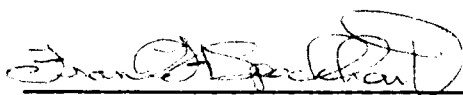


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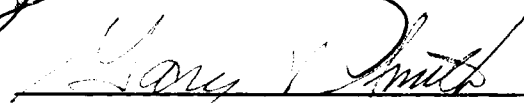
I am submitting herewith a thesis written by David J. Chaffin entitled "Simulated and Experimental Performance of a Solar-Augmented Heat Pump." I recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Engineering, with a major in Mechanical Engineering.


Robert L. Reid, Major Professor

We have read this thesis
and recommend its acceptance:







Accepted for the Council:


Vice Chancellor
Graduate Studies and Research

SIMULATED AND EXPERIMENTAL PERFORMANCE OF A
SOLAR-AUGMENTED HEAT PUMP

A Thesis
Presented for the
Master of Engineering
Degree
The University of Tennessee, Knoxville

David J. Chaffin

December 1980

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ABSTRACT

The heating system performance of the UT-TVA Solar Augmented Heat Pump Test Facility has been simulated using a modified version of the computer code TRNSYS. Simulation results have been compared with experiment for a portion of the 1979-80 heating season.

The Test Facility, located near Knoxville, Tennessee, is an unoccupied converted garage containing an instrumented experimental heating system. The heating system includes an air-heating solar collector array, an air-to-air heat pump, rock bed thermal storage, and an off-peak resistance heater. Hourly performance data from the system have been collected for the 1979-80 heating season by a computerized data acquisition system.

The heating system model was initially formulated using component models available as standard capability in the TRNSYS program. Several of the component models were extended and correlated with experimental performance to provide acceptable overall simulation accuracy. The modifications also produced an approximate 80% reduction in computer execution time as compared to the standard TRNSYS model.

Simulation and experimental results have been compared for a ten week period in midwinter 1979-80. The simulation model was found to produce accurate

representations of total heat and power flows and average storage temperatures for the period.

Using complete heating season meteorological data, the performance of the experimental system and several alternatives has been simulated. An annual power cost analysis applied to the results indicates that the cost of peak period electricity must rise to at least six times the cost of off-peak electricity for the off-peak heating capability of the experimental system to become cost-effective. The solar portion of the experimental system was found to effectively conserve energy, but its cost effectiveness requires further capital cost analysis.

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LIST OF SYMBOLS

SYMBOL	DESCRIPTION	UNITS
A	cross-sectional area of rock bed	2
A	heat loss area of building	ft
A	solar collector plate area	
Bi	Biot number	dimensionless
Cp	specific heat of air	$\frac{\text{BTU}}{\text{lb F}}$
Dr	equivalent spherical rock diameter	ft
$Fr_{\tau\alpha}$	solar collector flow coefficient-transmittance-absorptance product	dimensionless
Fr_{U_L}	solar collector flow coefficient-heat loss coefficient product	$\frac{\text{BTU}}{\text{hr F ft}^2}$
hamb	enthalpy of outdoor air	$\frac{\text{BTU}}{\text{lb}}$
hsp	enthalpy of indoor air	
hcAs	rock bed convection heat transfer-rock surface area product	$\frac{\text{BTU}}{\text{hr F ft}^3}$
I	insolation	$\frac{\text{BTU}}{\text{hr ft}^2}$
kr	rock conductivity	$\frac{\text{BTU}}{\text{hr F ft}}$
L	duct length	ft
$\dot{m}$	air mass flow rate	$\frac{\text{lb}}{\text{hr}}$
P	rock bed or duct perimeter	ft
$\dot{Q}_c$	rate of solar energy collection	$\frac{\text{BTU}}{\text{hr}}$
$\dot{Q}_{\text{cond}}$	rate of conduction heat transfer to rock element	
$\dot{Q}_{\text{conv}}$	rate of convection heat transfer to rock element	
$\dot{Q}_d$	rate of heat delivery by duct to storage	
$\dot{Q}_{\text{inf}}$	rate of building infiltration heat loss	

$\dot{Q}_{\text{gain}}$	rate of heat gain by rock element	$\frac{\text{BTU}}{\text{hr}}$
$\dot{Q}_h$	rate of heat addition to duct by heater	
$\dot{Q}_{\text{loss}}$	rate of heat loss from rock bed	
T	temperature surrounding rock bed	F
T_a	rock bed or duct air temperature	F
T_{amb}	outdoor temperature	F
$T_{a,i}$	rock bed air temperature at node i	F
$T_{f,i}$	solar collector fluid inlet temperature	F
T_r	rock bed rock temperature	F
$T_{r,i}^n$	rock bed rock temperature at node i at time n	F
T_{sp}	indoor space temperature	F
U	overall heat transfer coefficient	$\frac{\text{BTU}}{\text{hr F ft}^2}$
V_{wind}	wind velocity	$\frac{\text{mi}}{\text{hr}}$
x	distance along rock bed in flow direction or distance along duct	ft
Δx	length of rock bed element	ft
Δt	timestep	hr
$(\rho C)_r$	rock density-specific heat product	$\frac{\text{BTU}}{\text{F ft}^3}$
η	solar collector efficiency	dimensionless
μ	mean	dependent on
σ	standard deviation	application

CHAPTER I

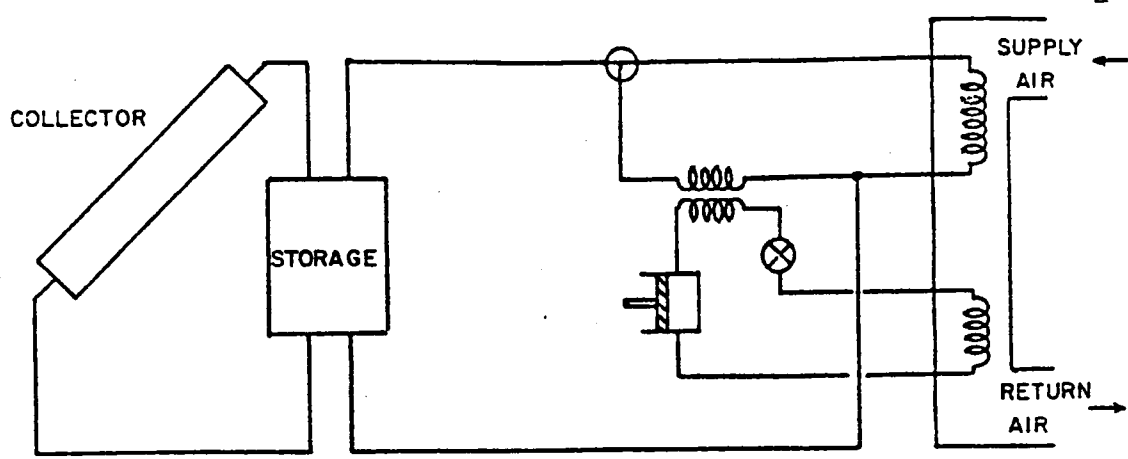
INTRODUCTION

Solar-Heat Pump Systems

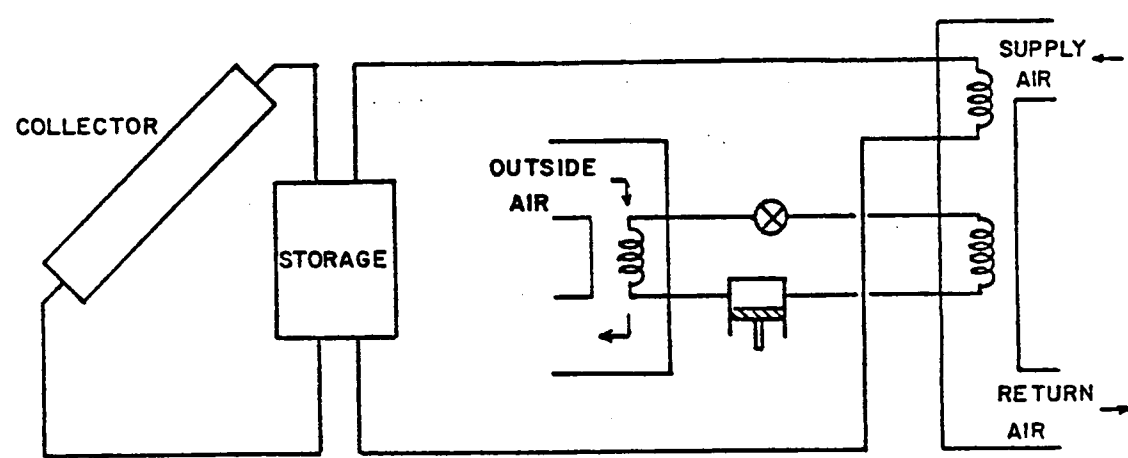
Solar energy systems and heat pumps are two accepted methods for reducing energy consumption for residential space heating. Combinations of solar heating equipment and heat pumps also appear to have merit for energy conservation, since in most climates a solar heating system requires an auxiliary heat source, which may be a heat pump. The heat pump also provides summer cooling capability which is not readily available with current solar technology.

Solar-heat pump space heating systems have been constructed in several different configurations, and no accepted "standard" method has emerged to date. Most solar-heat pump designs may be categorized by the heat source of the heat pump as "parallel," "series," or "dual-source" systems.

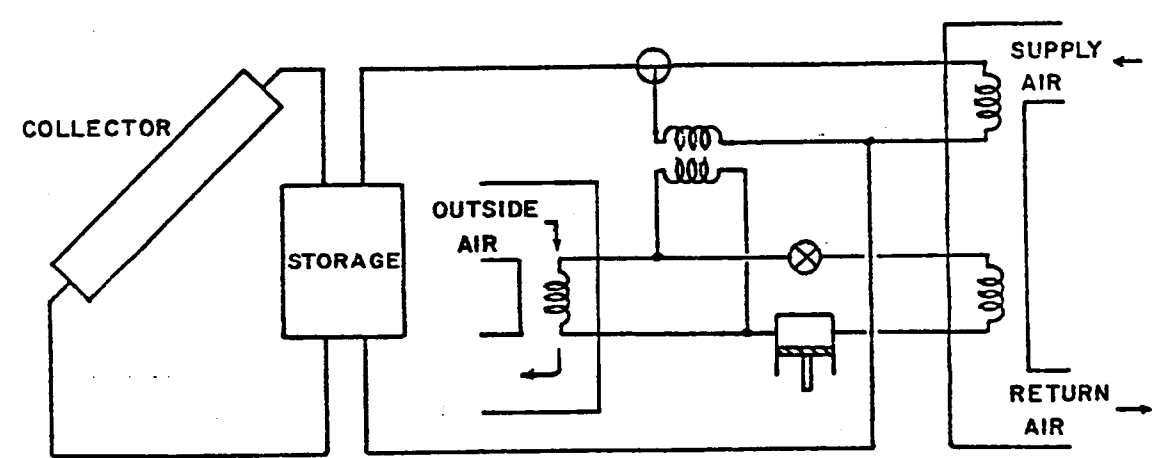
A parallel solar-heat pump space heating system is shown schematically in Figure 1. The parallel system consists of solar-storage and heat pump subsystems which do not exchange heat with each other. The solar subsystem supplies the space heating requirement directly or from thermal storage when possible. During periods of



SERIES SYSTEM



PARALLEL SYSTEM



DUAL SOURCE SYSTEM

Figure 1. Solar-Heat Pump Systems

insufficient insolation and insufficient stored solar energy, the heat pump supplies the space heating requirement by extracting energy from an ambient source, typically outdoor air.

A series solar-heat pump system is also shown in Figure 1. In the series system, the heat pump evaporator coil is typically placed, physically or through ductwork, in the thermal storage unit of the solar-storage subsystem. The solar subsystem then delivers solar energy, when available, to storage. In most series systems the heat pump supplies the space heating requirement at all times by extracting energy from storage, but some systems also have the capability to heat directly from solar collectors or storage.

A dual source solar-heat pump system combines the capabilities of the parallel and series systems. The dual source system, also shown in Figure 1, may have two heat pump evaporator coils or other hardware accomplishing the same end. The dual-source system is capable of extracting energy either from ambient sources or from solar heated storage. The space heating requirement is supplied directly from solar collectors or storage, or indirectly using the heat pump, and a control system selects the more advantageous heat source at any given time.

Presently there is no consensus on the best solar-heat pump system design. Most available analyses in the literature [1,2,3] favor the parallel system. The major advantage of the parallel system is its ability to supply a significant fraction of the heating load directly from storage using only minimal fan or pump energy. Series and dual-source systems, on the other hand, must usually run the heat pump to deliver any heat to space, since the systems generally maintain too low a storage temperature to heat directly from storage. The run time of the heat pump is then increased so that the series and dual-source systems consume more auxiliary energy than the parallel system, even though the series and dual-source heat pumps may operate more efficiently while on.

Utility Load Management

Although solar-heat pump systems and solar systems in general have demonstrated a capability to conserve energy, solar systems have been less successful from the standpoint of electrical utilities. Published results [4,5] have shown that solar-heated houses with electrical auxiliary heat have peak electrical demands nearly as high as those of conventionally heated houses. Since the solar-heated houses have much lower average demand, their load factor (ratio of peak to average demand) is higher, resulting in higher costs to the electrical utility.

Those costs associated with peak electrical demands may at some time be reflected explicitly in electrical rates [6], either through on-peak and off-peak rates or through demand charges.

