX1X0, F3X)
   FX1X0 = (W(I-1) - W(I-2)) / (X1-X0)
   FX2X1 = (W(I) - W(I-1)) / (X2-X1)
   F3X = (FX2X1 - FX1X0) / (X2-X1)
   WIN = P(FX0, X0, X1, X, FX1X0, F3X)
   FX1X0 = (AM(I-1) - AM(I-2)) / (X1-X0)
   FX2X1 = (AM(I) - AM(I-1)) / (X2-X1)
   F3X = (FX2X1 - FX1X0) / (X2-X1)
   TAM = P(FX0, X0, X1, X, FX1X0, F3X)
   FX1X0 = (THX(I-1) - THX(I-2)) / (X1-X0)
   FX2X1 = (THX(I) - THX(I-1)) / (X2-X1)
   F3X = (FX2X1 - FX1X0) / (X2-X1)
   TH = P(FX0, X0, X1, X, FX1X0, F3X)
   WRITE(6,*) NEXTIME, RAD, WIN, TAM, TH, TDAY
4 CONTINUE
2 CONTINUE
END
```
Program 2.

```
PROGRAM ZIP(INPUT, OUTPUT, TAPE6, TAPE7, TAPE8=/80, TAPE9=/80)
REAL KT, KTNW
RD=0.933 S N=0 $ NDOLD=1
READ*, IFD, E, RR
DO 2 I=1, 3000
READ(6,*) TIME, HWT, WIND, TA, TDAY
IF(EOF(6))3, 1
1 IF(HWT GT 0.) GO TO 6
   HD=HWT*3.6
   HB=0.
   GO TO 4
6 HT=HWT*3.6
   ND=INT(TIME/24.)
   IF(ND.EQ.NDOLD) GO TO 5
   NDOLD=ND
   N=IFD+ND
   DELTA=0.40928*SIN((284.+N)*0.017214)
   WS=5.2958*ACOS(-1.0222*TAN(DELTA))
   TD=0.133333*WS $ RISE=12.-TD/2. $ SET=12+TD/2.
5 SOLT=TIME-ND*24.+E
   IF(SOLT LE RISE+0.5) SOLT=RISE+0.5
   IF(SOLT GT SET-0.5) SOLT=SET-0.5
   W=0.2618*(12.-SOLT)
   CSH=0.69929*COS(DELTA)*COS(W) + 0.7148*SIN(DELTA)
   CST=0.26943*SIN(DELTA) + 0.96302*COS(W) *COS(DELTA)
   RB=CST/CSH
   IF(CST GT 0.) GO TO 9
   HB=0.
   HD=HT/RD
   GO TO 4
9 HO=4871.*(1.+0.0333*COS(0.01721*N)) *CSH
   KT=HT/HO
   IF(KT GT 0.98) KT=0.95
   DO 7 J=1,20
      FKT=1.0045+2.6313*KT*KT-3.5227*KT*KT+0.04349*KT
      HNEW=HT/(RB+FKT*(RD-RB)+RR)
      KTNW=HNEW/HO
   7 CONTINUE
WRITE(8,*) "DID NOT CONVERGE AT TIME = ", TIME
8 CONTINUE
   KT=KTNW
   HD=HNEW*(1.0045+2.6313*KT*KT-3.5227*KT*KT+0.04349*KT)
   HB=HNEW-HD
   HTP=HB*RB+HD*RD+HNEW*RR
   IF(ABS(KTNW-KT) LT (0.01)) GO TO 8
   KT=KTNW
7 CONTINUE
WRITE(8,*) " "
8 CONTINUE
   KT=KTNW
   HD=HNEW*(1.0045+2.6313*KT*KT-3.5227*KT+0.04349*KT)
