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SOLAR HEATING SYSTEMS

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UNIQUE SOLAR SYSTEM COMPONENTS

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INTEGRATED TANK/HEAT EXCHANGER MODELING/EXPERIMENTS

As noted in the last progress report, we have decided to commit to using TRNSYS 14 for the development of a new wrap-around heat exchanger/tank model. Specifically, the tank model developed by B. J. Newton (1995). The first step was to get TRNSYS 14.101 up and running on our DEC 5000 workstation. This effort turned out to be more than a little difficult.

TRNSYS version 14.101 was developed and delivered for the Microsoft FORTRAN PowerStation Compiler. The operating system on our DEC station is ULTRIX V4.2 (Rev. 96) with the DEC FORTRAN for ULTRIX RISC Systems Version 3.2 compiler. Although both compilers are written to the Fortran 77 standard it is nearly impossible to write Fortran code that does not include some nonstandard syntax or code elements. The problems encountered typically involved machine specific path names, the use of machine and compiler specific run-time functions and subroutines, or (in one case) the use of an intrinsic functions which are not part of the Fortran 77 standard.

As a check, I ran each of the example decks provided with TRNSYS 14.101 on both platforms and compared results. As a result of this process, I identified 3 programming errors (BUGS). All of the errors have been reported to Nathan Blair at the University of Wisconsin.

I did identify one potential problem with the TRNSYS TYPE4 tank model that should be of general interest. Comparison of the results from Example 8 between the two platforms exposed a source of numerical error the Fortran code written for the TYPE4 Thermal Storage Tank Model. Item 4 shows by example that the results of the TYPE4 model are sensitive to precision errors, and, that the errors may be significant.

Example 8 models a large solar hot water system for industrial use. A large array (700 m^2) of CPC collectors is couple to a large (37.5 m^3 (~1000 gal.)), well insulated ($U = 0.42 \text{ kJ/hr-m}^2\text{-K}$ (~R-48), thermal storage tank. The tank is divided into three nodes. The system is designed to supply 85 C water for 45 minutes of each hour, 16 hours a day, 5 days a week. An external auxiliary heater insures that the load is met. The simulation is run for a seven day period with a summer data set for Albuquerque, New Mexico. Note, these values come from the simulation deck 'ex8.dck' and not from the manual. There are some discrepancies between the manual write up and the simulation deck.

When I compared the results for Example 8 (supplied with TRNSYS 14.101) from our DECstation with the results from the manual (and from our PC). I found that the agreement was not as good as I had found for the other examples I had run. More importantly, the PC predicted a $1.393 \times 10^6 \text{ kJ}$ loss due to venting of the tank on the seventh (and last) day of the simulation while the DECstation predicted no loss.

My next step was to compare the predicted tank temperatures during the seventh day of the simulation on a time step by time step basis. What I found was quite disturbing. The tank temperature profiles predicted by TRNSYS on the PC behaved as expected. They are no draws

scheduled on the sixth or seventh day (Saturday and Sunday) so the tank continues to heat up until it reaches the specified boiling temperature of 100 C. During each succeeding time step, the tank then vents enough energy to maintain the tank at 100 C.

On the DECstation, something quite different happened. At the time step at which the collector pump controller turns ON on the seventh day, the average temperature (TRNSYS outputs the average tank temperatures, not the final temperatures) of the tank nodes suddenly drops even though the tank receives a net gain of energy from the collectors. The average temperature of the tank dropped over a time step in which the energy gain was positive. The calculation violated the energy balance and in fact violated the first law. In succeeding time steps, the tank temperature slowly increased toward 100 C and then experienced a sudden drop again.

After some detailed study of the Fortran code, I tracked the error to the coding of the analytical solution of the differential equations governing the energy balance on the tank nodes. The error was caused by a lack of numerical precision in this calculation as is illustrated below.

The easiest way to fully explain the problem is to quickly go through the solution of these equations and then give a couple of examples.

The energy balance on the i^{th} constant volume node of the tank is given by Equation (4.3.1.1) of the TRNSYS manual and duplicated here.

$$\begin{aligned}
 M_i C_{pf} \frac{dT_i}{dt} = & \alpha_i \dot{m}_h C_{pf} (T_h - T_i) + \beta_i \dot{m}_L C_{pf} (T_L - T_i) + UA_i (T_{env} - T_i) \\
 & + \gamma_i (T_{i-1} - T_i) C_{pf} \quad \text{if } \gamma_i > 0 \\
 & + \gamma_i (T_i - T_{i+1}) C_{pf} \quad \text{if } \gamma_i < 0 \\
 & + Q_i
 \end{aligned} \tag{1}$$