The poor load management of solar heating systems is caused by control strategies intended to minimize auxiliary energy consumption [5], strategies which do not consider the timing of the consumption. However, a solar heating system using altered control strategy might be able to substantially reduce peak electrical usage, since solar heating systems, unlike conventional heating systems, typically include thermal storage which may be used to supply the space heating requirement during peak electrical usage hours. If the thermal storage can be used to reduce peak loads without degrading solar performance, and if electrical utilities institute lowered off-peak electrical rates, then solar systems may become economically more viable relative to conventional heating systems without thermal storage.

Experimental System

Few solar-heat pump heating systems or load management heating systems have been experimentally evaluated under weather conditions similar to those of the Tennessee Valley. For this reason, the Tennessee Valley Authority has funded the design, construction, and

evaluation of an experimental space heating system near Knoxville, Tennessee. The parallel configuration was chosen for the solar-heat pump for the reasons discussed previously. The load management strategy chosen was: (1) charging of thermal storage using off-peak electricity while (2) supplying space heating during the off-peak period with the parallel heat pump and (3) supplying space heating during the on-peak period using previously stored off-peak energy. An air-based solar-storage system was chosen rather than a liquid-based system for the experiment, since the large thermal storage capacity and thermal stratification required was much more economically supplied by an air-rock packed bed than by the alternative of one or more water tanks. Resistance heat, rather than an additional heat pump, was chosen to supply the off-peak thermal energy. The choice was based on the lower capital cost and higher outlet temperature of the resistance heater as compared with a heat pump.

A schematic of the experimental heating system is shown in Figure 2. The system includes:

1. Air-heating solar collectors (324 sq. ft.).
2. Air-to-air heat pump (2.5 ton) with electric resistance auxiliary heat (10.2 kW).
3. Rock bed thermal storage (254 cu. ft. of 1.5 inch diameter rock).

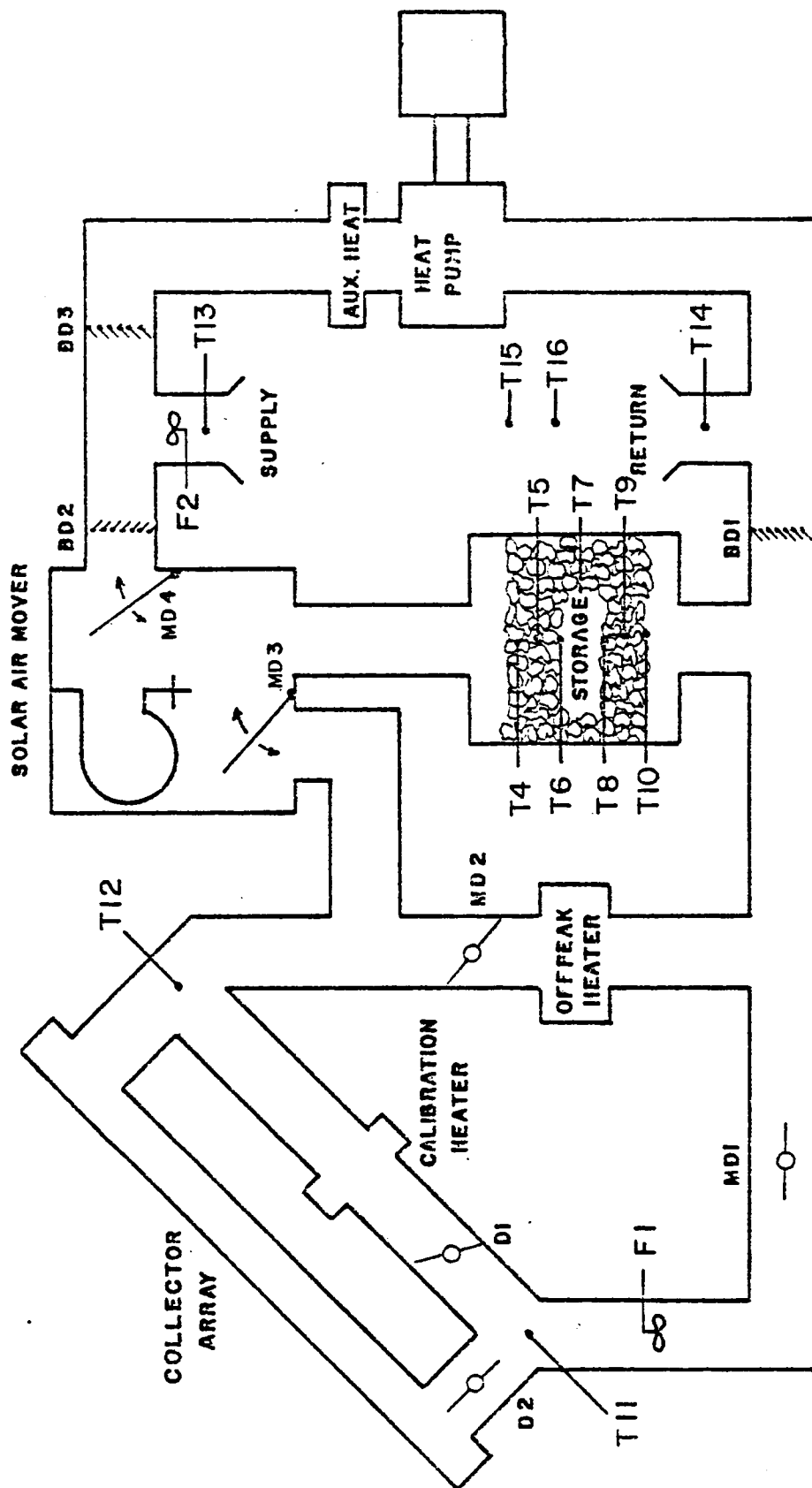


Figure 2. Experimental System and Instrumentation

4. Duct heaters for charging thermal storage during off-peak hours (16.8 kW).
5. Air handling unit with fan (0.5 hp) and control system including four motorized dampers.

The system is capable of heating the space in any of three mutually exclusive modes: (1) direct solar-to-space, (2) storage-to-space, and (3) heat pump-to-space, which may include auxiliary resistance heating. The thermal storage may be charged in one of two mutually exclusive modes: (1) solar-to-storage, or (2) off-peak heater-to-storage. Both methods of charging storage may proceed simultaneously with the heat pump heating the space, but both methods of charging storage are mutually exclusive with solar-to-space and storage-to-space modes.

A block diagram of the system control logic is shown in Figure 3. Primary control is exerted by a three-heating-stage room thermostat. Auxiliary controls include: (1) solar collection control based on the difference between solar collector outlet and rock bed lower outlet temperatures; (2) storage-to-space control based on rock bed upper outlet temperature; (3) off-peak heater control based on a timer, outdoor temperature, and two temperatures at varying positions in the rock bed.

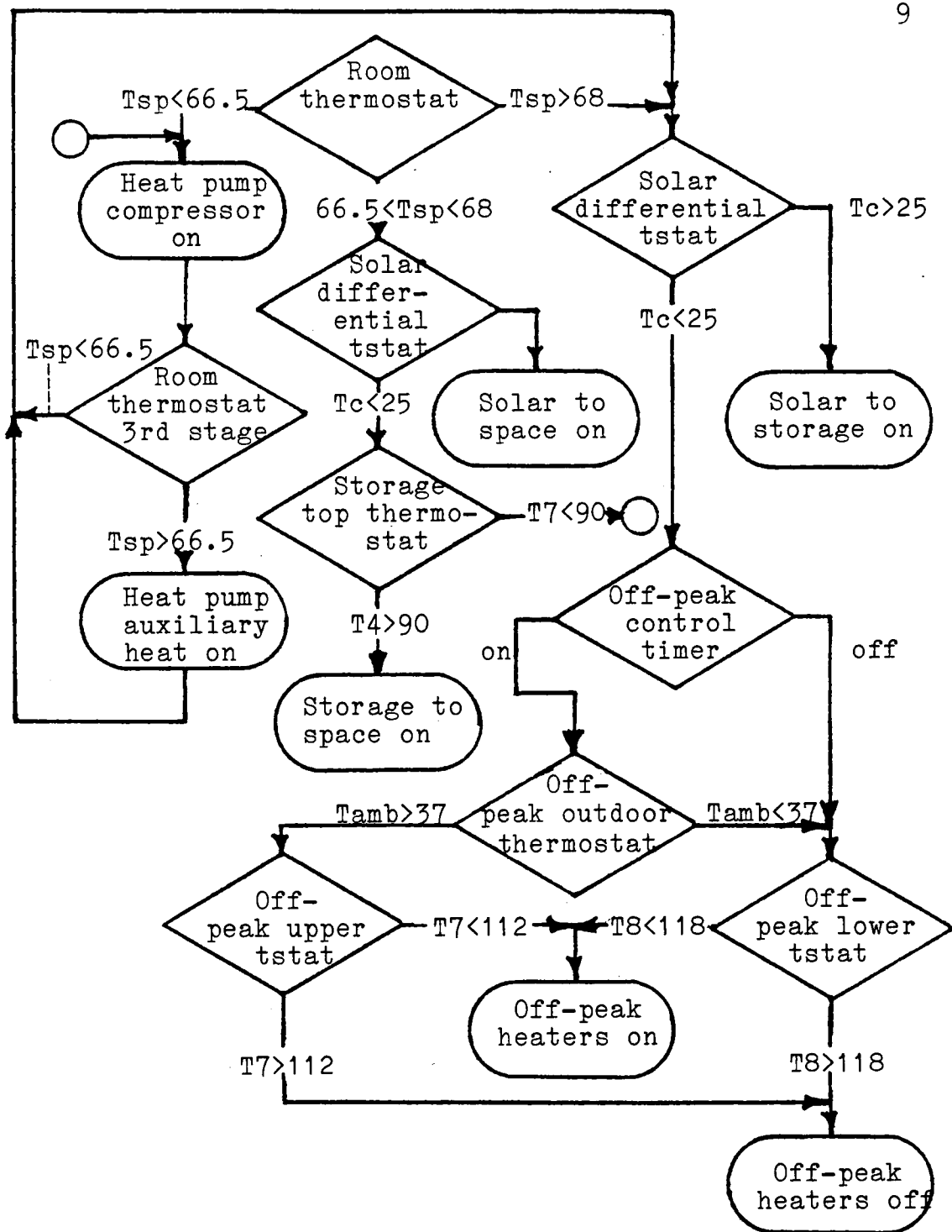


Figure 3. Experimental System Control Logic

Modeling Methods

Although several simple graphical methods are available for analyzing solar heating systems, the complexity of the experimental system exceeds their modeling capabilities. Several more versatile modeling methods are available, all based on hour-by-hour simulation of the performance of each component of a heating system. The most widely used of the methods, a computer program called TRNSYS [7], was chosen for the simulation of the experimental system. TRNSYS, developed by the University of Wisconsin, includes sufficient capability to model all of the components of the experimental system. The component models are discussed and compared with experiment in the following chapter.

CHAPTER II

COMPONENT MODELS

The TRNSYS computer program includes models to represent all of the components of the experimental system. However, the heating system model, as initially formulated from standard TRNSYS models, exhibited convergence problems and excessive computer execution time. These difficulties were avoided by reformulation of the computer routines, generally using similar physical models. The likely source of the convergence problems was found to be on/off control functions which induced oscillations in predictor-corrector integration routines. The integration routines were then changed where possible from iterative solutions to direct solutions, and convergence problems were greatly reduced. The most significant changes made during the modification of integration routines were in the rock bed heat transfer model.

Rock Bed Model

Several heat transfer models for an air-rock packed bed have been published [8,9]. The developers of the TRNSYS program have described [8,10] the physical model most widely used, which includes the following assumptions:

1. One-dimensional air flow.
2. Constant properties.
3. Negligible temperature variations within the rock particles (for $Bi < 0.1$ [10], satisfied for the experimental system).
4. Negligible transverse temperature gradients.
5. Negligible fluid heat capacity.
6. Negligible radiation.
7. Gravel approximately homogeneous for purposes of calculating conduction.
8. Negligible free convection (due to a stratified temperature gradient).