   HB=HNEW-HD
   HTP=HB*RB+HD*RD+HNEW*RR
   IF(ABS(KTNW-KT) LT (0.01)) WRITE(8,*) TIME, HB, HD, RB, HT
   V, DELTA, CSH, CST, SOLT
   IF((HB, LT, (-0.2)) OR (HNEW, GT, HO)) WRITE(9,*) "TIME", TIME
   HB, HD, RB, HT, CSH, CST, HO, W
4 IF(WIND LT 0.) WIND=0.
   WRITE(7,*) TIME, HB, HD, WIND, TA, THX, TDAY
2 CONTINUE
3 CONTINUE
END
```
APPENDIX B

Component Subroutines

I. Solar Collector
II. Attic
III. Absorption Cooler
IV. Controller
V. Heat Pump
I. Solar Collector

The parameters used in this program are:

1. Latitude of the collector
2. Longitude of the collector
3. The standard meridian
4. The correction factor from the equation of time (in hrs)
5. Day of the year at the start of the simulation
6. Collector tilt from horizontal
7. The diffuse radiation enhancement factor
8. Ratio of the absorber tube diameter to the tube spacing
9. Outer radius of delivery tube
10. Reflectivity of the reflector
11. Inner radius of absorber tube
12. Outer radius of absorber tube
13. Thermal conductivity of fluid
14. Inner radius of cover tube
15. Length of collector tube
16. Product of cover tube transmittance and the absorptivity of the selective surface
17. Number of collector tubes
18. Capacitance of the collector
19. Number of equal volume, fully mixed segments in the collector
20. Collector pump power
The inputs used in this program are:

1. Temperature of the fluid into the collector
2. Fluid mass flow rate
3. Ambient temperature
4. Beam radiation on the tilt plane of the collector
5. Diffuse radiation on the tilt plane of the collector
6. Absorber tube temperature (average value)
7. Total radiation on the tilt plane of the collector

The outputs from this program are:

1. Temperature of the fluid exiting the collector
2. Fluid mass flow rate
3. Rate at which energy is collected
4. Absorber tube temperature
5. The effective radiation on the absorber tube
6. The collector pump power
7. The instantaneous efficiency

The derivatives used in this program are:

1. Temperatures of the first segment
2. Temperature of the i-th segment
3. Temperature of the n-th segment
SUBROUTINE TYPE30(TIME, XIN, OUT, T, DTDT, PAR, INFO)
C THIS SUBROUTINE MODELS THE OWENS-ILLINOIS TUBULAR COL-
C LECTOR. THE FORM OF SOLUTION FOLLOWS THAT OF BEEKLY
C AND MATHER WITH ASSUMPTIONS TO ACCOUNT FOR WATER-
DIMENSION XIN(8), OUT(20), PAR(20), INFO(9), T(10), DTDT(10)
REAL LOSS
DATA PI/3.1416/, SIGMA/2.*04095E-07/, CP/4.186/
DATA C10/0.53341/, C20/-0.27153/, C30/0.69543/, C40/-1.0
14177/
N= PAR(19) $ NTB=PAR(17)
IF(INFO(7).GE.0) GO TO 1
INFO(6)=7
CALL TYPECK(1, INFO, 7, 20, N)
1 CONTINUE
TIN=XIN(1) $ FLT=XIN(2)/NTB $ TA=XIN(3)
HBT=XIN(4) $ HT=XIN(5) $ T3=XIN(6) $ NTB=XIN(7)
ALAT=PAR(1)*0.0174533 $ ALOC=PAR(2)*0.0174533
ALT=PAR(3)*0.0174533 $ E=PAR(4) $ DAYS=PAR(5)
S=PAR(6)*0.0174533 $ GD=PAR(7) $ RD=PAR(8) $ RWO=PAR(9)
RHO=PAR(10) $ R3I=PAR(11) $ R30=PAR(12) $ WK=PAR(13)
RI=PAR(14) $ X=PAR(15) $ ATAU=PAR(16) $ CAPC=PAR(18)
AR3=2*PI*R30*X $ TAK=TAK+273.15
C DETERMINE SOLAR HOUR ANGLE, W.