The variables are as defined as in section 4.3.1 of the manual. The first two terms in Equation (1) account for heat addition due flows directly from the collector and to the load. The third term accounts for the loss of energy to the environment. The combination of the fourth and fifth terms account for the net flow up or down in the tank. The last term accounts for additional heat added to the tank from the auxiliary heaters. When Equation (1) is rearranged as,

$$\begin{aligned}
 \frac{dT_i}{dt} = & \frac{-1}{M_i C_{pf}} \left(\alpha_i \dot{m}_h C_{pf} + \beta_i \dot{m}_L C_{pf} + UA_i + \gamma_i C_{pf} \right) \cdot T_i \\
 & + \frac{1}{M_i C_{pf}} \left(\alpha_i \dot{m}_h C_{pf} T_h + \beta_i \dot{m}_L C_{pf} T_L + UA_i T_{env} + \gamma_i C_{pf} [T_{i-1} (\gamma_i > 0) \text{ or } -T_{i+1} (\gamma_i < 0)] \right) \\
 & + \frac{Q_i}{M_i C_{pf}}
 \end{aligned} \tag{2}$$

it is in the form,

$$\left[\frac{dT_i}{dt} = a_i T_i + b_i \right] + \frac{c_i}{M_i C_p} \quad (3)$$

Which can be integrate analytically as described in section 1.10.1 of the TRNSYS manual to determine the final temperature at the end of a time step Δt .

$$\left[T_{i,\tau} = \left(T_{i,\tau-\Delta t} - \frac{\bar{b}_i}{a_i} \right) \cdot e^{a_i \Delta t} - \frac{\bar{b}_i}{a_i} \right] + \bar{c}_i \cdot u_i \quad \text{where } u_i = \left(e^{a_i \Delta t} - 1 \right) \cdot \frac{1}{a_i \cdot M C_p} \quad (4)$$

The average temperature over the time step is then calculated from

$$\bar{T}_i = \frac{1}{\Delta t} \int_0^{\Delta t} T_i \cdot d\Delta t.$$

$$\left[\bar{T}_{i,\tau} = \left(T_{i,\tau-\Delta t} - \frac{\bar{b}_i}{a_i} \right) \cdot \frac{(e^{a_i \Delta t} - 1)}{a_i \Delta t} - \frac{\bar{b}_i}{a_i} \right] + \bar{c}_i \cdot v_i \quad \text{where } v_i = \left(\frac{(e^{a_i \Delta t} - 1)}{a_i \Delta t} - 1 \right) \cdot \frac{1}{a_i \cdot M_i C_p} \quad (5)$$

The section of code in 'type4.f' which solves these equations is shown below.

```

352 C   FIND TEMPERATURE OF NODE AT END OF TIMESTEP AND AVERAGE NODE
353 C   TEMPERATURE OVER TIMESTEP.
354     KI = INIT + NODE
355     KF = FINAL + NODE
356     KAVG = AVG + NODE
357     AA = -(FL1 + FL2 + UA + UAFI) / MCPN
358     BB = (FL1 * T1 + FL2 * T2 + UA * TENV + UAFI * TFLUE + QB (NODE, 1) + QB (NODE, 2)) / MCPN
359     TI = S (KI)
360     CALL DIFFEQ (TIME, AA, BB, TI, TF, TAVG)
361     S (KF) = TF
362     S (KAVG) = TAVG
363
364
365 C   MAINTAIN U AND V ARRAYS FOR AUXILIARY HEATER ALGORITHM. IF AUX HEAT
366 C   IS ADDED TO NODE, THE FINAL NODE TEMPERATURE IS INCREASED BY
367 C   AUX*U(NODE) AND THE AVERAGE TEMPERATURE IS INCREASED BY AUX*V(NODE)
368     IF (ABS (AA) .GT. 0.) GO TO 84
369     U (NODE) = DELT / MCPN
370     V (NODE) = U (NODE) / 2.
371     GO TO 85
372 84   U (NODE) = (EXP (AA * DELT) - 1.) / AA / MCPN

```

$$V(\text{NODE}) = ((\text{EXP}(\text{AA} \cdot \text{DELT}) - 1.) / \text{AA} / \text{DELT} - 1.) / \text{AA} / \text{MCPN}$$

The solution of Eq. (1) in the form of Eq. (3) is carried out in several steps. First, the final and average temperatures of the node due only to the portion of Eq. (3) that is inside of the square brackets are calculated by the call to the subroutine DIFFEQ (reference section 3.44 of the TRNSYS manual) at line 360. This part of the calculation is performed in DOUBLE PRECISION within DIFFEQ.

The coefficients u_i ($U(\text{NODE})$) and v_i ($V(\text{NODE})$) are then calculated and saved for use in correcting the calculated final and average temperatures for the addition of auxiliary heat as in Eqs. (4) and (5). These coefficients are also used in the mixing routine to correct temperature inversions that may occur in the tank after the first set of the solution. Note that the calculation of these coefficients is performed in SINGLE PRECISION. This is the source of the error in Example 8 on our DECstation.