Using these assumptions, an energy balance may be written for a rock element of length Δx . The rock element is shown schematically in Figure 4, with positive x defined in the flow direction, either up or down. An energy balance applied to the rock element gives:

$$\dot{Q}_{\text{gain}}(x) = \dot{Q}_{\text{cond}}(x) - \dot{Q}_{\text{cond}}(x+\Delta x) - \dot{Q}_{\text{loss}}(x) + \dot{Q}_{\text{conv}}(x) \quad (1)$$

$$\dot{Q}_{\text{gain}}(x) = A \Delta x (\rho C) r \frac{\partial T_r(x)}{\partial t} \quad (2)$$

$$\dot{Q}_{\text{cond}}(x) - \dot{Q}_{\text{cond}}(x+\Delta x) = -krA \frac{\partial T_r(x)}{\partial x} + krA \frac{\partial T_r(x+\Delta x)}{\partial x} \quad (3)$$

$$\dot{Q}_{\text{loss}}(x) = UA \Delta x (T_r(x) - T_{\infty}) \quad (4)$$

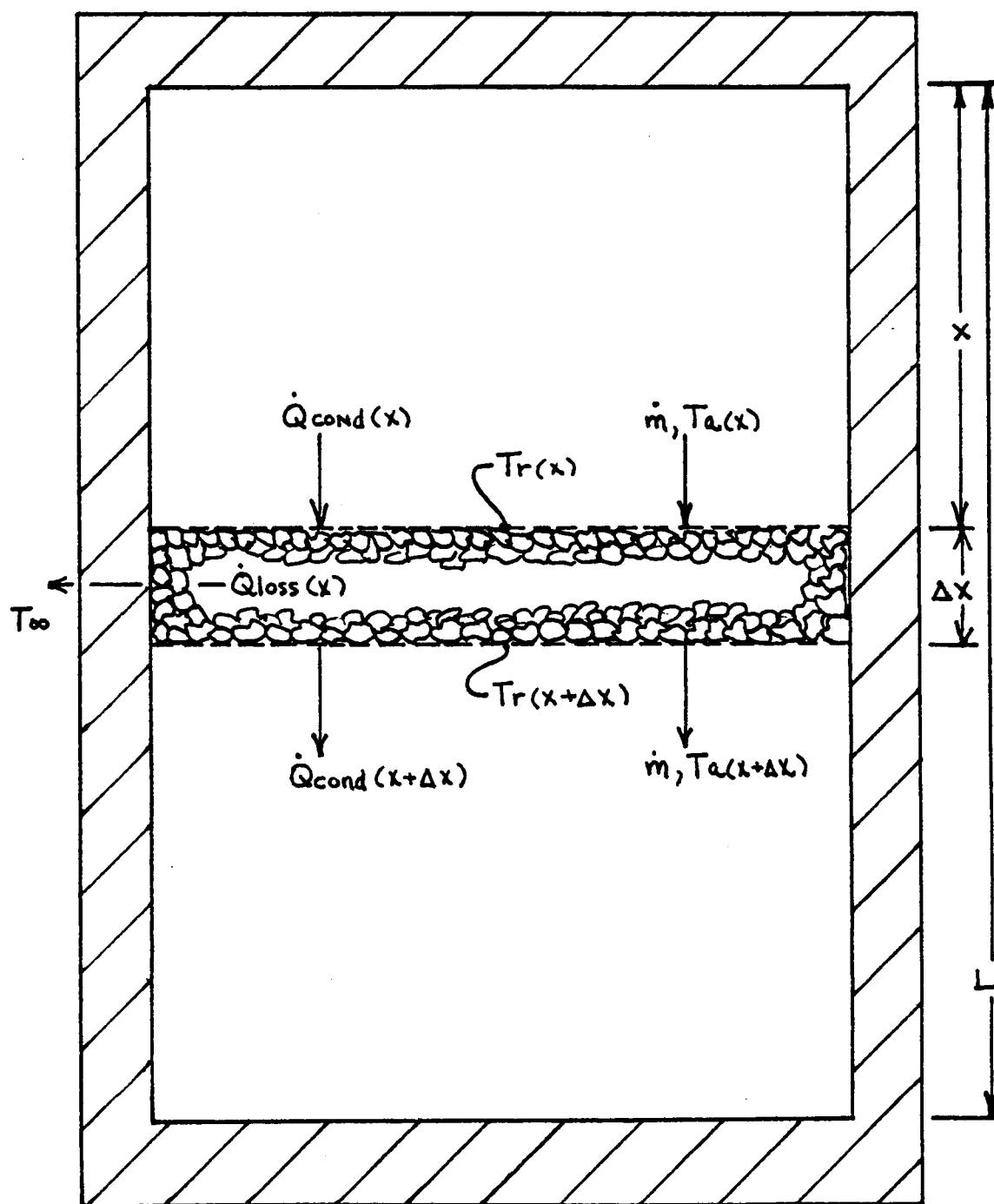


Figure 4. Schematic of Rock Bed Element

$$\dot{Q}_{\text{conv}}(x) = hcAs(Ta(x) - Tr(x)) \quad (5)$$

For the air occupying the void fraction in the rock:

$$\dot{Q}_{\text{conv}}(x) = hcAs(Ta(x) - Tr(x)) = \dot{m}Cp(Ta(x) - Ta(x+\Delta x)) \quad (6)$$

Equations (1) through (5) may be combined, and x shrunk to infinitesimal size:

$$A\Delta x(\rho C)r\frac{\partial Tr(x)}{\partial t} = krA\Delta x\frac{\partial^2 Tr(x)}{\partial x^2} + hcAs(Ta(x) - Tr(x)) - UP\Delta x(Tr(x) - T_\infty) \quad (7)$$

Similarly for the air:

$$hcAs(Ta(x) - Tr(x)) = -\dot{m}Cp\Delta x\frac{\partial Ta(x)}{\partial x} \quad (8)$$

In the TRNSYS program, Equations (7) and (8) are approximated by a standard modified-Euler difference routine. The convection coefficient coupling the two equations is assumed to be infinite, and the solution presents no difficulty. The assumption of infinite hc is based on the unimportance of its magnitude to the simulated heat flows [8]. However, hc was found to be significant in determining the temperature profile in the rock bed. Since the experimental system, unlike most solar air systems, includes four temperature sensors in the rock bed as inputs to the control system, a more accurate representation of the temperature profiles was desirable. The convection coefficient hc was then reintroduced to the model when the model was reformulated

for an integration routine with a direct solution. Using an experimental correlation [10] for the product of the convection coefficient and rock surface area, $hcAs$:

$$hcAs \left[\frac{BTU}{hr F} \right] = 0.7882 A \Delta x \left(\frac{\dot{m}}{A D_r} \right)^{0.7} \quad (9)$$

Equations (7), (8), and (9) may be combined and solved directly using finite difference techniques. With a modified implicit formulation, Equation (7) becomes:

$$\begin{aligned} \frac{2A\Delta x(\rho C)_r}{\Delta t} (T_{r,i}^{n+1} - T_{r,i}^n) = & hcAs(2T_{a,i} - T_{r,i}^{n+1} - T_{r,i}^n) + \\ & - \frac{k_r A}{\Delta x} (T_{r,i+1}^{n+1} + T_{r,i+1}^n + T_{r,i-1}^{n+1} + \\ & T_{r,i-1}^n - 2T_{r,i}^{n+1} - 2T_{r,i}^n) - \\ & UP\Delta x (T_{r,i}^{n+1} + T_{r,i}^n - T_{\infty}^{n+1} - T_{\infty}^n) \quad (10) \end{aligned}$$

Examination of Equation (10) reveals that the rock temperature at a node in the bed depends upon the following unknowns: (1) the air temperature at the node at the current timestep, $T_{a,i}$; (2) the rock temperature at the upstream node (in the flow direction) at the current timestep, $T_{r,i-1}^{n+1}$; (3) the temperature of the surroundings at the current timestep, T_{∞}^{n+1} ; and (4) the rock temperature at the downstream node (in the flow direction) at the current timestep, $T_{r,i+1}^{n+1}$. Unknowns (1) and (2) may be found through one boundary condition (the inlet air temperature) and a finite difference representation of Equation (8):

$$hcAs(2T_{a,i} - T_{r,i}^{n+1} - T_{r,i}^n) = 2\dot{m}C_p(T_{a,i-1} - T_{a,i}) \quad (11)$$

Unknown (3), the space temperature, may be found through an iterative loop containing the entire heating system model. A more difficult problem is the downstream rock temperature, unknown (4). Although the entire system of rock and air temperatures may be solved directly by reduction methods, significant computational effort is required. A more economical method is the application of an order-of-magnitude analysis. Since the downstream rock temperature appears only in the conduction term, and conduction coefficients for a rock bed are small in relation to thermal mass, the conduction term may be calculated using rock temperatures from the previous timestep. For example, using values from the experimental system, the dimensionless ratio of the coefficients of conduction and thermal mass is:

$$\frac{k_r \Delta t}{\Delta x (\rho C)_r} = 0.043 \quad (12)$$

At any one timestep the conduction term is therefore not extremely significant and may be calculated using the temperature values from the previous timestep. With that simplification and the inlet air temperature boundary condition, the rock bed temperatures may be solved for directly by combining Equations (10) and (11) and proceeding in a downstream direction.

The preceding rock bed model has been coded in FORTRAN and incorporated in the complete solar system model (shown as Appendix A). In accordance with published experimental results [11], the model has been modified to include plenum heat losses from the first and last nodes, in a similar manner to the perimeter loss term given in Equation (10).

The finite difference model as developed above was found to significantly decrease convergence problems and execution time and was therefore used in the complete heating system model. To give an indication of the model's accuracy, temperature profile results from the model are compared with experimental temperature profiles in Figure 5. Results are shown as mean simulated and experimental temperature profiles, with experimental profiles taken as the mean of about 150 profiles from data hours following off-peak charging of the rock bed. For this comparison the model was supplied with measured heat flows from those hours, temperature profiles from preceding hours, and measured rock bed parameters. Also shown is the mean simulated profile calculated by the model using an infinite convection coefficient, in an attempt to approximate the results that would be generated by the TRNSYS model.

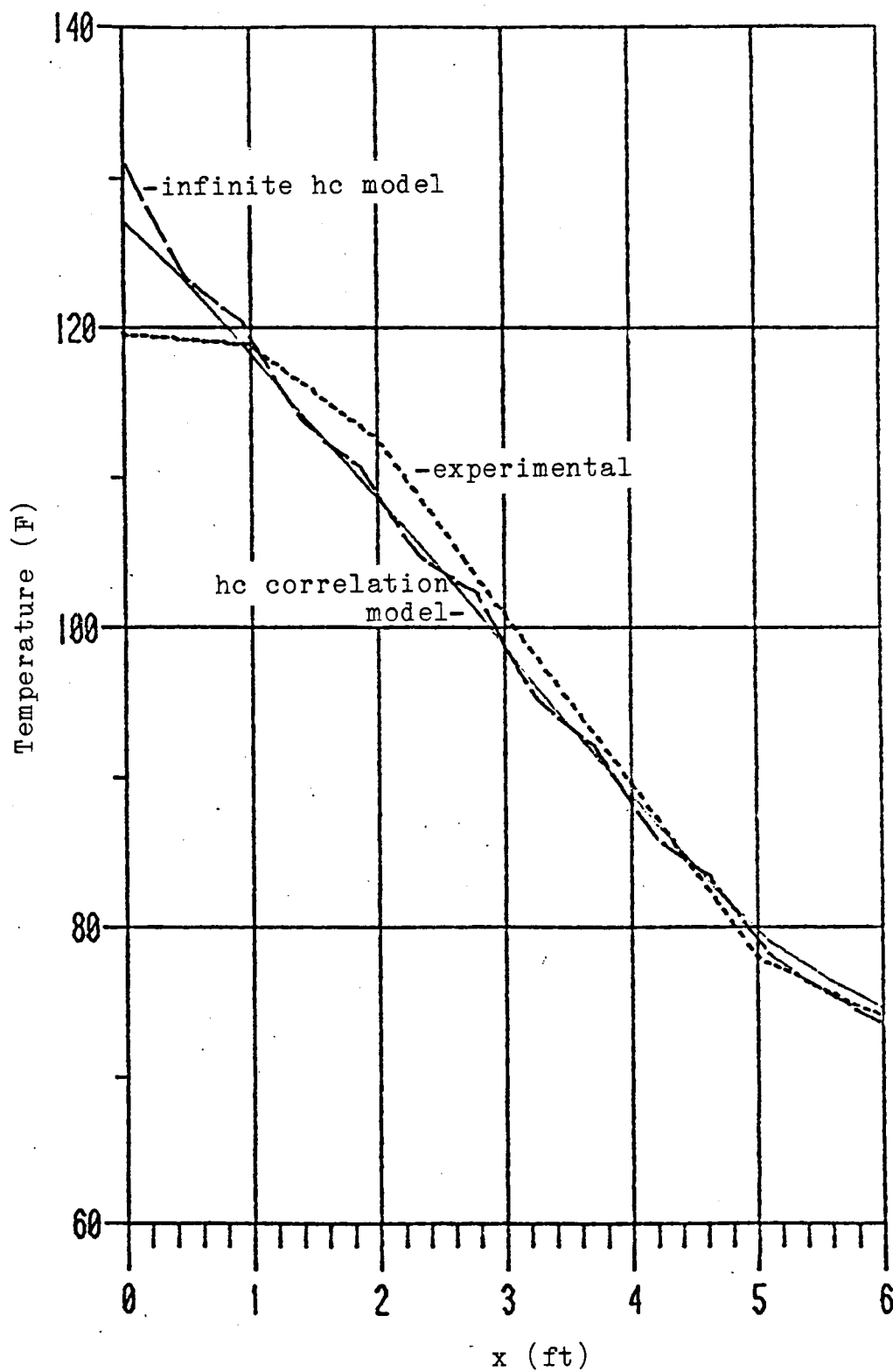


Figure 5. Mean Rock Bed Temperature Profiles, Experimental and Simulated by Two Models

As shown in Figure 5, the model developed above produces slightly more accurate temperature profiles than the infinite hc model. Although neither model is extremely accurate, the model developed above is clearly more accurate in the most significant areas at the inlet and outlet of the rock bed, where exiting air temperatures are determined in up-flow or down-flow modes. Both rock bed models were run for the comparison at a nodal spacing (Δx) of approximately 0.5 feet. Closer spacing of the nodes did not produce any appreciable gain in accuracy for either model.

The rock bed model developed above was used for the simulation of the complete heating system, since the model was the most accurate that was available. Nodal spacing was maintained at approximately 0.5 feet for the heating system simulation.