C
NDAYS=DAYS+INT(TIME/24.)
DELTA=0.40928*SIN((284+NDAYS)*0.017214)
SOLART=TIME+E+(ALAT-ALOC)*3.81972
W=0.2618*(12*SOLART)
OMEGA=ATAN(COS(DELTA)*SIN(W)/(COS(ALAT-ALOC)*COS(DELTA)*
SIN(W)+COS(ALAT-ALOC)*SIN(DELTA)))
OS=ACOS(RI*RD/R30)
CME=COS(OMEGA)
G=R30*CME/(RI*RD)
IF(ABS(Omega)<1E-6) G=1.0
OMEGA=ABS(OMEGA)
F=G10*C20*OMEGA+C30*OMEGA**2+C40*OMEGA**3.
IF(F<FLT .LT. 0.) F=0.
SEFF=GD*HDT
IF(ABS(CME) .LT. 0.0001) GO TO 43
C EVALUATE SEFF.
C
SEFF=(G/CME+RHO*F/RD)*HBT*GD*HDT
43 CONTINUE
C DETERMINE CONSTANTS IN SOLAR COLLECTOR MODEL.
C
H3=5.099*WK/(2*(R3I-RWO))
HR=SIGMA*(T3**2+TAK**2)*(T3+TAK)/((13.2857*RI/R30)+RI/
R30+0.1111)
UL=HR*RI/R30
FP=1/(1+(UL*RI/R30/(H3*R3I)))
B1=((0.782*SEFF*R30/R3I)+UL*TAK*R30/R3I)/(H3+UL*RI/R3I)
CAFN=CAPC/N $ LOSS=0.
C SOLVE DIFFERENTIAL EQUATIONS

DTDT(1) = (FLRT*CP*(TIN-T(1)) - UL*AR3*FP*(T(1)-TA)/N
1 + ATAU*SEFF*2*R30*X*FP/N)/CAPN
LOSS = UL*(TIN-TA)/N
DO 2 J = 2,N
K1 = J - 1 $ K2 = J
DTDT(K2) = (FLRT*CP*(T(K1) - T(K2)) - UL*AR3*FP*(T(K2) - TA)/N
1 + ATAU*SEFF*2*R30*X*FP/N)/CAPN
LOSS = LOSS + UL*(T(K1) - TA)/N
2 CONTINUE
QDOT = POWER = EFF = 0. $ T3 = T(N) + 273.15
IF (FLRT < 0.1) GO TO 3
POWER = PAR(20)
IF (HT < 0.1) GO TO 3
W2 = UL*R30*FP/(2*H3*R3I)
CD = H3*2*PI*R3I*X/(FLRT*CP)
FR = FP*SINH(W2*CD)/(W2*CD*(COSH(W2*CD) + SINH(W2*CD)))
SBAR = FP*(1 - EXP((FP - 1.)*CD))*SEFF/(FR*CD*(1 - FP))
QDOT = FR*2*R30*X*(ATAU*SBAR - PI*LOSS)*NTB
EFF = RD*FR*(ATAU*SEFF - PI*LOSS)/HT
T3 = BI + FP*(T(N) + 273.15)
3 CONTINUE
OUT(1) = T(N)
OUT(2) = FLRT*NTB
OUT(3) = QDOT
OUT(4) = T3
OUT(5) = SEFF
OUT(6) = POWER
OUT(7) = EFF
RETURN
END
II. Attic

The parameters used in this program are:

1. The loss coefficient for the ceiling
2. Capacitance of the attic
3. Number of volume changes per hour
4. Volume of the attic space

The inputs to this program are:

1. The control room temperature
2. Heat transfer rate from the ceiling
3. Heat transfer rate from the walls
4. The ambient temperature

The outputs to this program are:

1. The attic temperature