When the tank is in standby mode (no flows), a_i (AA) = -1.8×10^{-4} and b_i (BB) = 4.52×10^{-3} for the middle node. The time step is 0.25 hr. Then, $\text{EXP}(\text{AA} \cdot \text{DELT})$ is very close to one and the quantity $(\text{EXP}(\text{AA} \cdot \text{DELT}) - 1.)$ retains at most only 2 or 3 significant digits in SINGLE PRECISION. As the calculations of $U(\text{NODE})$ and $V(\text{NODE})$ in lines 372 and 373 continue, $U(\text{NODE})$ retains only 2 or 3 significant digits and $V(\text{NODE})$ may retain none.

In fact the correct (DOUBLE PRECISION) values of U and V should be 4.77×10^{-6} and 2.39×10^{-6} respectively. The calculated values are 4.77×10^{-6} and -3.14×10^{-5} respectively. In this case, the sign of V actually changes! If even a small amount of heat were added to this node due to the mixing calculation or the operation of an internal auxiliary heater, the final temperature would increase, but the average temperature would decrease, possibly below the initial temperature. Also remember, TRNSYS attempts to converge the inputs and outputs of each component which are the average temperatures. So this error may have a direct effect on the convergence of the components during the time step.

So why doesn't this calculation spit blood and die? Mostly because there is no internal auxiliary heater and 45 minute draws each hour, sixteen hours a day, five days a week. As a result, the value of AA is mostly a function of the flows. Also, the tank remains well stratified due to the draws and the mixing routine is not required to correct large temperature inversions.

The next question is this a problem for typical residential size tanks. Consider an 80 gal (0.303 m^3) tank with an R -value of 16.7 (U -value = $1.22 \text{ kJ/hr-m}^2\text{-K}$) divided into 3 nodes with a 4500 W (16,200 kJ/hr) auxiliary heater adding heat to the top node. $\text{AA} = -2.88 \times 10^{-3}$ and $\text{BB} = 7.19 \times 10^{-2}$ for the top node. With a 15 minute time step, $U = 5.91 \times 10^{-4}$ and $V = 2.81 \times 10^{-4}$. The correct value of V should be $V = 2.95 \times 10^{-4}$ and the error in the average temperature at the end of the iteration will be $\sim 0.3 \text{ C}$. If the time step is reduced to 5 minutes, $U = 1.97 \times 10^{-4}$ and $V = 8.48 \times 10^{-6}$. The correct value of V should be $V = 9.85 \times 10^{-5}$ and the error in the average temperature at the end of the iteration will be $\sim 1.5 \text{ C}$.

It not hard to imagine cases where this precision error could significantly affect the outcome of a simulation. For instance during a mid-day draw when the collector pump is running the net flow capacitance $(\dot{m}C_p)_{\text{net}}$ might nearly balance the tank losses making AA very small. This is also

possible nearly anytime in systems with natural convection heat exchangers with low flow rates (or other low flow systems).

The next obvious question is "why didn't the error occur on the PC?" The answer is that the Microsoft FORTRAN PowerStation compiler is a 32-bit compiler. As part of the implementation of the IEEE floating-point arithmetic standard, the compiler "uses the extended double format internally in the math coprocessor (or emulation software)" (reference Microsoft FORTRAN PowerStation User's Guide, Section 15.1.1, page 254). That means that this compiler performs all intermediate calculations in double precision and then rounds the result when it writes it to memory. Most UNIX f77 compilers have a -r8 compiler option which should cause the same affect. This option did not fix the problem on our DECstation.

As a result of these findings, I have modified both the 'type4.f' routines and the 'type67.f' (the wrap-around heat exchanger model built from an older version of 'type4.f') on our DECstation to force the calculations of U(NODE) and V(NODE) to be in double precision. Though, this is not a problem when running on PC's with Microsoft FORTRAN PowerStation, I would strongly recommend fixing this problem before the next release of TRNSYS. You never know where TRNSYS might be installed and this problem will usually cause inaccurate results without aborting a run.

One additional note, this problem will definitely make obtaining a good overall energy balance on the tank more difficult.

References

Newton, B. J., Schmid, M., Mitchell, J. A., and Beckman, W. A., "Storage Tank Models," *Proceedings of the 1995 ASME Solar Energy Conference*, March 19-24, Maui, Hawaii, pp. 1111-1116, 1995.

RATING AND CERTIFICATION OF DOMESTIC WATER HEATING SYSTEMS

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The rough draft of a paper accepted for the 1996 International Solar Energy Conference in San Antonio, Texas has been completed and is included here. Current work involves additional instrumentation and outdoor testing of the system.

