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**IMPROVING THERMOSYPHON SOLAR DOMESTIC HOT WATER
SYSTEM MODEL PERFORMANCE**

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THESIS

**IMPROVING THERMOSYPHON SOLAR DOMESTIC HOT WATER
SYSTEM MODEL PERFORMANCE**

Submitted by
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In partial fulfillment of the requirements
for the degree of Master of Science
Colorado State University
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Fall 1996

ABSTRACT OF THESIS

IMPROVING THERMOSYPHON SOLAR DOMESTIC HOT WATER SYSTEM MODEL PERFORMANCE

Data from an indoor solar simulator experimental performance test is used to develop a systematic calibration procedure for a computer model of a thermosyphoning, solar domestic hot water heating system with a tank-in-tank heat exchanger. Calibration is performed using an indoor test with a simulated solar collector to adjust heat transfer in the heat exchanger and heat transfer between adjacent layers of water in the storage tank. An outdoor test is used to calibrate the calculation of the friction drop in the closed collector loop.

Additional indoor data with forced flow in the annulus of the heat exchanger leads to improved heat transfer correlations for the inside and outside regions of the tank-in-tank heat exchanger.

The calibrated simulation model is compared to several additional outdoor tests both with and without auxiliary heating. Integrated draw energies are predicted with greater accuracy and draw temperature profiles match experimental results to a better degree. Auxiliary energy input predictions improve significantly.

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NOMENCLATURE

A	Area
C_p	Specific Heat (kJ/kg·K)
D	Diameter (m)
d	Heat Transfer Depth (m)
F	Friction Head (Pa)
$F_{av}(\tau\alpha)$	Collector Energy Absorption Parameter
$F_{av}U_L, F_RU_L$	Collector Energy Loss Parameter (W/m ² ·K)
f	Darcy Friction Factor
G_T	Radiation (W/m ²)
Gr	Grashoff Number
Gz	Graetz Number
g	Acceleration Due to Gravity (m/s ²)
H	Thermal Head (Pa)
h	Heat Transfer Coefficient (W/m ² ·K)
i	Loop Counter
k	Conductivity (W/m·K)
L	Length (m)
$\dot{m}$	Mass Flow Rate (kg/s)
N	Node Number (Collector or Tank)
Nu	Nusselt Number
Pr	Prandtl Number
$\dot{Q}, \dot{q}$	Energy Transfer Rate (J/s)
R	Thermal Resistance (K/W)
Ra	Rayleigh Number
Re	Reynolds Number
SG	Specific Gravity
T	Temperature
UA	Heat Transfer Coefficient-Area Product (W/K) (no subscript refers to tank)
V	Velocity (m/s)

Greek

α	Collector Tilt Angle (degrees)
β	Coefficient of Volumetric Expansion
Δ	Change In (Time, Temperature, Location)
ϵ	Heat Exchanger Effectiveness
μ	Dynamic Viscosity (kg/m-s or N-s/m)
ν	Kinematic Viscosity (m ² /s)
ρ	Density (kg/m ³)

Subscripts

a	Ambient

av	Average
c	Collector
D	Hydraulic Diameter
elem	Tank Element
g	Glycol
HX	Heat Exchanger
h	Header
i, in	In
o, out	Out
r	Riser
w	Water

Abbreviations

GPM	Gallons Per Minute
HT	Heat Transfer
RMS	Root Mean Square
SDHW	Solar Domestic Hot Water
SEAL	Solar Energy Applications Laboratory

CHAPTER ONE

INTRODUCTION

This thesis summarizes the experimental and computer modeling results of tests performed on a horizontal tank thermosyphon solar domestic hot water heating system with a tank-in-tank heat exchanger. The objective is to improve the accuracy of numerical simulation models of the system's performance so that these models may be used to predict performance accurately in different situations. This chapter introduces thermosyphon solar domestic hot water heating systems, discusses factors affecting their performance, briefly describes earlier modeling attempts, and provides an overview of this project.