2. The heat transfer rate to the room

The derivatives in this program are:

1. Temperature of the attic
SUBROUTINE TYPE 31 (TIME, XIN, OUT, T, DTDT, PAR, INFO)
C ATTIC MODEL DEVELOPED BY W. C. LAYTON
DIMENSION XIN(4), OUT(2), PAR(4), DTDT(1), T(1), INFO(9)
IF(INFO(7) .GE. 0) GO TO 1
INFO(6) = 2
CALL TYPECK(1, INFO, 4, 4, 1)
1 CONTINUE
TROOM = XIN(1)
QROOF = XIN(2)
QUALLS = XIN(3)
TAMB = XIN(4)
UA = PAR(1)
SCAFAC = PAR(2)
RATE = PAR(3)
VOL = PAR(4)
QINF = RATE * VOL * 1.2185 * (TAMB - T(1))
DTDT(1) = (QROOF + QUALLS - UA * (T(1) - TROOM) + QINF) / CAPAC
GROOM = UA * (T(1) - TROOM)
OUT(1) = T(1)
OUT(2) = GROOM
RETURN
END
III. Absorption Cooler

The parameters used in this program are:

1. Specific heat of fluid entering the absorption cooler
2. Coefficient of performance
3. Upper limit of absorption unit in terms of its nominal value
4. Power use of absorption unit

The inputs used in this program are:

1. Temperature of the fluid from the storage tank
2. Mass flowrate of the fluid from the storage tank
3. Temperature of the fluid entering the absorption cooler from the collector
4. Mass flowrate of the fluid entering the absorption cooler from the collector

The outputs from this program are:

1. Temperature of the fluid returning to the storage tank
2. Mass flowrate of the fluid returning to the storage tank
3. Temperature of the fluid returning to the collector
4. Mass flowrate of the fluid returning to the collector
5. Rate of heat rejection of the fluid from the storage tank
SUBROUTINE TYPE36(TIME, XIN, OUT, T, DTDT, PAR, INFO)
DIMENSION XIN(5), OUT(5), PAR(4), INFO(9)
C THIS SUBROUTINE IS A MODEL OF A 15 TON ABSOPRTION COOLER
C AS TESTED BY CARRIER FOR THE BPA PROJECT.
C
IF(INFO(7).GE.0)GO TO 1
INFO(6)=5
CALL TYPECK(1, INFO, 4, 4, 0)
1 CONTINUE
TSIN=XIN(1) $ FRS=XIN(2) $ TCI=XIN(3) $ FRC=XIN(4)
CP=PAR(1) $ COP=PAR(2) $ QMAX=PAR(3)*12660.
TSOUT=TSIN $ TCO=TCI $ QDOT=POWER=0.
IF((FRS.EQ.0.).OR.(FRC.EQ.0.))GO TO 2
POWER=PAR(4)
TFCI=32.0+1.8*TCI
QDOT=(-34.4077+0.2989*TFCI)*12660.