Space Heating Load Model

A space heating load may be approximated as the sum of an infiltration component due to air exchange and a conduction component. Several methods are available for simulating each component, and none of the methods has been clearly established as the best.

For some time the standard method for calculating conduction loads has been the ASHRAE UA-factor method [12], a steady-state parallel resistance network method.

The UA-factor method was developed for calculating maximum design loads used in sizing equipment and does not take into account the conduction time lag due to construction thermal mass. More recent methods, such as the response factor method [12] and various finite-difference algorithms, have attempted to account for transient effects in wall and roof conduction. Such methods are not usually applied to very light construction such as that of the experimental system, so a UA-factor was considered adequate for the heating system simulation. The accuracy of any of the conduction calculation methods is difficult to determine due to a lack of experimental heating systems with sufficiently comprehensive instrumentation to determine transient effects.

Several methods are also available for calculating infiltration loads. The basic mechanism is understood as a combination of wind pressure and stack effect, but evaluation of the actual magnitude is difficult. Calculation of infiltration from basic principles requires knowledge of total crack length, width, and distribution throughout the building, as well as complete pressure distribution. Variations in construction quality, building exposure, and building geometry make such calculations prohibitive. In order to simplify the calculations, a constant infiltration rate is commonly assumed, depending on construction type and building

volume. This procedure, known as the air-change method [12], has been shown to be significantly in error for the calculation of hourly heating loads [13].

Since the experimental heating system was installed in a converted garage with a relatively high infiltration load, a more accurate infiltration modeling method was desirable. Examination of the governing infiltration equations [12] reveals that the flow rate should be approximately proportional to windspeed and indoor-outdoor temperature differential. Since infiltration heating load is the infiltration volume times the indoor-outdoor enthalpy difference, the following equation results [14]:

$$\dot{Q}_{inf} = (h_{sp} - h_{amb})(C_1 + C_2(T_{sp} - T_{amb}) + C_3 V_{wind}) \quad (13)$$

Since weather data were being collected on site, Equation (13) with some modification could be fitted by regression analysis to experimental data. The following assumptions were made to simplify the data analysis:

1. The heat delivered to the space, as measured by the data acquisition system, was assumed to be equal to the space heating load.
2. Humidity effects were neglected, so indoor-outdoor enthalpy difference became proportional to indoor-outdoor temperature difference.

3. Conduction heating load was assumed proportional to instantaneous indoor-outdoor temperature difference.

These assumptions allowed the combination of infiltration and conduction loads into a single UA-factor, a function of weather variables:

$$\begin{aligned}
 UA &= \frac{\dot{Q}_{load}}{T_{sp} - T_{amb}} \\
 &= C_1 + C_2(T_{sp} - T_{amb}) + C_3 V_{wind}
 \end{aligned}
 \quad (14)$$

The unknown constants in Equation (14) were found by a multivariable regression analysis using ten weeks of hourly experimental data. Since heating load was approximated by the measured heat delivered to space, the hourly heat load and weather data were averaged over 24-hour periods. At any one hour the measured heat delivered to space might be considerably different from the heating load, due to thermostat operation.

Using the daily average data, the completed regression analysis resulted in the following equation:

$$\begin{aligned}
 UA \left[\frac{BTU}{hr \cdot F} \right] &= 534 + 17.5(V_{wind}) \left[\frac{mi}{hr} \right] + \\
 &\quad 1.78(T_{sp} - T_{amb})[F]
 \end{aligned}
 \quad (15)$$

This regression equation, when compared to the daily average experimental data, was found to have a standard deviation of 61.3 BTU/hr/F. For comparison, the best constant-UA model (UA=661.6 BTU/hr/F) had a standard deviation of 86.8 BTU/hr/F, about 40% higher.

Equation (15) was modified slightly by the assumption of a constant space temperature for purposes of calculating infiltration. The modification simplified calculation by uncoupling the space temperature from the UA-factor and did not significantly affect the accuracy of the correlation. Replacing the variable space temperature in Equation (15) by the ten-week mean (68.3 F):

$$UA \left[\frac{\text{BTU}}{\text{hr F}} \right] = 556 + 17.5(V_{\text{wind}}) \left[\frac{\text{mi}}{\text{hr}} \right] + 1.78(T_{\text{amb}})[F] \quad (16)$$

Several additional variations were attempted in the regression model to improve the fit. Another independent variable, solar radiation, was included in the regression analysis to account for solar heating of the exterior building surfaces. Solar radiation was dropped from the model, as it was found to have essentially zero correlation with the space heating load. In another attempted modification of the model, wind velocity was replaced as an independent variable by the wind velocity raised to the power 1.3, in accordance with the governing equations [12]. The change did not improve the correlation and was also dropped from the model. A third possible modification was a change in the 24-hour period used for averaging the experimental heating load data. Changes in the period from 1 hour to 48 hours produced small variations in the regression constants, and the

24-hour averaging period produced the most accurate correlation by comparison of the simulated and experimental heating loads over long time periods.

An effective thermal mass was added to the heating load model to provide a reasonable estimate of the hour-to-hour variation in space temperature, since, with a thermostatic control system, the simulated heat delivered to space generally did not equal the simulated space heating load. The effective thermal mass was most significant in simulating the operation of the three-stage room thermostat and did not greatly affect the numerical values of the heating load. Since the experimental system lacked instrumentation to determine mass effects, effective thermal mass was obtained by trial-and-error comparison of simulation and experimental results.

Internal Heat Generation

The space heating load as defined above is the total heat loss from the building and in most buildings includes more energy than that delivered by the building's heating system. The building containing the experimental system was unoccupied and contained a small timed resistance heater designed to simulate the internal generation of people and appliances in an occupied residence. The heat generation varied daily from hour to hour and was simulated by its hourly mean experimental

value. Reduced usage of the experimental building as the winter progressed caused a steady decrease in internal heat generation which was simulated by an additional correction factor.

Solar Collector Model

The most widely used method of describing the performance of a solar collector is the Hottel-Whillier-Bliss model [10]:

$$\eta = \frac{\dot{Q}_c}{IA} = Fr\tau\alpha - \frac{FrU_L}{I}(T_{f,i} - T_{amb}) \quad (17)$$

Performance data for solar collectors are typically reported in terms of this model as the constants $Fr\tau\alpha$ and FrU_L . Performance data for the collectors used in the experimental system were not available, as the collectors were prototype models and had not undergone standard testing in that form. However, performance constants were obtained from experimental data by regression analysis. Constants were obtained from 47 data points for which the experimental system had collected solar energy for one full hour. Although the data had considerable spread about the regression line, a reasonable correlation was obtained by averaging of adjacent data points, thus reducing transient effects. The correlation obtained from the averaged data was:

$$\begin{aligned} Fr\tau\alpha [\text{dimensionless}] &= 0.48 \\ FrU_L \left[\frac{\text{BTU}}{\text{hr F ft}^2} \right] &= 0.56 \end{aligned} \quad (18)$$

Equation (18) fit the time-averaged experimental efficiency data with a normalized standard deviation, $\frac{\sigma}{\mu}$, of 3.0% and was used in the heating system simulation. Although standard collector tests have not been performed for the prototype modes used in the experimental system, Equation (18) does appear to represent reasonable performance constants for a high-efficiency air-heating solar collector.

Although thermal mass effects were evident in the operation of the collector array [15], Equation (18) was found to produce accurate values of total solar energy collection over long time periods. The thermal mass of a solar collector is known to reduce energy collection, particularly under marginal weather conditions which produce on-off cycling. However, the heating season energy totals are primarily influenced by the energy collected under very favorable conditions when continuous operation creates approximately steady-state conditions. In addition, the time constant [10] of the collector array is only about 15 minutes as calculated from experimental performance constants. The steady-state model therefore appeared to be an adequate representation for long-term analysis.

Heat Pump Model

The manufacturer's published steady-state performance data [16] were found to produce reasonable operating performance estimates for the air-to-air heat pump. The published data were reported as capacity and Coefficient of Performance (COP) at 5 F intervals of ambient temperature. As shown in Figure 6, a curve fitted to the manufacturer's COP data corresponds reasonably well with experimental COP data. Capacity data were more difficult to determine experimentally but also appeared to be adequately approximated by the manufacturer's data.

The curves fitted to the manufacturer's data were then used in the complete heating system model, and total heat pump seasonal performance was represented within 3% of experimental values when the continuous 60 W power draw of the crankcase heater was included in the model. The crankcase heater was found to significantly increase the seasonal power consumption, primarily due to the low load factor of the heat pump (run time was about 10% of that possible).

Although transient performance for heat pumps has usually been reported to be below manufacturer's steady-state data, the heat pump in the experimental system generally operated in the sub-freezing temperature region, where steady-state data have been found to adequately approximate actual performance [17]. The

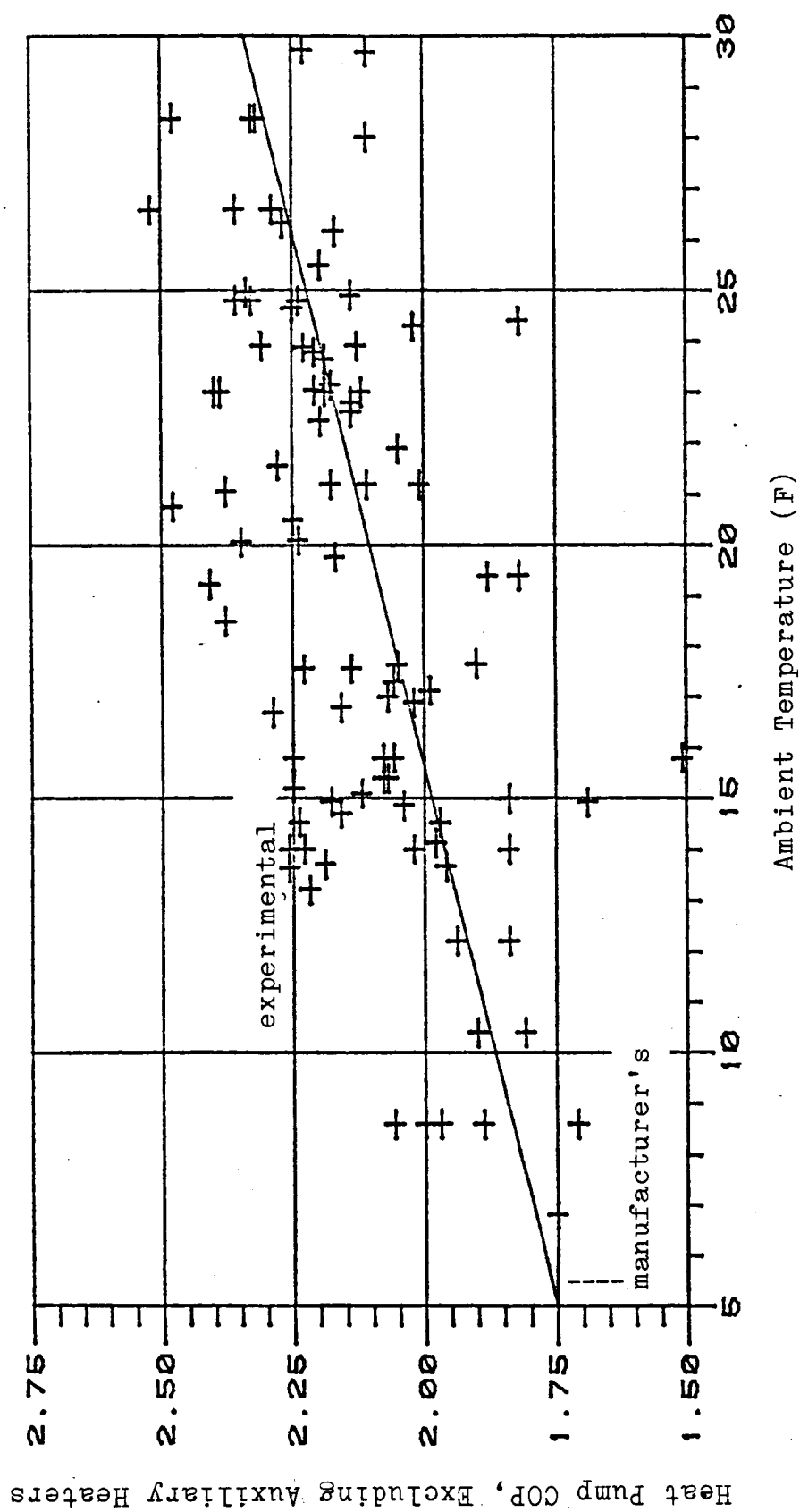


Figure 6. Heat Pump COP, Manufacturer's and Experimental Data

cyclic losses experienced in normal heat pump operation above the balance point were not significant for the experimental system, which was controlled to use the heat pump only during severe weather.

Room Thermostat Model

The three-stage room thermostat model was adapted from a two-stage model in the TRNSYS program. The model includes a deadband for two stages and hysteresis. The third stage of heating, heat pump auxiliary, would normally be controlled primarily by the outdoor anticipator. The anticipator had been shorted in the experimental system to diminish on-peak heat pump operation, and a constant third stage differential (without consideration of ambient condition) was found to adequately simulate the modified thermostat.

One thermostat parameter, the deadband of 1.5 F, was available from manufacturer's data [18], but information was not available on the remaining parameters. Since data reduction or calibration measurements were impractical at the required precision, several parameters were found by trial-and-error comparison of simulated and experimental heating system performance results. This method was used to evaluate hysteresis (1.4 F), third-stage differential (1.25 F), and effective room thermal mass ($42000 \cdot \Delta t$ BTU/F). The effective room

thermal mass was significant in the operation of the thermostat, since it determined the space temperature swings at each timestep.