A SYSTEMATIC APPROACH TO IMPROVING THERMOSYPHON SDHW MODEL PERFORMANCE

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ABSTRACT

A systematic procedure is used to adjust three parameters in a computer model of a thermosyphoning, horizontal tank solar domestic hot water heating system with a tank-in-tank heat exchanger to better match an indoor solar simulator experimental performance test. Excellent agreement with the experiment is achieved with the collector useful energy gain and the energy delivered during a draw by ensuring that the model gives an accurate prediction of the temporal history of both the system flow rates and temperatures. As a result, simulation accuracy of two additional one-day outdoor tests is greatly improved and confidence in an annual performance prediction is increased.

INTRODUCTION

The simulation of solar domestic hot water (SDHW) system performance is important to enable efficient and reliable energy saving predictions for these systems. The Solar Rating and Certification Corporation (SRCC) has chosen to use TRNSYS (1992) models to rate SDHW systems sold in the United States under its OG300 Protocol (Solar Rating & Certification Corporation, 1995). These ratings are used by utilities, governmental agencies, and consumers to compare different systems. Thus, ensuring that simulated performance matches experimental results is an important and ongoing area of research (Burch 1993).

Model development and calibration becomes especially difficult for thermosyphon SDHW systems. The performance of these systems is dependent on the complex thermal interaction between the collector and the combination of heat exchanger and thermal storage tank which determines the thermosyphon flow rate at various operating conditions. This makes simulation of these systems more complex than constant flow rate pumped systems. Several sources of modeling uncertainties can be identified. First, the flow in the collector loop is typically laminar. Pressure drops due to pipe fittings and developing flow are significant and must be taken into account (Morrison and Ranatunga, 1980). As a result, accurate estimation of friction loss coefficients in thermosyphon systems is difficult and largely uncertain. Second, the heat exchanger is integrated with the storage tank. Natural convection driven flows on both sides of the heat exchanger make the performance of the combined system

extremely difficult to model. These combinations of heat exchanger and tank have not been studied as extensively as in forced flow situations. Finally, vertical temperature stratification in horizontal tanks is not as well understood as it is in vertical tanks and has a major effect on both collector inlet temperature and thermosyphon flow rate (Morrison and Braun, 1985). Tank destratification due to conduction between the fluid layers and in the tank wall is simulated in the current model, but destratification due to convective mixing in the tank, likely more significant than that due to conduction, is not currently modeled.

Bickford and Hittle (1995) recently collected data on a horizontal-tank thermosyphon SDHW system with an annular tank-in-tank heat exchanger using the indoor solar simulator at the Solar Energy Applications Laboratory (SEAL) at Colorado State University. The solar simulator is a unique facility that allows for full-size SDHW systems to be tested under constant irradiation conditions over extended periods of time. The design and operation of the CSU solar simulator is described in detail by Kenny (1993).

Figure 1 is a schematic of the thermosyphon SDHW system they tested. The system was fully instrumented with thermocouples to measure the fluid temperatures. A low-pressure-drop Coriolis-effect mass flow meter was used to measure flow in the collector loop. The collector performance parameters and the tank heat loss coefficient were determined from separate tests and are noted in Figure 1. They performed a variety of four and eight hour constant irradiation tests with and without draws. Figures 2 thru 5 show the results of a four-hour test with a 420 second draw at an average flow rate of 0.21 liters per second at two hours. The draw volume was about one-half of the total tank volume. Draw flow rate was controlled with a pressure control valve and was measured with a turbine in-line flow meter.

Bickford and Hittle simulated the results of this experiment using the solar simulation program TRNSYS (Klein, 1992) with a recently updated thermosyphon system model originally developed by Morrison and Braun (1985). Their model calculates the thermosyphon flow rate by requiring the sum of changes in pressure around the collector heat exchanger loop due to buoyancy forces and frictional losses to be zero. Recent upgrades (Morrison, 1994) allow the simulation of several different kinds of heat exchangers, including an annular tank-in-tank arrangement. The modeling of systems with heat exchangers is an improvement over the TRNSYS version 13.1 thermosyphon system model. The model now also includes conduction between adjacent water layers in the tank and along the tank wall.

Figure 2 shows the experimental and predicted collector inlet and outlet temperatures during the four hour test. The predicted temperature rise across the collector is larger than in the experiment. Figure 3 shows the predicted and actual collector flow rate. The qualitative shapes of these curves have been discussed previously (Bickford and Hittle, 1995), but note that the model predicts significantly lower flow than observed which counteracts the overprediction of the collector temperature rise in a calculation of collector useful energy. Over the four hour experiment, the predicted collector useful energy is 21053 kJ which is about seven percent larger than the experimental value of 19626 kJ. This is a reasonable result except that it arises due to compensating errors in the model.

Figure 4 shows the average tank temperature as a function of time over the four hour test for the model and the experiment. The experimental average temperature is computed as a volume weighted average of the temperatures of four thermocouples placed in a vertical tree near the center of the tank as described by Bickford and Hittle (1995). The TRNSYS average temperature is computed as a weighted average of the tank node temperatures. The simulation predicts a significantly higher average tank temperature before the draw and drops to a lower average tank temperature after the draw. The predicted average tank temperature then catches up to and again surpasses the experimental

temperature.