IF(QDOT.GT.QMAX)QDOT=QMAX
TSOUT=TSIN-QDOT/(FRS*CP)
TCO=TCI-QDOT/(COP*CP*FRC)
2 CONTINUE
OUT(1)=TSOUT
OUT(2)=FRS
OUT(3)=TCO
OUT(4)=FRC
OUT(5)=QDOT
OUT(6)=POWER
RETURN
END
IV. Controller - cooling mode

The parameters used in this program are:

1. Number of calls to controller in a time step after which \( y_0 \) remains constant
2. Temperature at which the room thermostat is set
3. Upper dead band for room
4. Lower dead band for room
5. Temperature at which the collector turns off
6. Upper dead band for collector
7. Difference between the incoming and outgoing collector fluid temperature at which the collector shuts off
8. Lower dead band for collector
9. Maximum allowable temperature the collector is allowed to reach before the dump heat exchanger is turned on
10. Upper dead band for dump heat exchanger
11. Lower dead band for dump heat exchanger
12. Temperature of the bottom tank segment at which the absorption cooler shuts off
13. Upper dead band for absorption cooler
14. Temperature of the fluid entering the absorption cooler from the collector for which the absorption cooler can no longer operate
15. Upper dead band for fluid entering absorption cooler from the collector
16. Lower dead band for absorption cooler
17. Lower dead band for fluid entering absorption cooler from the collector
18. Upper dead band for heat pump cooling
19. Lower dead band for heat pump cooling
20. Storage tank temperature at which cooling is initiated by the heat pump

The inputs used in this program are:

1. Temperature of the fluid entering the collector
2. Temperature of the fluid leaving the collector
3. Temperature of the bottom segment in the storage tank
4. The room temperature
5 - 9. Control functions inputs
10. The temperature of the fluid entering the absorption cooler

The outputs from this program are:

1. Control function of the collector pump
2. Control function of the dump heat exchanger
3. Control function of the absorption cooler
4. Control function of the cooling water
5. Control function of the heat pump
SUBROUTINE TYPE37(TIME, XIN, OUT, T, DTDT, PAR, INFO)
DIMENSION XIN(10), PAR(20), OUT(20), INFO(9)
IF/INFO(7) .GT. 0) GO TO 2
OUT(11) = XIN(5) $ OUT(12) = XIN(6) $ OUT(13) = XIN(7)
OUT(14) = XIN(8) $ OUT(15) = XIN(9)
IF/INFO(7) .GE. 0) GO TO 2
INFO(6) = 5
CALL TYPECK(1, INFO, 10, 20, 0)
2 NSTX= PAR(1)
OUT(1) = XIN(5) $ OUT(2) = XIN(6) $ OUT(3) = XIN(7)
OUT(4) = XIN(8) $ OUT(5) = XIN(9) $ THXO = XIN(10)
TCI = XIN(1) $ TCO = XIN(2) $ TN = XIN(3) $ TR = XIN(4)
IF/INFO(7) .LE. 0) OUT(6) = 0
OUT(6) = OUT(6) + 1.
IF/OUT(6) .GT. NSTK) RETURN
TSET = PAR(2) $ DELTR1 = PAR(3) $ DELTR2 = PAR(4)
TMIN = PAR(5) $ DMAX = PAR(6) $ DIFF = PAR(7) $ DMIN = PAR(8)
TMAX = PAR(9) $ DHXH = PAR(10) $ DHXL = PAR(11) $ TOFF = PAR(12)
DON = PAR(13) $ TABS = PAR(14) $ DABS = PAR(15) $ DOFF = PAR(16)
DABC = PAR(17) $ DCool = PAR(18) $ DCold = PAR(19)
TCool = PAR(20)
IF/OUT(14) .GE. 3
3 CONTINUE
ICOOL = 0
IF/((TR - TSET) .GT. DELTR1) ICOOL = 1
GO TO 5
4 CONTINUE
ICOOL = 1
IF/((TR - TSET) .LT. DELTR2) ICOOL = 0
5 IF/ICOOL .EQ. 1, 6
6 CONTINUE
OUT(4) = OUT(5) = 0.
17 IF/OUT(11) .GE. 7
7 CONTINUE
OUT(1) = 0.
IF/((TCO - TMIN) .GT. DMAX) OUT(1) = 1.
GO TO 9
8 CONTINUE
OUT(1) = 1.
IF/((TCO - TCI) .GT. DIFF) AND/((TCO - TMIN) .LT. DMIN)) OUT(1) = 0.
9 IF/OUT(1) .EQ. 0.) OUT(2) = OUT(3) = 0.
IF/((OUT(1) .EQ. 0.) .AND. ((ICOOL .EQ. 0)) RETURN
IF/((OUT(1) .EQ. 0.) .AND. ((ICOOL .EQ. 1)) GO TO 16
IF/OUT(12) .GE. 10
10 CONTINUE
OUT(2) = 0.
IF/((TCO - TMAX) .GT. DHXH) OUT(2) = 1.