1.1 Solar Thermosyphon Hot Water Heaters

Thermosyphon solar domestic hot water (SDHW) heaters are the most widely used of all solar energy conversion devices due to the fact that they are cost competitive with other energy sources for heating water in many regions of the world (Kudish et al., 1985; Sokolov and Arbel, 1992). Thermosyphon SDHW systems operate using buoyancy forces generated by temperature-induced density changes in the system fluid. As most fluids heat up, their density decreases. This causes a positive buoyancy force relative to the fluid around them. If the fluid is in a solar collector, the heated fluid will tend to rise to the top of the collector. By attaching a storage tank above the level of the collector, the hot

fluid will exit the collector and enter the storage tank. Cooler fluid from the bottom of the storage tank can then be used to replace the hot fluid leaving the collector. The most obvious advantages of these types of systems are the lack of controllers and pumps and their associated expenses and complications. Figure 1.1 illustrates a typical thermosyphon SDHW system with a tilted collector and vertical tank. Hot collector fluid enters the tank somewhere near the top and fluid from the bottom of the tank returns to the bottom of the collector. Hot water from the top of the storage tank is drawn off and used for domestic purposes and cold mains water is fed into the bottom of the tank.

Figure 1.1 Thermosyphon Hot Water System

In cooler climates, where water in a collector could potentially freeze, a heat exchanger may be added between the collector and the tank. The collector fluid loop is

thus isolated from the potable water supply and an anti-freeze solution may be used in the closed loop to prevent freezing. This is illustrated in Figure 1.2. The trade-off for increased freeze protection is typically decreased performance due to higher average collector temperatures caused by extra resistance to heat transfer in the heat exchanger.

In most cases, an auxiliary heating element is used when solar gains are not sufficient to meet the hot water demand. This can be done by inserting a heating element into the solar storage tank as is illustrated in Figure 1.1 (one could also be inserted into the system in Figure 1.2) or by using a separate auxiliary tank plumbed in series after the solar tank. The one-tank system is simpler, but the tank volume above the auxiliary element is lost as solar energy storage since the element keeps it at a constant high temperature. The two-tank system provides more solar storage, but adds expense to the system.

Figure 1.2 Thermosyphon Hot Water System With Heat Exchanger

The system used in this study consists of a standard flat-plate solar collector and a horizontal tank that rests entirely above the collector. The tank includes an annular heat exchanger so that the system may be freeze-protected with a special solution in the closed collector loop. Details of the system are presented in Chapter Two.

1.2 Factors Affecting Performance of Thermosyphon SDHW Systems

A significant amount of research has been reported on the performance of thermosyphon SDHW systems. Since thermosyphon circulation depends on the temperature distribution in the system, the performance of a thermosyphon system is strongly dependent on system configuration and hot water usage. The primary factors affecting system performance are discussed in this section.

1.2.1 Thermosyphon Collector Loop and Storage Tank

Buoyancy-driven flow in the collector loop of thermosyphon systems has been shown to closely follow the solar irradiation through the day (Abd-Al Zahra and Joudi, 1984; Morrison and Ranatunga, 1980; Kudish et al., 1985). Kudish et al. also attempted to correlate flow rate with temperature difference across the collector. They found that the flow rate and temperature difference stabilized to nearly constant values during midday, but fluctuated considerably in the morning and afternoon hours. Huang (1989) and Wenxian and Enrong (1992) both found that thermosyphon flow rate could be increased by increasing the height difference between the collector and the tank or by reducing the loop friction, but found that neither of these increased the energy collecting efficiency significantly. Wenxian and Enrong (1992) found that the position that hot fluid from the

collector enters the tank has a large effect on system performance. Fluid entering closer to the top promotes water stratification and improves performance.

Flow rates in thermosyphon systems are typically an order of magnitude lower than in traditional pumped systems. Low collector flow rates promote increased tank stratification which improves thermosyphon performance by allowing a reduced collector inlet temperature and providing higher temperature water at the top of the tank for use. Morrison and Tran (1984) found that maximum solar contribution from a thermosyphon system occurred when the collector flow corresponded to approximately one tank volume of fluid circulating through the collector every day. The advantages of a low flow rate in the collector loop have also been acknowledged for active pumped SDHW systems. Fanney and Klein (1988) and Hollands and Lightstone (1989) both recommend that flow rates be reduced to about twenty percent of their typical values in pumped systems which corresponds to the result obtained by Morrison and Tran (1984).

Huang (1989) also found that system yearly efficiency had a strong dependence on the collector loss parameter ($F_R U_L$) and recommended the use of high quality collectors.