Several additional features were added to the thermostat model to speed the convergence of the iterative loop which solved for system operational mode and space temperature. The additional features provided estimates of the system operational mode at the beginning of the iterative process or after a failed iterative process, a condition occurring in about 2% of the timesteps.

Descriptions of much more complex and presumably more accurate room thermostat models have been published [19], but a literature search did not reveal a model that could be implemented under conditions of relatively large simulation timesteps and scarcity of known thermostat parameters. Validation of the simple model used had to be based on overall simulation accuracy, since comparable experimental thermostat results were not available. Thermostat parameters, as assigned by comparison of simulation results and experiment, did appear representative of published results for comparable thermostats [20].

Ductwork Model

The experimental system included ductwork and duct heaters which, in operation, lost heat to surrounding spaces by conduction through duct walls and heater enclosures. Since all ductwork and heaters were located inside conditioned space, the duct "losses" were actually uncontrolled transfers to the space. The magnitude of the "losses" then became significant during charging of the rock bed thermal storage, when an appreciable fraction of the heat directed to storage was instead delivered to the space.

Since the ductwork had been found to achieve steady-state conditions in several minutes in calibration tests, ductwork thermal mass was neglected in estimating the magnitude of the "losses." Applying the resulting steady-state heat balance to a cross-section of ductwork with air flow:

$$UP(T_a(x) - T_{\infty}) = \dot{m}C_p(T_a(x) - T_a(x + \Delta x)) \quad (19)$$

Allowing Δx to shrink to infinitesimal size and solving the resulting differential equation:

$$\frac{T_a(x) - T_{\infty}}{T_a(0) - T_{\infty}} = \exp \left[\frac{-UPx}{\dot{m}C_p} \right] \quad (20)$$

By a heat balance on the rock bed, the energy delivered to storage is:

$$\dot{Q}_d = \dot{m}C_p(T_a(L) - T_{out,st}) \quad (21)$$

Similarly, the energy directed to storage by the off-peak heater or solar collector is:

$$\dot{Q}_h = \dot{m}C_p(T_{a(0)} - T_{out,st}) \quad (22)$$

Since $T_{out,st} = T_w$, Equations (20), (21), and (22) may be combined to give:

$$\dot{Q}_d = \dot{Q}_h \cdot \exp \left[\frac{-UPL}{\dot{m}C_p} \right] \quad (23)$$

The duct "losses" could then be approximated as a constant fraction of the heat directed to storage.

UA-factors for the ductwork and rock bin had been calculated by standard ASHRAE methods [12], and flow rates had been measured [15]. The conduction "losses" as calculated by Equation (23) and a similar loss for the rock bin were found to account for 79% of the measured heat "losses" as determined by an energy balance on the rock bin for a ten week period. The nominal UA-factors for the duct and rock bin were then increased by 27% as a reasonable method of allocating the difference, whether due to conduction, damper leakage, or sensor error. The modified UA-factors were then used to calculate duct and rock bed "losses" in the heating system simulation.

Model Formulation

The heating system component models as described above were combined into a single FORTRAN subroutine and incorporated into the TRNSYS program. The TRNSYS program was used only as an executive routine to control timing and input/output operations to the FORTRAN subroutine, except for some processing of solar radiation data performed by TRNSYS and described in the following chapter.

CHAPTER III

SYSTEM MODEL VALIDATION

Results from the completed heating system model have been compared to experimental results for a ten week period in midwinter 1979-80. The ten weeks were the longest period of continuous experimental data collection and included about 60% of the annual degree days. Data from the same period were used to correlate heating load and solar collector performance as previously described.

Total simulated and experimental heat and power flows for the period are shown in Table 1 for each mode of operation. As shown in Table 1, the simulation provided close approximations for most of the heat and power flows. An overall simulation accuracy may be approximated by a normalized standard deviation, $\frac{\sigma}{\mu}$, taken on the set of heat flow totals for each mode of operation for the ten week period and on the similar set of power flow totals. For the results shown in Table 1, $\frac{\sigma}{\mu}$ of the heat flows is 5.3% and of the power flows is 7.8%. This compares with the heat flow measurement accuracy of 5% and power consumption measurement accuracy of 1% [15].

Mean thermal storage temperature for the ten week period also agreed well with experiment at +0.8 F mean error. As shown in Figure 7 for a portion of the period, the hourly storage temperature deviated more significantly

Table 1

Simulated and Experimental Heat Flow and Power
Consumption, 12/17/79 to 2/24/80

Mode	<u>Heat Flow</u>		<u>Power Consumption</u>	
	<u>Exp.</u>	<u>Sim.</u>	<u>Exp.</u>	<u>Sim.</u>
	⁶ (10 BTU)		(kWh)	
Solar to Space	2.17	2.19	40	40
Solar to Storage	4.78	4.78	90	90
Off-peak to Storage	14.03	14.09	4110	4130
Off-peak to Space	1.26	1.27	370	370
Heat Pump Compressor	5.35	5.48	800	840
Heat Pump Auxiliary	0.81	0.97	230	280
Storage to Space	18.03	17.79	290	260
Storage Loss	1.40	1.38	--	--
System Total	29.02	29.07	5930	6010

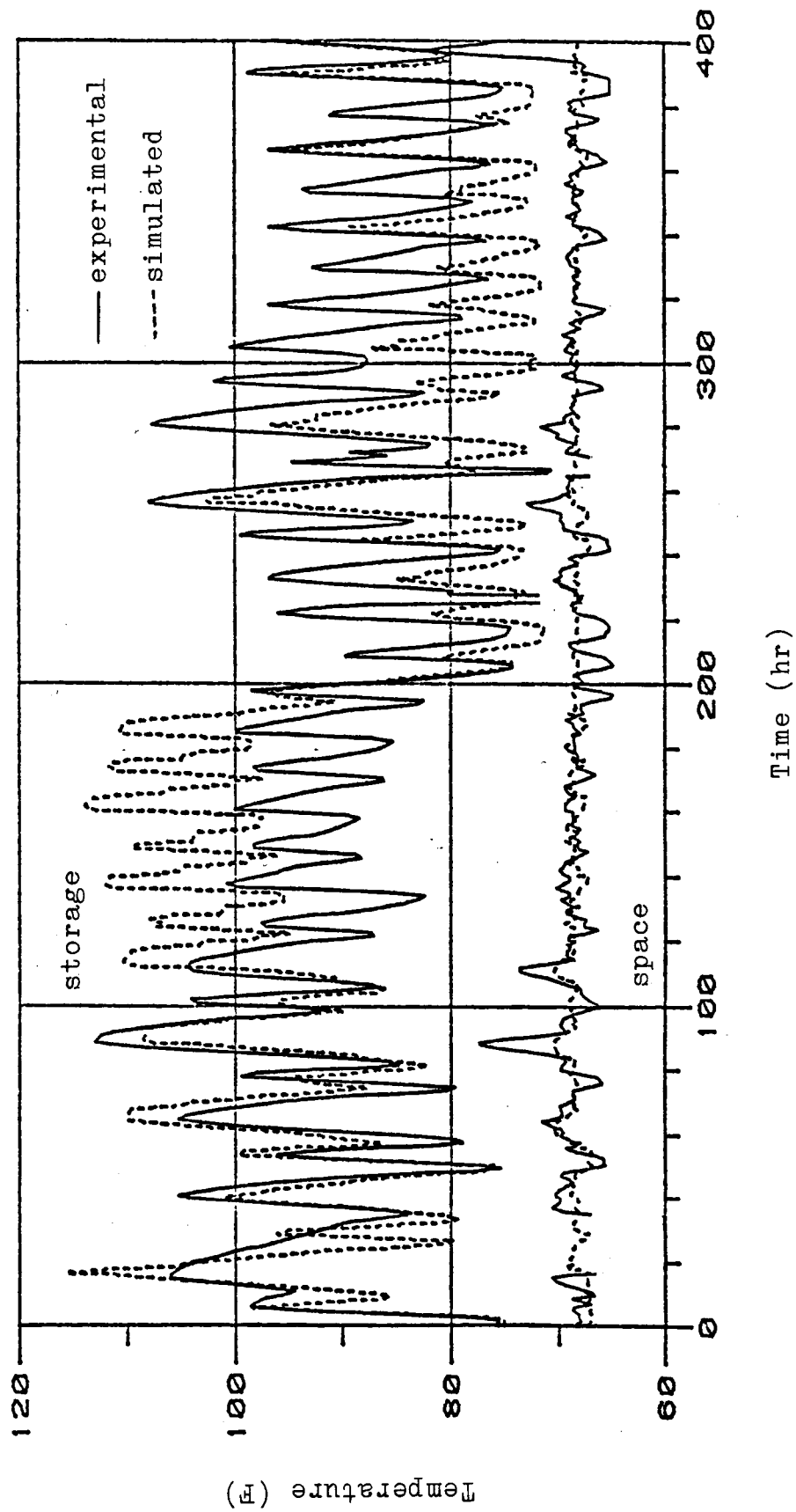


Figure 7. Simulated and Experimental Storage and Space Temperatures, 12/17/79 to 1/2/80

from experimental results. Simulated and experimental space temperatures, also shown in Figure 7, corresponded more closely, as both were controlled within limits by the room thermostat, model or hardware.

Simulation Timestep

The simulation results given above were calculated using a 30 minute timestep. Larger timesteps were unable to accurately follow the off-peak control timing and were not used, but simulation results were obtained at smaller timesteps of 15 and 7.5 minutes. Using 15 minute timesteps, the normalized standard deviation, $\frac{\sigma}{\mu}$, of the total heat flows in each operating mode for the ten week simulation period was 7.2%, and for power consumption 8.6%. For 7.5 minute timesteps the deviations were 12.4% and 7.1% respectively, and for 30 minute timesteps 5.3% and 7.8%, as given previously. Since the differences in accuracy at various timesteps were small, the largest timestep of 30 minutes was used for the results presented here. The slightly better accuracy at the largest timestep was probably due to its use in the simulations which determined the empirical constants in the room thermostat model. As previously described, room thermostat hysteresis and third-stage differential and effective space thermal mass were obtained by comparison of simulated and experimental heating system performance, and therefore may have been dependent to some degree on

the simulation timestep. The time-consuming method of determining the empirical constants was not repeated at the various timesteps, so the constants were not changed when the timestep was varied. An exception was the effective room thermal mass, which was found to produce approximately correct results when varied directly with the timestep.

Weather Data

The validated heating system model required a source of temperature, insolation, and windspeed data for comparative heating season simulations given in the following chapter. The weather data collected on site were useful for generating results for validating the model by comparison with experiment. However, the 1979-80 winter season included some untypically severe weather and did not appear suitable for long-term performance predictions. Long-term average weather conditions were also not suitable since the averaging process removes the essential variability to which solar systems react in a non-linear fashion. Probably the best available weather data were the Typical Meteorological Years (TMY's), a collection of the most "typical" weather months from the period of record for most NOAA weather stations [21]. The "typical" months were chosen from the 25-year record by an arbitrary statistical weighting which emphasized

typicality of solar radiation and ambient temperature. The TMY's have been found to adequately approximate the long-term record for their intended purpose of solar energy simulations [22].

The Knoxville airport weather station was the nearest source (approximately 7 miles) of TMY data to the experimental system. At the time of the simulation study, the Knoxville TMY months had been chosen, but the data had not yet been distributed, so the TMY was assembled from SOLMET [23] data tapes containing the complete Knoxville weather record in a similar format. The SOLMET tapes included measured ambient temperature and windspeed. However, no insolation data had been recorded at Knoxville. The insolation values in the SOLMET tapes were generated by a correlation procedure from cloud cover data. The correlation, developed by NOAA contractors from measured insolation data at several sites, has been found to adequately predict solar radiation [24]. An additional correlation included in the TRNSYS program was used to convert the horizontal-surface radiation data to the tilted surface of the solar collectors. The correlation method used was found to be one of the two most accurate available methods [25]. Unfortunately, the correlated SOLMET insolation data end in 1976, before experimental data became available, so direct comparison of the correlated Knoxville SOLMET data and the insolation data taken on site was not possible.

CHAPTER IV

COMPARATIVE SYSTEM RESULTS

The performance of the experimental heating system has been simulated for an entire heating season using the verified system model and TMY data for Knoxville. The simulated seasonal performance of the experimental system has also been compared with that of several alternative heating systems. The systems include:

1. The experimental system as of February, 1980.
2. The experimental system with off-peak heater control made proportional to the outdoor temperature and with solar collector cut-off differential lowered from 25 F to 15 F (These were the most advantageous of several possible control changes in the experimental system that were tried by simulation).
3. The experimental system without off-peak heaters and with solar collector cut-off differential lowered from 25 F to 15 F (a parallel solar-assisted heat pump).
4. The experimental system without off-peak heaters, without solar collectors, and without the rock bed (a conventional heat pump).

Seasonal performance results for the four systems are shown in Table 2 as: (1) total heat delivered to the space by the heating system, not including internal heat generation; (2) total collected solar energy; (3) total on-peak and off-peak power consumption, neither including internal generation; and (4) power costs under time-of-day and standard electrical rates.