GO TO 12
11 CONTINUE
OUT(2) = 1.
IF/((TCO - TMAX) .LT. DHXL) OUT(2) = 0.
12 IF(OUT(13)) 14, 13
13 CONTINUE
   OUT(3) = 0.
   IF(((TN-TOFF) .GT. DON) .AND. ((THX0-TABS) .GT. DABS)) OUT(3) = 1.
   GO TO 21
14 CONTINUE
   OUT(3) = 1.
   IF(((TN-TOFF) .LT. DOFF) .OR. ((THX0-TABS) .LT. DABC)) OUT(3) = 0.
21 IF((OUT(3) .EQ. 0.) .AND. ((TC0-TABS) .GT. DABS)) OUT(2) = 1.
   IF(I COOL .EQ. 0) RETURN
16 IF(OUT(15)) 19, 18
18 CONTINUE
   OUT(4) = 1. $ OUT(5) = 0.
   IF((TCOOL-TN) .GT. DCOOL) OUT(5) = 1.
   RETURN
19 CONTINUE
   OUT(4) = 1. $ OUT(5) = 1.
   IF((TCOOL-TN) .LT. DCOLD) OUT(5) = 0.
   RETURN
FIN
V. Heat Pump

The parameters used in this program are:

1. Specific heat of the fluid entering the heat pump
2. Mode of operation, 1-cooling, 2-heating
3. 1st order coefficient for heat transfer
4. 2nd order coefficient for heat transfer
5. 3rd order coefficient for heat transfer
6. Constant coefficient for "A"
7. 1st order coefficient for "A"
8. Constant coefficient for power calculation
9. 1st order coefficient for power calculation
10. 2nd order coefficient for power calculation

The inputs to this program are:

1. Temperature of the fluid entering the evaporator
2. Mass flow rate of the fluid entering the evaporator in the cooling mode--mass flowrate of the fluid entering the condensor in the heating mode.
3. Temperature of the fluid entering the condensor

The outputs to this program are:

1. Evaporator heat rejection in the cooling mode condensor heat rejection in the heating mode
2. Temperature of the fluid leaving the evaporator in the cooling mode
   Temperature of the fluid leaving the condensor in the heating mode
3. Mass flowrate of the fluid leaving the evaporator in the cooling mode—mass flowrate of the fluid leaving the condenser in the heating mode

4. Power usage of the heat pump
SUBROUTINE TYPE38(TIME, XIN, OUT, T, DTDT, PAR, INFO)
DIMENSION XIN(3), OUT(4), PAR(10), INFO(9)
TINC=XIN(1) $ FR=XIN(2) $ TEWT=XIN(3)
C THIS SUBROUTINE MODELS THE PER SYSTEM HEAT PUMP BY THE.
C METHOD OF COMPONENT MODELING.
C
IF(INFO(7) .GE. 0) GO TO 1
INFO(6)=4
CALL TYPECK(1, INFO, 3, 10, 0)
1 CONTINUE
MODE=PAR(2) $ TOUTC=TINC $ QDOT=POWER=0.
IF(MODE .EQ. 2) TOUTC=TEWT
IF(FR .EQ. 0.) GO TO 2
CP=PAR(1) $ B1=PAR(3) $ B2=PAR(4)
B3=PAR(5) $ B4=PAR(6) $ B5=PAR(7) $ B6=PAR(8)
B7=PAR(9) $ B8=PAR(10)
C WATER TEMP ENTERING CONDENSOR IS ASSUMED CONSTANT FOR
C SIMULATION DURING COOLING ONLY
C
TIN=1.8*TINC+32.
TF=1.8*TEWT+32.