Figure 5 shows the tank draw temperature as a function of fractional tank volume over the 420 second draw. The model draw temperature starts off very high and drops with a greater slope than the corresponding experimental draw temperature. This indicates, as does Figure 4, that there is a higher degree of vertical stratification in the model tank than in the experimental tank. The actual energy delivered over the mains temperature during the 420 second draw is 7412 kJ. The model predicts an energy delivery of 9503 kJ which is 28 percent larger than the actual value. With no adjustment, the TRNSYS Thermosyphon System model is not effectively simulating this four hour solar-only constant irradiation test with a fixed volume draw.

This work seeks to improve the accuracy of the simulation by systematically identifying and adjusting key parameter inputs and requiring that the model output better match the time evolution of the collector and heat exchanger/tank temperatures and flow rates. This is accomplished using the TRNSYS program to first examine the accuracy of individual component models and then check the results of the simulation as a whole. Finally, the accuracy of the adjusted model is used to predict performance during two additional one-day outdoor tests made at SEAL.

IMPROVING THE SIMULATION

Clearly, the simulation model can be adjusted to match any single experimental outcome, such as the draw energy, by adjusting one parameter such as the tank heat loss (UA) value or the collector heat loss parameter ($F_{\text{r}}U_{\text{L}}$). However, if the model does not match the details of the development of the system temperatures and the thermosyphon flow rate there can be little confidence in the simulation model's ability to predict system performance under widely varying conditions. A better method is called for.

In order to obtain a more satisfying correlation between the simulation and the experiment several parameters may have to be adjusted. However, the thermosyphon model consists of a standard flat plate collector that is coupled to a heat exchanger and a thermosyphon calculation. Since these components are coupled, modification of any one of them will affect the others making multiple parameter adjustment difficult. For example, adjusting the UA value of the heat exchanger will affect the thermosyphon flow and collector outlet temperature. Therefore, as a first step, the thermosyphon flow calculation in the collector loop is disabled and the collector model is forced to exactly follow the measured collector inlet temperature and flow as shown in Figures 2 and 3. The accuracy of the collector and heat exchanger/tank models can then be assessed and, if necessary, adjusted individually. The thermosyphon calculation is then enabled and performance of the entire model is checked and adjusted if necessary. By using this approach, several adjustments can be made to the model in logical succession allowing the experiment to be simulated more accurately.

DECOUPLED MODEL SIMULATION RESULTS

Figure 6 shows the predicted and experimental collector outlet temperatures when the measured experimental flow rate and collector inlet temperature shown in Figures 2 and 3 are used as inputs to the model (this is called the "forced model" hereafter). With two exceptions, the predicted collector outlet temperature prediction is within about one half of one degree of the experimental collector outlet temperature. The discrepancy at the beginning of the test is a result of the capacitance in the system. From Figure 3, the experimental flow rate, being forced into the model, is very low at the start of the test which causes the model to predict a very large collector temperature rise. In reality, the collector takes some time to respond to the step change in irradiation and does not experience a large

temperature rise immediately. The other discrepancy is during the load draw. This can again be explained by the capacitance of the system as the TRNSYS temperature prediction drops rapidly with the change in collector inlet temperature while the experimental temperature responds with a larger time constant. Even with the discrepancies mentioned above, the experimental and predicted collector useful energy outputs differ by less than one percent. The collector model works very well and the previously measured collector gain ($F_{av}(\tau\alpha)$) and loss parameters ($F_{av}U_L$) are appropriate.

After the fluid leaves the collector, it passes through the heat exchanger and then goes back to the collector inlet. Figure 7 shows the collector inlet temperature predicted from the forced model and the experimental temperature throughout the test. The model does much worse in this prediction. The predicted temperature falls immediately below the experimental temperature and by the end of the draw is nearly ten degrees below the experimental value. The temperature remains well below the experimental value for the remainder of the test. Since the temperature entering the heat exchanger is essentially the same in the model and in the experiment, this result implies that more heat is being transferred to the tank water in the model than in the experiment. This suggests a modification of the heat exchanger UA value is required to reduce the heat transferred to the tank water.

Figure 8 shows the average tank temperature for the model and the experiment. As discussed in relation to figure 4 in the previous section, the temperature rises at a greater rate in the model than in the experiment. Since, in this case, the predicted collector energy output is very close to the experimental value, the predicted heat exchanger UA value must be larger than the experimental value (assuming correct thermal capacity in the tank). Again, this suggests a modification of the heat exchanger UA value.

Figure 9 shows the predicted and actual draw temperature over the 420 second draw. The model predicts higher draw temperatures and higher stratification as discussed in relation to Figure 5. The predicted draw energy of 9896 kJ is about thirty percent larger than the experimental value of 7412 kJ. A modification of the destratification calculation in the tank model may also be required.