Time-of-Day Electrical Rates

Since analysis of heating systems is clearly an economic as well as a thermal problem, a limited economic comparison was performed. The comparison was based on annual power costs under a time-of-day electrical rate structure, a structure which was assumed in the design of the experimental system. Capital costs were not included in the economic analysis due to the number of additional variables introduced and to the number of somewhat arbitrary assumptions required.

The time-of-day rate structure used in the annual power cost comparisons was adapted from the structure used in a time-of-day rate test conducted in Knoxville by the Tennessee Valley Authority [6]. Winter on-peak times were defined for the test as 7 a.m. to 1 p.m. and 6 p.m. to 10 p.m. weekdays, with other times off-peak.

Table 2
Simulated Heating Season Performance of Experimental and Alternative Heating Systems

System	Heat Delivered to Space	Collected Solar Energy	Power Consumption		Time-of- Day Rates	Power Cost Standard Rates
			On-Peak	Off-Peak		
			(kWh)			
		⁶ (10 BTU)				
1. Experimental system	53.94	21.77	608	9038	\$230	\$330
2. System with improved controls	54.11	24.49	610	8424	\$217	\$309
3. Solar-assisted heat pump without off-peak capability	53.25	26.47	1502	3298	\$159	\$164
4. Conventional heat pump	49.09	0.0	2029	4801	\$223	\$234

Rates used in the test, set in 1977, were \$0.0382/kWh on-peak and \$0.0140/kWh off-peak. An assumption stated in the test report was that the time-of-day structure was designed to produce the same utility revenue as standard rates when 34% of actual usage was consumed on-peak. Based on this assumption, and the TVA 1980 standard residential rate of \$0.03419/kWh [26], the time-of-day rates have been updated to 1980 equivalents of \$0.0588/kWh on-peak and \$0.0215/kWh off-peak, with the same ratio of on-peak to off-peak rates as the 1977 rates. These time-of-day rates and the 1980 standard rate were used in power cost comparisons of the alternative heating systems.

Thermal and Economic Performance

As shown in Table 2, the simpler alternatives to the experimental system appear to have superior economic and thermal performance. Although the annual power cost of the experimental system may be reduced about 6% by improving the controls (system 2), the annual power cost may be further reduced 26% by eliminating the off-peak capability altogether (system 3). The conventional heat pump (system 4) is also slightly less expensive to operate than the experimental system, although slightly higher than the experimental system with improved controls (system 2).

On-peak power consumption is effectively limited by the two off-peak systems, but the rate differential is apparently insufficient to justify the penalty in total power consumption. As shown in Figure 8, the relative economics of the four systems are strongly influenced by the ratio of on-peak to off-peak electrical rates. Simulation results are shown in Figure 8 as relative power cost for the four systems as a function of on-peak to off-peak pricing ratio, varied from one to ten. A ratio of one corresponds to a standard electrical rate, and the ratio used by TVA as discussed previously was 2.63. Unity on the relative cost scale used in Figure 8 is defined as the cost of operating a conventional heat pump under a standard electrical rate.

Figure 8 shows that the experimental system, on an annual power cost basis, would not be competitive with a conventional heat pump unless the peak pricing ratio exceeded 3.5 and would not be competitive with the solar assisted heat pump (system 3) unless the ratio exceeded 6.5.

System 2, the proportionally controlled off-peak system, is somewhat more competitive economically. For the simulation results shown in Figure 8, control settings were varied for system 2 to minimize the annual cost at each pricing ratio (those results then do not correspond with Table 2). The system 2 results show that the

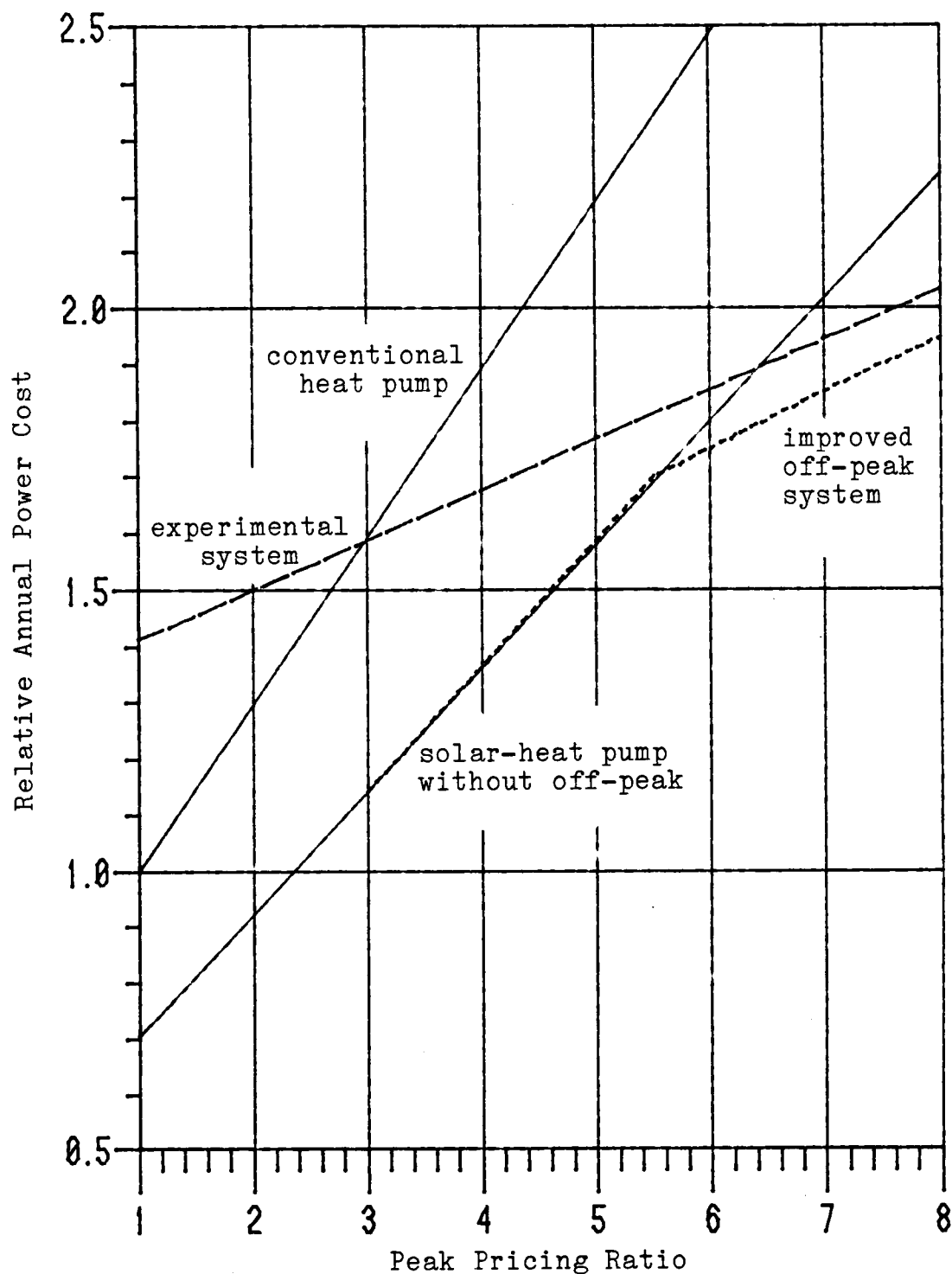


Figure 8. Relative Annual Power Cost for Experimental System and Alternatives at Various Peak Pricing Ratios

off-peak capability is not useful if the peak pricing ratio is less than 5, as the optimum operating condition would be essentially non-use of that capability.

In light of the relatively poor thermal performance of the experimental system, the effects of several variations in the system were simulated. The rock bed size was found to have very little effect on performance, as a +25% change corresponded to about -3% in annual power cost. Conduction losses from system components were also found to be relatively insignificant to overall system operation, as a +25% change in duct and rock bed insulation produced a -2% change in annual power cost. A similar +25% change in the size of the solar collector array was more significant in producing a -10% change in annual power cost, although such an increase might be excessive in capital cost.

Also simulated was an additional possible modification to the experimental system, the replacement of the off-peak resistance heaters by a heat pump. Such a replacement should theoretically decrease the excessive power consumption of the heaters. Simulation results were obtained using the proportionally controlled system 2 described above with the off-peak heater replaced by an additional heat pump identical to the one used for backup heating. Simulated performance was similar to that of system 3 as shown in Table 2. The heat pump in the

off-peak loop was found to store little energy, as its outlet temperature of about 95 F was insufficient to significantly charge the rock bin. Further development would apparently require either a high-temperature heat pump or a standard model and phase-change or other low temperature storage.

CHAPTER V

CONCLUSIONS

The model used to simulate the performance of the experimental heating system was found to produce acceptable accuracy, considering the complexity of the system and the limitations of the available component models. Models of the rock bed and space heating load were found to produce accurate total heat flows over extended periods, although both models produced significant deviations from experiment on an hourly basis. Other heating system components were modeled with acceptable long-term accuracy by steady-state models, although hourly results also deviated significantly. Development of more accurate component models appears to await, in part, the accurate and comprehensive data needed to validate the models. Many of the models used, not well validated, required extensive correlation and comparison with the experimental data taken on site.

Convergence problems in the heating system model were significantly reduced by reformulating the model for direct solution of the applicable differential equations. A related benefit was an approximate 80% reduction in computer execution time as compared to the iterative method.

Since the heating system model was verified for the simulation of a complex and well instrumented experimental heating system, the model could also be assumed with reasonable confidence to apply to the simulation of the simpler heating systems available commercially. The location, climactic conditions, and equipment of the heating system could then be varied systematically using the model, and the results of the single experiment could be extended by simulated comparisons with alternatives to the experimental heating system.

Heating system simulations of the experimental system and alternatives have shown that the off-peak capability of the experimental system may only be economically justifiable if the cost of peak period electricity rises to at least six times the cost of off-peak electricity. Such a rise has apparently not yet occurred, and the only apparent justification of the off-peak capability would be economic. The simulation results have also shown that the solar portion of the experimental system is justifiable thermally as an addition to a heat pump. Its economic justification requires further capital cost analysis.

LIST OF REFERENCES

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1. Freeman, T. L., J. W. Mitchell, and T. E. Audit. "Performance of Combined Solar-Heat Pump Systems." Solar Energy, 22 (1979), 125-135.
2. Oonk, R. L., L. E. Shaw, and H. H. Hopkinson. "Modelling of Combined Air Based Solar/Heat Pump Heating and Cooling Systems." Proceedings of the American Section of the International Solar Energy Society 1978 Annual Meeting, Vol. 2.1, 308-312.
3. Hoover, E. R., and C. B. Winn. "Design and Computer Simulation of Solar-Heat Pump Systems." Proceedings of the American Section of the International Solar Energy Society 1978 Annual Meeting, Vol. 2.1, 313-317.
4. Lorsch, H. G. "Effect of Solar Home Heating on Electric Utilities." Proceedings of the American Section of the International Solar Energy 1976 Annual Meeting, Vol. 9, 97-112.
5. Lorsch, H. G., R. L. Oswald, and R. E. Crane. "Control Strategies for Solar Heating/Cooling Systems that Reduce Utility Peak Loads." Proceedings of the American Section of the International Solar Energy Society 1978 Annual Meeting, Vol. 2.1, 499-505.
6. Knoxville Residential Time-of-Day Rate Test. Chattanooga: Tennessee Valley Authority, 1979.
7. TRNSYS, A Transient Simulation Program. Madison: University of Wisconsin Engineering Experiment Station Report 38-10, 1979.
8. Hughes, P. J., S. A. Klein, and D. J. Close. "Packed Bed Thermal Storage Models for Solar Air Heating and Cooling Systems." Trans. ASME: J. Heat Transfer, 98 (1979), 336-337.
9. Hamilton, N. I. "A Numerical Simulation of Reversible Heat Transfer in Packed Beds." Proceedings of the American Section of the International Solar Energy Society 1978 Annual Meeting, Vol. 2.1, 671-675.
10. Duffie, J. A., and W. A. Beckman. Solar Energy Thermal Processes. New York: Wiley, 1974.

11. Persons, R. W., J. A. Duffie, and J. W. Mitchell. "Comparison of Measured and Predicted Rock Bed Storage Performance." Solar Energy, 24 (1979), 199-201.
12. Handbook of Fundamentals. New York: American Society of Heating, Refrigeration, and Air-Conditioning Engineers, 1977.
13. Tamura, G. T. "The Calculation of House Infiltration Rates." Trans. ASHRAE, Vol. 85 Pt. 1 (1979), 58-71.
14. Sepsy, C. F., R. S. Blancett, and M. F. McBride. Heat Transfer Models and Energy Needs for Residential Homes. Washington, D. C.: National Technical Information Service, 1975.
15. Tomlinson, J. J. "Performance Evaluation of a Solar-Augmented Heat Pump with Offpeak Heating." Thesis, University of Tennessee, 1980.
16. Split System Heat Pump Model HPO36D. Norman, Okla.: Westinghouse Electric Corporation, 1979.
17. Didion, D. A., and G. E. Kelly. "New Testing and Rating Procedures for Seasonal Performance of Heat Pumps." ASHRAE Journal, 21 No. 9 (September 1979), 40-44.
18. BAY28X138 Weathertron Thermostat. Louisville, Ky.: General Electric Corporation, 1977.
19. McBride, M. F. "Measurement of Residential Thermostat Dynamics for Predicting Transient Performance." Trans. ASHRAE, Vol. 85 Pt. 1 (1979), 684-694.
20. Cape, R. C., and R. H. Tull. "Performance of Low-Voltage Thermostats." Trans. IEEE, Ind. and Gen. App., 5 (1969), 350-361.
21. Typical Meteorological Year. Asheville, N. C.: National Climactic Center TD-9734, 1979.
22. Freeman, T. L. "Investigation of SOLMET Typical Meteorological Year." Proceedings of the American Section of the International Solar Energy 1979 Annual Meeting, 2183-2187.
23. SOLMET. Asheville, N. C.: National Climactic Center TD-9724, 1979.