A=B4+B5*TF
POWER=B6+B7*QDOT+B8*QDOT*QDOT
TOUTC=TINC-QDOT/(FR*CP)
IF(MODE .EQ. 2) TOUTC=TEWT+QDOT/(FR*CP)
2 CONTINUE
OUT(1)=QDOT
OUT(2)=TOUTC
OUT(3)=FR
OUT(4)=POWER
RETURN
END
APPENDIX C

The Method of Component Balancing
This appendix details the procedure of component balancing as applied to the PER system heat pump. The success of this method is dependent upon accurate performance data for each of the components involved in the process. In this example, there are three major components which comprise the heat pump: a condenser, compressor, and evaporator. In general, performance data is available from the manufacturer in the form of graphs or tables.

The result desired from the method of component balancing is to have the system's performance (the refrigeration capacity in the cooling mode, and the condenser heat rejection in the heating mode) defined in terms of external variables (such as the fluid temperatures entering the condenser and evaporator). The capacity of each of the system's components can be defined in terms of internal variables (such as refrigerant temperatures in the evaporator and condenser) and external variables. Determining the performance of the system as a whole involves the elimination of the internal variables by the process of simultaneously solving a set of equations, each represent a component's performance. Alternatively, this procedure may be applied with an equivalent graphical method. This graphical method is illustrated in Figures C-1 to C-5.

Assumptions used in this procedure are that (i) the thermostatic expansion valve maintains the stated evaporator superheat, (ii) the effects of refrigerant piping are neglected and (iii) there is a constant amount of subcooling in the condenser.
Figure C-1 shows the condenser heat rejection plotted against the condensing temperature for several values of entering water temperature \( T_{ewt} \). The compressor performance based on the condenser heat rejection, in terms of the saturated suction temperature (SST) and \( T_{ewt} \), is superimposed on Figure C-1 to develop Figure C-2.

Figure C-3 is generated by first plotting the refrigerant capacity of the compressor in terms of the condensing temperature and SST. The refrigeration capacity of the condenser can be determined from Figure C-2. The heat rejection by the condenser is a function of the condensing temperature and \( T_{ewt} \). For a given condensing temperature and \( T_{ewt} \), SST is specified. By transferring the points of intersection between \( T_{ewt} \) and SST at a given condensing temperature to Figure C-3 at the same condensing temperature and SST, the condenser performance in terms of refrigerant capacity can then be specified.

The evaporator performance based on the refrigerant capacity is plotted on Figure C-4 in terms of SST and entering fluid temperature \( T_{ent} \). The combined compressor-condenser performance in Figure C-3 is modified by eliminating the condensing temperature as a variable and specifying the performance in terms of SST and \( T_{ewt} \). The result of this modification is then superimposed on Figure C-4.

By applying the same technique again to Figure C-5 to eliminate SST as a variable (internal variable), the performance for the combined components is specified by \( T_{ewt} \) and \( T_{ent} \) (external variables). As Figure C-5 represents the estimated performance of the system with the limits of the assumptions made, TRNSYS simulation of the heat pump is a matter
representing the family of curves with the two external variables, $T_{ewt}$ and $T_{ent}$. The procedure used to estimate each curve, was with a least squares curve fit. The refrigeration capacity in terms of $T_{ent}$ was represented by a second degree polynomial for each value of $T_{ewt}$:

$$\text{Refrig. Capac} = A(T_{ent})^2 + b(T_{ent}) + c$$

where $a$, $b$, and $c$ are constants determined by the least squares curve fit. The results of this procedure indicated that $a$ and $b$ were approximately constant for any value of $T_{ewt}$, but that $c$ varied linearly. Thus, the terms $a$ and $b$ were assumed to be constants and the term $c$ was determined by a linear function of $T_{ewt}$:

$$c = d(T_{ewt}) + e$$

where $d$ and $e$ are constants.
Figure C-1. Condenser performance diagram.
Figure C-2 Combined performance (in terms of condenser heat rejection) for the compressor and condenser.
Figure C-3. Combined performance (in terms of refrigeration capacity) for the compressor and condenser.
**Figure C-4.** Combined performance diagram for evaporator, compressor and condenser.
Figure C-5. Performance diagram for overall system.