ADJUSTMENTS USING THE DECOUPLED MODEL

The decoupled thermosyphon model is used as a starting point for adjustment of the original model parameters. The amount of heat transferred through the heat exchanger is first reduced and then the tank stratification problem is considered. With the collector flow rate forced to the experimental value, working with the heat exchanger and then the tank stratification is a logical progression as heat flows first through the heat exchanger and then into the tank water.

Since the predicted inlet temperature profile to the heat exchanger (collector outlet) is very close to the experimental profile, adjusting the UA value of the heat exchanger until the heat exchanger outlet temperature (collector inlet) profile matches the experimental profile will ensure that the model heat exchanger is transferring the correct amount of energy into the tank water. To do this, a previous eight hour warm-up test with no load draw made by Bickford and Hittle (1995) is used. This is to avoid the complications associated with the capacitance in the system of matching the collector inlet temperature profile during the draw. The eight hour test is modeled by forcing the flow rate and collector inlet temperature as with the four hour test and adjusting the heat exchanger UA value until the predicted collector inlet temperature profile matches the experimental profile as closely as possible. A multiplicative factor of 0.7 applied to the heat exchanger UA calculation in the model minimizes the RMS error in the prediction.

Note, an attempt was made to adjust the heat exchanger UA until the experimental and simulation tank average temperature profiles (specifically, slope of the temperature rise) matched. The

new predicted temperature at the heat exchanger outlet was much lower than the experimental value. The experimental value of the average tank temperature is calculated on a volume weighted average of only four temperature measurements. As a result, the uncertainty in the experimental average is large and thus this temperature may not be the best value to use to characterize the heat transfer to the tank.

Having modified the heat exchanger calculations, the tank stratification problem is considered. Tank destratification in the model includes conduction along the tank wall and between adjacent water layers. These two effects are combined into an equivalent conductivity term. There is currently no convective destratification mechanism modeled. To decrease tank stratification, a multiplicative factor is applied to the equivalent conductivity term and adjusted until the 420 second draw temperature profile is matched as closely as possible. A factor of 27 is required to minimize the RMS error in the temperature profile. Two causes have been suggested previously for the large size of this adjustment. First, an assumption is made in the model that the collector fluid rises without mixing to the top of the annular space in the heat exchanger. Measurements show that this is not the case (Bickford and Hittle, 1995) and mixing will decrease tank stratification. Second, there may be a convective mixing process occurring in the potable water inside the tank (Bickford and Hittle, 1995).

FINAL SIMULATION RESULTS

The modifications calculated using the forced model are applied to the non-forced model. The heat exchanger UA value is multiplied by 0.7 and the equivalent conductivity term is multiplied by 27 throughout the simulation. When this is done, collector and draw energy quantities match very well, but the collector flow rate is still underpredicted and the temperature rise is overpredicted (fortuitously resulting in a very accurate integrated collector energy prediction). A final adjustment is made by reducing the total friction factor in the collector loop piping until the flow rate (and thus temperature rise) match. This requires that the friction factor be reduced by approximately 40 percent.

Figure 10 shows the collector inlet and outlet temperatures for the final model and the experiment. The profiles match within two degrees except at the start of the experiment and during the draw. Figure 11 shows the collector flow rate. Again the profiles match very well. Figure 12 shows the predicted and actual draw temperature profiles. The predicted temperatures in the profile have an RMS error of 1.9 degrees C compared to the original RMS error of 5.6 degrees C (Figure 5). The energy results are summarized in Table 1.

	Experiment	Original Model	Adjusted Model	% Error In Original Model	% Error in Adjusted Model
Coll. Energy (kJ)	19626	21053	19828	+ 7	+ 1
Draw Energy (kJ)	7412	9503	7445	+ 28	+ 0.4

Table 1. Comparison of Integrated Energy Simulation Results for Indoor Test

SIMULATION OF OUTDOOR TESTS

In an effort to test the robustness of the model with the adjusted parameters, the thermosyphon system was mounted on the outdoor testbed at SEAL and two one-day tests were conducted. The first test occurred on a very clear day and the second occurred on a partly cloudy day. A 420 second draw at a flow rate of approximately 0.2 liters per second was made sometime after noon and a complete three

volume draw off was made later in the afternoon for each test. Horizontal radiation, ambient temperature, and tank inlet and outlet temperatures were recorded and used to construct TRNSYS input decks to simulate the outdoor experiments. The simulations were run with the adjusted model parameters and the unadjusted original parameters. Figures 13 and 14 show the experimental, unadjusted, and adjusted model results for the draw temperature profiles for both draws of the first test and Figures 15 and 16 show the same information for the second test. In all cases, the adjusted model greatly increases the accuracy of the temperature predictions and the integrated energy results. Table 2 summarizes the integrated energy results for the outdoor tests.