24. Hall, I. J., et. al. "Solar Radiation Model Validation." American Section of the International Solar Energy Society 1979 Annual Meeting, 2178-2183.
25. Lantz, L. J., and C. B. Winn. "Validation of Computer Models used for Predicting Radiation Levels." Proceedings of the American Section of the International Solar Energy Society 1978 Annual Meeting, Vol. 2.2, 693-697.
26. Residential Electricity, Water, and Gas Rates. Knoxville Utilities Board, 1980.

APPENDICES

APPENDIX A

SYSTEM MODEL FORTRAN SUBROUTINE

```

SUBROUTINE TYPE39(TIME,XIN,OUT,T,DTDT,PAR,INFO)
DIMENSION XIN(10),OUT(15),PAR(13),INFO(9),
1TSTG(21),TOLD(21),TA(21),IMO(13)
COMMON /LUNITS/ LUR,LUW,IFORM
COMMON /SIM/TIMEO,TFINAL,DELT
DATA IMO/13*0/
IF(INFO(7).GT.-1) GO TO 10
C FIRST CALL OF SIMULATION
XCHEK=PAR(1)
DET=PAR(2)
CPA=PAR(3)
ERR=PAR(4)
ICOUNT=IFIX(ERR)
IF(ICOUNT.LT.2)ICOUNT=2
ERR=ERR-FLOAT(ICOUNT)
IF(ERR.LT.1.OE-05) ERR=1.OE-05
FL1=PAR(5)
FL2=PAR(6)
FL3=PAR(7)
FL4=PAR(8)
FL6=PAR(9)
PS1=PAR(10)
PS2=PAR(11)
PS3=PAR(12)
CHP1=PAR(13)
CHP2=PAR(14)
CHP3=PAR(15)
CHP4=PAR(16)
CHP5=PAR(17)
CHP6=PAR(18)
CHP7=PAR(19)
TMINH4=PAR(20)
COP1=PAR(21)
COP2=PAR(22)
COP3=PAR(23)
COP4=PAR(24)
COP5=PAR(25)
COP6=PAR(26)
COP7=PAR(27)
TMINH5=PAR(28)
CAX1=PAR(29)
CAX2=PAR(30)
CAX3=PAR(31)
CAX4=PAR(32)
CAX5=PAR(33)

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CAX6=PAR(34)
CAX7=PAR(35)
TMINH6=PAR(36)
HST=PAR(37)
TMINX=HST+5.
TMAXX=HST-5.
DE1=PAR(38)
DE2=PAR(39)
DE3=PAR(40)
DE4=PAR(41)
STMIN=PAR(42)
DECOLL=PAR(43)
ACOLL=PAR(44)
FRTA=PAR(45)
FRULC1=PAR(46)
FRUL2=PAR(47)
FRULT=PAR(48)
UAON=PAR(49)
CVW=PAR(50)
XNVW=PAR(51)
CHT=PAR(52)
CTAMB=PAR(53)
CAPRM=PAR(54)
UAD2=AMIN1(PAR(55),FL2*CPA)
UAD6=AMIN1(PAR(56),FL6*CPA)
XLSTG=PAR(57)
ASTG=PAR(58)
DSTG=PAR(59)
C PAR (60) SEE BELOW
C PAR (61) SEE BELOW
    UAT=PAR(62)
C PAR (63) SEE BELOW
    UAB=PAR(64)
C PAR (65) SEE BELOW
    HC=PAR(66)
    HV=PAR(67)
    RHOCPR=PAR(68)
    NSTG=IFIX(PAR(69)+0.1)
    IF(NSTG.LT.3)NSTG=3
    IF(NSTG.GT.19)NSTG=19
    FNSTG=FLOAT(NSTG)
    UAS=PAR(60)/FNSTG
    DO 11 JA=2,NSTG+1
    TOLD(JA)=PAR(JA+68)
    TSTG(JA)=PAR(JA+68)
    TSTG(1)=TSTG(2)
    TOLD(1)=TOLD(2)
    TSTG(NSTG+2)=TSTG(NSTG+1)
    TOLD(NSTG+2)=TOLD(NSTG+1)
    TRM=HST
    DIR=0.

```

11

```

      IOSC=0
      IMODE=0
      INON=0
      TOLDRM=HST
10      IF(INFO(7).GT.0) GO TO 48
C INITIALIZATION ROUTINE
      TOLD2=TOLDRM
      TOLDRM=TRM
      KMODES=0
      DO 12 JB=1,NSTG+2
      TA(JB)=TSTG(JB)
12      TOLD(JB)=TSTG(JB)
      TAMB=XIN(1)
      HTOTAL=XIN(2)
      VWIND=XIN(3)
      YOFPK=XIN(4)
      QINT=XIN(5)
      NCONLW=IFIX(0.1+XIN(6))
      IF(NCONLW.GT.NSTG) NCONLW=NSTG
      IF(NCONLW.LT.1) NCONLW=1
      NCONUP=IFIX(0.1+XIN(7))
      IF(NCONUP.GT.NSTG) NCONUP=NSTG
      IF(NCONUP.LT.1) NCONUP=1
      STLCHT=XIN(8)
      STUCHT=XIN(9)
      TAMBOP=XIN(10)
C MAIN ITERATIVE LOOP
48      KCOUNT=ICOUNT
49      CONTINUE
50      DO 9999 ITERAT=1,KCOUNT
      TRMSAV=TRM
      IMODES=IMODE
      IF(ABS(FLOAT(ITERAT)/2.-FLOAT(ITERAT/2)).LT.0.25)
1 KMODES=IMODE
      IF(ABS(FLOAT(ITERAT)/2.-FLOAT(ITERAT/2)).LT.0.25)
1 TRMS2=TRM
      IMODE=0
      TINFS=PAR(61)
      TINFT=PAR(63)
      TINFB=PAR(65)
      QHP=0.
      PHP=0.
      QOP=0.
      POP=0.
      POPST=0.
      POPSP=0.
      QOPST=0.
      QOPSP=0.
      QAX=0.
      PAX=0.
      TINSP=TRM/2.+TOLDRM/2.

```

```

FINSP=0.
TIND=TA(1)
TOUTST=TA(NSTG+1)
IF(FLOWD.LT.1.0)TOUTST=TSTG(NSTG+1)/2.+
1TOLD(NSTG+1)/2.
FLOWD=0.
TINU=TRM/2.+TOLDRM/2.
FLOWU=0.
QSLSP=0.
PSLSP=0.
QSLST=0.
PSLST=0.
QSTSP=0.
PSTSP=0.
QLOSS=0.
DELTU=0.
C SYSTEM CONTROLLER
51 FRUL1=FRULC1+FRULT*TIME*DET
DTSLSLSP=(ACOLL/CPA/FL1)*(FRTA*HTOTAL-FRUL1*
1(TRM/2.+TOLDRM/2.-TAMB)-FRUL2*((TRM/2.+TOLDRM/2.-
1TAMB)**2.))
DTSLSLST=(ACOLL/CPA/FL2)*(FRTA*HTOTAL-FRUL1*
1(TOUTST-TAMB)-FRUL2*((TOUTST-TAMB)**2.))
DIR=-1.
IF(KCOUNT.EQ.ICOUNT) GO TO 57
IF(TOLDRM.GT.TOLD2) DIR=1.
GO TO 52
57 IF(ITERAT.GT.2) GO TO 56
IF(TRM.GT.HST-DE1/4.)GO TO 3000
IF(TRM.LT.HST-DE1*3./4.)GO TO 1000
GO TO 53
56 IF(TRM.GT.TOLDRM) DIR=1.0
52 IF(TRM.LT.HST-DE1+DIR*DE2) GO TO 1000
IF(TRM.GT.HST+DIR*DE2) GO TO 3000
53 IF(DTSLSLSP+TRM/2.+TOLDRM/2.-TOUTST.GE.DECOLL)
1GO TO 4000
54 IF(TSTG(2).GE.STMIN) GO TO 6000
C HEAT PUMP MODE
1000 IF(TAMB.LT.TMINH4) GO TO 2000
IMODE=1
QHP=CHP1+TAMB*CHP2+(TAMB**2.)*CHP3
COPHP=CHP4+TAMB*CHP5+(TAMB**2.)*CHP6
IF(COPHP.LT.0.1) COPHP=0.1
PHP=QHP/COPHP
TINSP=QHP/FL4/CPA+TRM/2.+TOLDRM/2.
FINSP=FL4
C AUXILIARY HEATERS
2000 IF(TRM.GE.HST-DE3-DE4*TAMB+DIR*DE2) GO TO 3000
IF(TAMB.LT.TMINH5) GO TO 3000
IMODE=IMODE+2
QAX=CAX1+TAMB*CAX2+(TAMB**2.)*CAX3

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```

COPAX=CAX4+TAMB*CAX5+(TAMB**2.)*CAX6
IF(COPAX.LT.O.1) COPAX=0.1
PAX=QAX/COPAX
TINSP=TINSP+QAX/FL4/CPA
FINSP=FL4
C OFF-PEAK HEATERS
3000 IF(DTSLST.GE.DECOLL) GO TO 5000
      IF(YOFFPK.LT.O.5) GO TO 7000
      IF(TAMB.LT.TAMBOP) GO TO 3500
      IF(TSTG(NCONUP+1).GT.STUCHT) GO TO 7000
      GO TO 3600
3500 IF(TSTG(NCONLW+1).GT.STLCHT) GO TO 7000
3600 IF(TAMB.LT.TMINH6) GO TO 7000
      IMODE=IMODE+4
      QOP=COP1+TAMB*COP2+(TAMB**2.)*COP3
      COPOP=COP4+TAMB*COP5+(TAMB**2.)*COP6
      IF(COPOP.LT.O.1) COPOP=0.1
      POP=QOP/COPOP
      QOPST=QOP*EXP(-UAD6/FL6/CPA)
      TIND=QOPST/FL6/CPA+TOUTST
      QOPSP=QOP-QOPST
      POPST=POP*EXP(-UAD6/FL6/CPA)
      POPSP=POP-POPST
      FLOWD=FL6
      GO TO 7000
C SOLAR TO SPACE
4000 IMODE=10
      QSLSP=FL1*DTSLSP*CPA+PS1
      TINSP=TRM/2.+TOLDRM/2.+QSLSP/FL1/CPA
      FINSP=FL1
      PSLSP=PS1
      GO TO 7000
C SOLAR TO STORAGE
5000 IMODE=IMODE+8
      QSLST=(FL2*DTSLST*CPA+PS2)*EXP(-UAD2/FL2/CPA)
      PSLST=PS2*EXP(-UAD2/FL2/CPA)
      TIND=TOUTST+QSLST/CPA/FL2
      QSLSP=(FL2*DTSLST*CPA+PS2)-QSLST
      PSLSP=PS2-PSLST
      FLOWD=FL2
      GO TO 7000
C STORAGE TO SPACE
6000 IMODE=12
      FLOWU=FL3
      FINSP=FL3
      TINU=TRM/2.+TOLDRM/2.
      PSTSP=PS3
C STORAGE TEMP & HEAT FLOW CALCULATION
7000 IF(TINFS.LE.-50.)TINFS=TOLDRM/2.+TRM/2.
      IF(TINFT.LE.-50.)TINFT=TOLDRM/2.+TRM/2.
      IF(TINFB.LE.-50.)TINFB=TOLDRM/2.+TRM/2.