APPENDIX D

Computer Input for Simulation of PER System
TRNSYS - A TRANSIENT SIMULATION PROGRAM
FROM THE SOLAR ENERGY LAB AT THE UNIVERSITY OF WISCONSIN
VERSION 0.2 12/20/76

SIMULATION 0. 1.000E+01 1.250E-01
TOLERANCES 1.000E-02 1.000E-02
LIMITS 70 10
WIDTH 132

UNIT 1  TYPE 9  DATA READER
PARAMETERS 3
5.000E+00 1.250E-01 3.600E+01

UNIT 2  TYPE 16  RADIATION PROBE
PARAMETERS 13
4.000E+00 2.140E+02 4.56E+01 5.000E+01 0. 4.871E+03 2.000E-01 1.000E+01 0.

UNIT 3  TYPE 30  SOLAR COLLECTOR NO. 3
PARAMETERS 20
4.300E+01 1.227E+02 1.200E-02 -1.000E-01 2.140E+02 3.000E+01 2.940E+00 4.719E-01 6.150E-02

UNIT 4  TYPE 3  PUMP
PARAMETERS 1
4.000E+01 4.600E+03

UNIT 5  TYPE 3  PUMP
PARAMETERS 1
5.216E+01 3.615E+01

UNIT 6  TYPE 3  PUMP
PARAMETERS 1
5.216E+01 3.615E+01

UNIT 7  TYPE 5  HEAT EXCHANGER
PARAMETERS 4
5.000E+00 2.000E-01 1.146E+00 1.416E+00

UNIT 8  TYPE 36  ABSORPTION COOLER
PARAMETERS 4
1.500E+00 7.100E-01 1.730E+01 1.250E+04
UNIT 24  TYPE 25  PRINTER 40 2
PARAMETERS
1.000E+00  1.000E+00  4.000E+01  0.000E+00
INPUTS 10
23, 6  23, 1  22, 3  22, 4  4, 7  22, 6

UNIT 25  TYPE 21  FORCING FUNCTION LIGHTS
PARAMETERS 12
-1.072E+03  1.000E+00  1.972E+03  9.000E+00  3.744E+03
INPUTS 0

UNIT 26  TYPE 14  FORCING FUNCTION PEOPLE
PARAMETERS 12
-1.000E+00  9.288E+02  1.700E+01
INPUTS 0

UNIT 27  TYPE 15  POWER USE
PARAMETERS 10
-1.000E+00  3.503E+03  1.000E+00
INPUTS 4

UNIT 28  TYPE 24  QUANTITY INTEGRATOR
PARAMETERS 4
1.000E+00  4.800E+01
INPUTS 10

UNIT 29  TYPE 24  QUANTITY INTEGRATOR
PARAMETERS 4
1.800E+00  1.000E+00  1.000E+00
INPUTS 10

UNIT 30  TYPE 15  STORAGE TANK TO LOAD QUOT
PARAMETERS 6
-1.000E+00  1.000E+00
INPUTS 3

UNIT 31  TYPE 25  PRINTER 40 3
UNIT 32  TYPE 25  PRINTER 40 4
PARAMETERS 4
1.000E+00  0.  +.800E+01  0.
INPUTS 10
29, 2  29, 3  29, 4  29, 5  29, 6  10, 7  17, 2  21, 1
Q04.UM GSHG IEAS QLUAU * DEFF SQOFT DELTAE 70 7 TATTIC

UNIT 33  TYPE 15  DAILY EFFICIENCY
PARAMETERS 1
INPUTS 2
4, 7  2, 1

UNIT 34  TYPE 15  INSTANTANEOUS EFFICIENCY
PARAMETERS 6
0.  1.000E+00  0.  4.000E+00  0.  4.186E+00
INPUTS 3
3, 4  3, 1  3, 9
4.910E+01  4.010E+01  0.

END