	Experiment	Original Model	Adjusted Model	% Error in Original Model	% Error in Adjusted Model
Test #1					
Short Draw Energy (kJ)	4551	7368	5371	+ 62	+ 18
Long Draw Energy (kJ)	26383	25042	26249	- 5	- 0.5
Test #2					
Short Draw Energy (kJ)	6534	8466	6140	+ 30	- 6
Long Draw Energy (kJ)	10774	8984	11030	- 17	+ 2

Table 2. Comparison of Integrated Energy Results for Outdoor Tests

CONCLUSIONS

The performance of the TRNSYS Thermosyphon System in modeling these particular solar-only warm-up tests with load draws can be significantly improved by making three parameter adjustments to the model. Collector useful energy, tank stratification, and draw energy are all overpredicted by the unadjusted model. Integrated energy quantities and temperature profile predictions improve greatly with a systematic multiple parameter adjustment of the model.

The required reductions of about 30 percent in the heat exchanger UA value and 40 percent in the collector loop friction factor are physically realistic and justifiable based upon the uncertainty in their modeling. However, the required adjustment to the effective conductivity term is too large to be realistic and it is probable that other physical processes are occurring in the experiment that are not included in the model. Mixing of the fluid in the collector loop as it rises in the annulus of the heat exchanger and convective mixing of potable water in the tank may explain the observed reduction in stratification compared to the original model prediction.

In future testing of thermosyphon systems, experimental tests that disable one portion of the system would be beneficial in the modeling process. For example, forcing the closed loop flow rate and the heat exchanger inlet temperature experimentally would provide a data set with only the heat exchanger performance and tank stratification as parameters.

The general importance of this work lies in the logical progression of the procedure that is used. First, the inlet conditions to the collector are forced to examine the performance of the collector model and to adjust the heat exchanger to match experimental temperature data. Then, using a forced model, the tank water stratification is adjusted to match the experimental draw temperature profile. Finally, with no model forcing, the collector friction factor is adjusted to match experimental collector flow rate. By following this logical progression around the system, the model can be adjusted to match time dependent temperature profiles very well and the original indoor experimental integrated energy values to within one percent. Use of the adjusted parameters in two additional outdoor tests confirms that the adjusted model is significantly better at predicting system performance than the original model.

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Figure 1. Thermosyphon System Overview

Figure 2.
Collector Inlet and Outlet
Temperatures vs. Time.
Outlet on Top and
Inlet on Bottom.

Figure 3.
Collector Flow Rate vs. Time.

Figure 4.
Average Tank Temperature
vs. Time.

Figure 5.
Draw Temperature vs. Volume
Fraction Over the 420
Second Draw.

Figure 6.
Collector Outlet Temperature
vs. Time With Forced Collector
Flow Rate and Inlet Temperatur

Figure 7.
Collector Inlet Temperature
vs. Time With Forced Collector
Flow Rate and Inlet Temperatur

Figure 8.
Average Tank Temperature
vs. Time With Forced Collector
Flow Rate and Inlet Temperatur

Figure 9.
Draw Temperature vs. Time
for Forced Model.

Figure 10.
Collector Inlet and Outlet
Temperatures vs. Time
For Final Model.
Outlet on Top and
Inlet on Bottom.

Figure 11.
Collector Flow Rate vs. Time
For Final Model

Figure 12.
Draw Temperature vs. Time
For Final Model.

Draw Temperature vs. Volume Fraction

Testbed Test #1 7/26/95 - Short Draw

Figure 13.
Draw Temperature vs
Time for Short Draw.
First Test.

Draw Temperature vs. Volume Fraction

Testbed Test #1 7/26/95 - Long Draw

Figure 14.
Draw Temperature vs
Time for Long Draw.
First Test.

Draw Temperature vs. Volume Fraction

Testbed Test #2 8/3/95 - Short Draw

Figure 15.
Draw Temperature vs
Time for Short Draw.
Second Test.

Draw Temperature vs. Volume Fraction

Testbed Test #2 8/3/95

Figure 16.
Draw Temperature vs
Time for Long Draw.
Second Test.

ADVANCED RESIDENTIAL SOLAR DOMESTIC HOT WATER (SDHW) SYSTEMS

This report is for August and September 1995. Experimental tests are being conducted on three side by side systems: ASN, NEG and Thermodynamics.

Experimental Work

The AC voltage readings from the solenoid valves controlling the draws were converted to DC voltages using a power supply and relays. The AC readings were causing the computer to shut down erratically. The data acquisition software and data reduction program were rewritten to reflect these changes.