```

```

IF(FLOWU.GT.1.) FLOWU=AMAX1(FLOWU,FLOWD,0.)
IF(FLOWU.GT.1.) FLOWD=0.
XMDOT=AMAX1(FLOWU,FLOWD,0.)
IF(XMDOT.LE.0.5) GO TO 7025
CV1=(HV*ASTG*XLSTG/FNSTG)*(XMDOT/ASTG/DSTG)**0.7
CV=(CV1*XMDOT*CPA)/(CV1+XMDOT*CPA)
GO TO 7030
7025 CV=0.; CV1=1.
7030 CM=RHOCPR*XLSTG*ASTG/FNSTG/DET
      CK=HC*ASTG*FNSTG/XLSTG
      TA(1)=TIND
      TA(NSTG+2)=TINU
      DO 7050 K50=2,NSTG+1
      DELN=0.
      DEL1=0.
      K60=K50
      IF(FLOWU.GT.1.0) K60=NSTG+3-K50
      IF(K50.EQ.2) DEL1=1.
      IF(K50.EQ.NSTG+1) DELN=1.
      K70=K60-1
      K80=K60+1
      IF(FLOWU.GT.1.) K70=K60+1
      IF(FLOWU.GT.1.) K80=K60-1
      CL=(UAS+DEL1*UAT+DELN*UAB)/2.
      TSTG(K60)=(TOLD(K60)*(CM-CL-CK-CV/2.)+TINFS*
1UAS+TINFB*DELN*UAB+TINFT*DEL1*UAT+CV*TA(K70)+
1CK/2.*(TOLD(K70)+TOLD(K80)))/(CM+CV/2.+CL)
7050 TA(K60)=(TA(K70)/CV1+
1(TOLD(K60)+TSTG(K60)))/(2.*XMDOT*CPA))*CV
      TSTG(1)=TSTG(2)
      TSTG(NSTG+2)=TSTG(NSTG+1)
      QLOSS=0.
      DELTU=0.
      TBARN=0.
      DO 7100 K100=2,NSTG+1
      IF(ABS(TINFS-TRM/2.-TOLDRM/2.).GT.0.01)
1TINFS=TOLD(K100)/2.+TSTG(K100)/2.
      QLOSS=QLOSS+UAS*(TOLD(K100)/2.+TSTG(K100)/2.-
1TINFS)
      TBARN=TBARN+TSTG(K100)/FNSTG
7100 DELTU=DELTU+CM*(TSTG(K100)-TOLD(K100))
      IF(ABS(TINFT-TRM/2.-TOLDRM/2.).GT.0.01)
1TINFT=TOLD(2)/2.+TSTG(2)/2.
      IF(ABS(TINFB-TRM/2.-TOLDRM/2.).GT.0.01)
1TINFB=TOLD(NSTG+1)/2.+TSTG(NSTG+1)/2.
      QLOSS=QLOSS+UAB*(TOLD(NSTG+1)/2.+TSTG(NSTG+1)/2.-
1TINFB)+UAT*(TOLD(2)/2.+TSTG(2)/2.-TINFT)
C LOAD & ROOM TEMP CALCULATIONS
8000 IF(FLOWU.GT.1.) TINSP=TA(2)+PSTSP/FL3/CPA
      IF(QSLST.GT.1.0) QINT=QINT+QSLSP
      UA=UACON+CVW*((ABS(VWIND))**XNVW)+
1CHT*HTOTAL+CTAMB*TAMB

```

```

TRM=(TOLDRM*(CAPRM/DET-FINSP*CPA/2.-UA/2.)+QINT+
1QOPSP+QLOSS+FINSP*CPA*TINSP+UA*TAMB)/(CAPRM/DET+
1FINSP*CPA/2.+UA/2.)
IF(QSLST.GT.1.0) QINT=QINT-QSLSP
QLOAD=(TRM/2.+TOLDRM/2.-TAMB)*UA
QDELIV=FINSP*CPA*(TINSP-TRM/2.-TOLDRM/2.)
IF(ITERAT.LT.3) GO TO 9999
IF(IMODE.EQ.IMODES.AND.ABS(TRM-TRMSAV).LE.ERR)
1GOTO 9001
IF(ITERAT.LT.7) GO TO 9999
IF(IMODE.EQ.KMODES.AND.ABS(TRM-TRMS2).LE.ERR)
1GOTO 9000
9999 CONTINUE
IF(KCOUNT.LE.1) GO TO 9001
KCOUNT=1
TRM=TOLDRM
INON=INON+1
GO TO 49
9000 IOSC=IOSC+1
9001 IF(IMODE.EQ.12)QSTSP=QDELIV
OUT(1)=QSLSP
OUT(2)=QSLST
OUT(3)=QOPST
OUT(4)=QOPSP
OUT(5)=QHP
OUT(6)=QAX
OUT(7)=QSTSP
OUT(8)=QLOSS
OUT(9)=PSLSP
OUT(10)=PSLST
POPCR=COP7*EXP(-UAD6/CPA/FL6)
OUT(11)=POPST+POPCR
OUT(12)=POPSP+COP7-POPCR
OUT(13)=PHP+CHP7
OUT(14)=PAX+CAX7
OUT(15)=PSTSP
OUT(16)=HTOTAL*ACOLL
OUT(17)=TSTG(NSTG+1)
OUT(18)=TSTG(2)
OUT(19)=TRM
OUT(20)=TBARN
DO 9005 KX=1,16
9005 OUT(KX)=OUT(KX)*DET
TMINX=AMIN1(TRM,TMINX)
TMAXX=AMAX1(TRM,TMAXX)
IF(XCHEK.LE.0.) GO TO 9004
NXB=NSTG+1
NXA=NSTG+2
ITIM=IFIX(TIME+0.1)
IF(ITIM.GT.9999) ITIM=ITIM-10000

```

```

WRITE(LUW,8991) (XIN(JJB),JJB=1,10),(OUT(JJC),
1JJC=1,ITIM,IMODE,ITERAT,TINSP,QLOAD,QDELIV,DELTU,
1(TSTG(JJD),JJD=2,NXB),(TA(JJE),JJE=1,NXA)
9004 DO 29 ICA=1,13
29 IF(IMODE+1.EQ.ICA)IMO(ICA)=IMO(ICA)+1
IF(TIME.LT.TFINAL) GO TO 999
WRITE(LUW,8999)TMINX,TMAXX,IOSC,INON
DO 30 IC=1,13
ID=IC-1
30 IF(IMO(IC).GT.0) WRITE(LUW,1002) ID,IMO(IC)
1001 FORMAT(// ' Mode',3X,'Frequency'/
1 ' -----')
999 RETURN
8999 FORMAT(// ' Minimum Space Temperature:',F6.2,
1/ ' Maximum Space Temperature:',F6.2,/
1 ' Oscillating Convergence During',I6,' Steps'
1/ ' No Convergence During',I6,' Steps'//
1/ ' Operational Mode Frequency'
1/ ' -----'
1/ ' Mode Frequency'
1/ ' -----')
8991 FORMAT(2X,10E13.7/2X,10E13.7/2X,10E13.7,
13I10,4E13.7/2X,10E13.7)
1002 FORMAT(I5,12X,I10)
END
SUBROUTINE TYPE29(TIME,XIN,OUT,T,DTDT,PAR,INFO)
DIMENSION XIN(10),OUT(15)
IF(XIN(10).EQ.0.) XIN(10)=1.
DO 25 J=1,7
25 OUT(J)=(XIN(J)*XIN(9)/XIN(10))+XIN(8)
RETURN
END

```

APPENDIX B

SYSTEM MODEL TRNSYS LISTING

SIM 0 3360 1
NOL
CON 50
DET= 0.5
* SPACE HEATING LOAD
UAC= 655.6
CVW= 17.53
CTA= -1.782
CAP= 42000 * DET
* SOLAR COLLECTORS
FTA= 0.478
FU1= 0.572
ARC= 370
* THERMOSTATS
TRM= 68.24
MIN= 80
DEC= 25
STU= 112
STL= 118
DE1= 1.5
DE2= 1.4
TOP= 37.5
DE3= 2.75
* STORAGE
KST= 0.29
CPR= 24.4
DOB= 0.125
NST= 14
NU1= 7
NL1= 9
NL2= 11
* POWER CONSUMPTION RATES
PS1= 2090
PS2= 2090
PS3= 2500
PCR= 196.6
OP1= 60000
AX1= 36620
HP5= 0.02
HP4= 1.607
* HEAT FLOW RATES
HP1= 13627
HP2= 396
* LOSS COEFFICIENTS
UA2= 131.5

```

A6= 76.1
UAS= 22.73
UAT= 5.72
UAB= 2.917
* FLOW RATES
FL1= 2340
FL2= 2450
FL3= 4950
FL4= 4410
FL6= 3610
CPA= 0.245
* OUTPUT OPTIONS
ITI= 1 / DET
AVG= 1 * ITI
YR= 1680 * ITI
MNT= 168 * ITI
XCH= 0
*
* TIME STEP DEPENDENT QUANTITIES
*
UNI 2 TYP 14 OFF-PEAK CONTROL CLOCK I
PAR 32
0 0 3 0 3 0.5 5 0.5 5 1 11 1 11 0.5 12 0.5
12 0 27 0 27 0.5 29 0.5 29 1 33 1 33 0 48 0
*
UNI 5 TYP 14
PAR 20
0 0 3 0 3 1 12 1 12 0 27 0 27 1 33 1 33 0 48 0
*
UNI 3 TYP 14 OFF-PK CONTROL CLOCK II
PAR 10
0 1 2208 1 2208 0 3360 0 3360 1
*
UNI 4 TYP 14 OFF-PK CONTROL CLOCK IV
PAR 10
0 NL1 2208 NL1 2208 NL2 3360 NL2 3360 NL1
*
UNI 23 TYP 14 OFF-PEAK CALCULATION CLOCK I
PAR 20
0 1 14 1 14 0 26 0 26 1 36 1 36 0 44 0 44 1 48 1
*
UNI 21 TYP 14 INTERIOR LOAD SIMULATOR
PAR 22
0 1700 14 1700 14 5080 18 5070 18 3390 34 3390
34 5080 44 5080 44 3390 48 3390 48 1700
*
UNI 10 TYP 14 INT. LD. SIM. 2
PAR 24
0 1.164 336 1.174 672 1.1 1008 .95 1344 1.07 1680 .937
2016 .95 2352 .898 2688 .933 3024 .939 3360 .929 3360 1.164
*

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UNI 22 TYP 14 OFF-PK CALCULATION CLOCK II

PAR 10

0 0 240 0 240 1 336 1 336 0

*

* TIME STEP INDEPENDENT QUANTITIES

*

UNI 20 TYP 9 WEATHER DATA INPUT I

PAR 13

5 ITI 1,1.8,32 2,0.3171,0 3,0.62137,0 32 1
(10X,E10.4,E10.4,E10.4,20X,E10.4//70X,E10.4/)

*

UNI 24 TYP 15 OFF-PK CONTROL CLOCK III

PAR 1

9

INP 2

2,1 3,1

0 1

*

UNI 6 TYP 15

PAR 1

1

INP 2

5,1 24,1

0 0

*

UNI 11 TYP 15 INT. LD. SIM. 3

PAR 1

1

INP 2

21,1 10,1

0 0

*

UNI 12 TYP 15 INT. LD. SIM. 3

PAR 1

1

INP 2

11,1 0,0

0 DET

*

UNI 25 TYP 15 OFF-PK CALCULATION CLOCK III

PAR 1

12

INP 2

22,1 23,1

1 1

*

UNI 1 TYP 39

PAR 83

XCH DET CPA 19.01 FL1 FL2 FL3 FL4 FL6 PS1 PS2 PS3 HP1 HP2

0 HP4 HP5 0 PCR -20 OP1 0 0 1 0 0 0 -100 AX1 0 0

1 0 0 0 -100 TRM DE1 DE2 DE3 0 MIN DEC ARC FTA FU1 0

O UAC CVW 1 O CTA CAP UA2 UA6 6 42.25 DOB UAS -99 UAT -99
 UAB 55 KST 0.7882 CPR NST
 87 85.5 84 81 78 75.5 73 71.5 70 69.5 69 69.5 70 70
 INP 10

20,1 20,2 20,3 6,1 11,1 4,1 0,0 0,0 0,0 0,0
 32 0 3 1 1570 NL2 NU1 STL STU TOP

*

UNI 7 TYP 29

INP 10

1,9 1,10 1,13 1,14 1,15 12,1 0,0 0,0 25,1 0,0
 0 0 0 0 0 0 0 0 0 1

*

UNI 8 TYP 29

INP 10

20,1 1,19 1,20 1,18 1,17 0,0 0,0 0,0 0,0 0,0
 0 0 0 0 0 MNT DET 0 1 MNT

*

UNI 27 TYP 24 INTEGRATOR#1

PAR 1

MNT

INP 10

1,1 1,2 1,3 1,4 1,5 1,6 1,7 1,8 12,1 1,16
 0 0 0 0 0 0 0 0 0 0

*

UNI 28 TYP 24 INTEGRATOR#2

PAR 1

MNT

INP 10

1,9 1,10 1,11 1,12 1,13 1,14 1,15 7,1 7,2 7,3
 0 0 0 0 0 0 0 0 0 0

*

UNI 29 TYP 24 INTEGRATOR#3

PAR 1

MNT

INP 10

7,4 7,5 7,6 8,1 8,2 8,3 8,4 8,5 8,6 8,7
 0 0 0 0 0 0 0 0 0 0

*

UNI 33 TYP 24 INTEGRATOR#4

PAR 1

AVG

INP 4

20,4 1,19 20,5 1,20
 0 0 0 0

*

UNI 34 TYP 25 PRINTER#4

PAR 4

AVG 0 YR 20

INP 4

33,1 33,2 33,3 33,4

A B C D

*

UNI 30 TYP 25 PRINTER#1

PAR 4

MNT MNT YR 31

INP 10

27,1 27,2 27,3 27,4 27,5 27,6 27,7 27,8 27,9 27,10

A B C D E F G H I J

*

UNI 31 TYP 25 PRINTER#2

PAR 4

MNT MNT YR 31

INP 10

28,1 28,2 28,3 28,4 28,5 28,6 28,7 28,8 28,9 28,10

A B C D E F G H I J

*

UNI 32 TYP 25 PRINTER#3

PAR 4

MNT MNT YR 31

INP 10

29,1 29,2 29,3 29,4 29,5 29,6 29,7 29,8 29,9 29,10

A B C D E F G H I J

*

END

VITA

David J. Chaffin was born in Knoxville, Tennessee on May 11, 1958. He graduated from Laurel High School in Knoxville in 1973 and attended Antioch and Allegheny Colleges before entering the University of Tennessee in 1976. He received the Bachelor of Science in Mechanical Engineering in August, 1979 and a Master of Engineering in Mechanical Engineering in December, 1980. He will be employed by the Tennessee Valley Authority following graduation.