Heat loss tests for the water heater tanks were performed. These tests were for comparison purposes to the heat loss tests done in 1994. Subsequent to the 1994 tests, "heat traps" were added to some pipes going into/out of the hot water tanks (see attached figures) as suggested by Byard Wood. The systems were configured in two different modes for the tests: 1) the flow control valves on the systems were positioned for normal operation and 2) the valves on the systems were closed. Any differences between the results from these two tests is attributed to free convective heat loss through the attached pipes. The heat loss coefficients for the tanks (UA's) are given in the table below. These values were obtained by a linear regression performed on a plot of the logarithm of the temperature difference (tank temperature minus the ambient temperature) versus time, as per the procedure given in ASHRAE 95.

UA (W/C)	Valves Open			Valves Closed		
	1994	1995	% Difference	1994	1995	% Difference
ASN: Primary	2.43	2.93	+21	1.82	2.56	+41
Auxiliary	2.40	2.04	-15	1.79	1.69	-6
NEG	4.3	2.59	-40	2.0	1.8	-10
Thermo: Primary	3.67	5.36	+46	2.97	2.82	-5
Auxiliary	2.21	2.06	-7	1.66	1.72	+4

The UA is observed to increase for the ASN primary and Thermo Dynamics primary tanks. For ASN, this is probably due to degradation of the insulation wrapped around the tank (we are about to upgrade the ASN tank system to a new natural convection system which we are testing off-line, so we have not taken corrective action to replace the insulation). For the Thermo Dynamics system, we have replaced the Micro Motion flow meter used in 1994, which was responsible for 1/3 of the pressure drop in the system, with one with 1/10 of that pressure drop. Therefore, we hypothesize that the UA of the primary tank increased due to increased recirculation through the natural convection heat exchanger loop. One additional heat loss test will be conducted with the new flow meter removed from the system to isolate its effect. All other tanks with heat traps show a decrease in UA, as expected.

The UA values for the valves closed are all reasonably close to one another, except for the ASN primary tank (same explanation as above) and the NEG tank, which has many complex plumbing connections, and therefore many possibilities for heat loss.

We will continue to collect data under various operating conditions as long as we have sunny weather. Once the last set of normal operation data are collected, the Micro Motion flow meter will be removed from the Thermo Dynamics convection loop, and the plumbing connecting the two NEG collectors will be re plumbed to parallel operation (instead of series). We are continuing to analyze the performance of the systems.

Optimization of Evacuated Tube Collectors

The plumbing and instrumentation of the new NEG collector with a specular backplane reflector is in process. This new collector initially will be tested out of door, off-line from the other systems, with a separate data acquisitions system.

Several problems with the collector were discovered while doing the plumbing. Upon initial pressurization of the collector with mains water, several leaks were noticed around the manifold cover. Removing the cover turned out to be difficult since the filler/insulation (spray type) melted onto the cover and pipes at the high temperatures seen in the manifold. This insulation was scraped away and the leaks were repaired.

Two of the collector tubes lost their vacuum (condensation was seen inside the collector glazing). These two tubes were replaced with spare tubes from previous experiments. Due to the way in which the tubes are manufactured, this was a very time consuming process. The collector has been reassembled and the manifold is being insulated.

The instrumentation for testing the system out of doors includes one thermocouple on the inlet and one on the outlet of the collector, a flow meter on the outlet of the collector, and a solenoid valve, operated based on a temperature difference between the collector inlet and outlet. The solenoid will be used to do energy draws on the collector. The weather data collected include horizontal and tilt radiation (two Eppley PSP pyranometers are currently being calibrated) as well as the ambient temperature (a shielded aspirated thermocouple is used).

NEG Without Heat Trap

NEG With Heat Trap

T	Temperature Measurement.	Pressure Relief Valve
P	Power Measurement.	Ball Valve
ΔP	Differential Pressure Measurement.	Check Valve
V-dot	Flow Measurement.	Resistive Heater Element
Pump		Solenoid Valve
Pressure Gauge and Regulator		Visual Flow Meter

FIGURE 1

ASN SYSTEM

Legend	
T	Temperature Measurement.
P	Power Measurement.
ΔP	Differential Pressure Measurement.
$\dot{V}$	Flow Measurement.
	Pump
	Pressure Gauge and Regulator
	Pressure Relief Valve
	Ball Valve
	Check Valve
	Resistive Heater Element
	Solenoid Valve
	Visual Flow Meter

FIGURE 2

(T) Shielded Room Ambient Temperature

Natural Convection HX

(T) Shielded Room Ambient Temperature

THERMO DYNAMICS

Legend	
T	Temperature Measurement.
P	Power Measurement.
ΔP	Differential Pressure Measurement.
V̇	Flow Measurement.
	Pump
	Pressure Gauge and Regulator
	Pressure Relief Valve
	Ball Valve
	Check Valve
	Resistive Heater Element
	Solenoid Valve
	Visual Flow Meter

FIGURE 3

MANAGEMENT AND COORDINATION OF COLORADO STATE/DOE PROGRAM

Coordination of research activities continued on the three technical research tasks under the DOE grant, and accounts were maintained and updated. Financial and technical reports were submitted as required.