The orientation of the storage tank, and thus the temperature distribution throughout the system, also has a strong effect on the performance of these systems. Past studies have dealt mostly with vertical tank systems which maintain stratification well. Morrison and Braun (1985) found that horizontal tank systems did not maintain vertical stratification as well due to the short vertical conduction path for heat in the tank. In single-tank, auxiliary boosted systems operating under nearly ideal conditions, they found that horizontal tank systems had solar fractions about seven percent lower than equivalent

vertical tank systems. They suggest possible use of a separate auxiliary tank to allow maximum stratification in the horizontal solar preheat tank. Young and Baughn (1981) found that vertical stratification in horizontal systems could be maintained to a larger degree when a supply water diffuser was used.

1.2.2 Heat Exchanger

There is a much smaller body of research pertaining to thermosyphon SDHW systems with heat exchangers as compared to those without due to the fact that the majority of such systems are installed in climates where freeze-protection is not necessary. Mertol et al. (1981) developed an analytical model of a thermosyphon system with vertical heat exchanger tubes inside a vertical storage tank that predicted performance within ten percent of the same thermosyphon system without a heat exchanger. However, no experimental studies were performed. Rakopoulos and Vazeos (1986) collected data on a system with an internal horizontal tube heat exchanger and describe a model that predicts performance fairly well, but they do not compare performance to a similar system without a heat exchanger. Webster et al. (1987) derived a "heat exchanger penalty factor" for a system with a horizontal tank and a multi-horizontal-tube heat exchanger and found significant performance degradations, likely due to reduced stratification in the horizontal tank due to the type of heat exchanger used in the study. They concluded that a relatively large heat transfer area may be required. Norton et al. (1992) compared the degree of tank stratification in vertical systems with and without an annular heat exchanger using a computer model. Their model predicted increased tank stratification in the system with a heat exchanger except at large load flow rates.

1.2.3 Hot Water Usage Patterns

The pattern of hot water use has been shown to have a large effect on thermosyphon system performance due to the fact that the buoyancy-driven collector flow rate depends on the temperature distribution throughout the system. Braun and Fanne (1983) found that a vertical-tank thermosyphon system collected twenty percent more energy in the morning hours than in the afternoon hours due to the reduced thermosyphon driving forces as the average temperature of the tank increased. They found that energy collection could be improved about fourteen percent by applying a load profile that kept the tank filled with close to mains temperature water. They recommend that systems be designed so that the average daily collector flow approximately matches the load flow.

Other studies have had similar results. Morrison and Sapsford (1983) found that the solar contribution to the annual hot water load was improved about eight percent when the principle draw-off time was in the morning for an off-peak auxiliary boosted system. Zahra and Joudi (1984) found that overall system efficiency increased from three and four percent for intermittent and continuous loading, respectively, compared to the no load case. Morrison and Braun (1985) found that thermosyphon systems were more sensitive to load profile than fully mixed tank pumped systems. They also found maximum performance when the daily collector flow volume was equal to the daily load volume. Wenxian and Enrong (1985) found a significant dependence of solar fraction on load draw profile for four equal volume draws and Michaelides (1992) found increased system efficiency with increasing load.

1.3 Modeling of Thermosyphon SDHW Systems

Model development and calibration is difficult for thermosyphon SDHW systems, especially those with heat exchangers. As discussed above, 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 computer simulation of these systems more complicated than constant flow rate pumped systems.

Several sources of modeling uncertainty may 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). Minor loss coefficients for fittings are not known as accurately in very low Reynolds number regimes. 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 in the system under investigation. Natural convection driven flows on both sides of the heat exchanger make the performance of the combined system extremely difficult to model. Experimental data and heat transfer correlations for this type of heat exchanger are scarce. 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).

The earliest relatively complex model of a thermosyphon system was put forth by Close (1962). His model assumed linear temperature distributions in the tank and collector and assumed that the mean collector and tank temperature were the same. Several

assumptions were also made to predict the thermosyphon flow. He predicted tank mean temperature fairly well on clear days with no draws. Many of the studies since Close's have built on his model.

Morrison and Ranatunga (1980) added developing flow correlations to the collector flow and achieved significantly better thermosyphon flow rate predictions. Mertol et al. (1981) developed a detailed set of one-dimensional equations for a thermosyphon system with a heat exchanger. However, they assumed a constant overall heat transfer coefficient for the exchanger and a linearly stratified tank. Also, they did not experimentally verify the model.

A thermosyphon subsystem without a heat exchanger was first included in version 12 of TRNSYS (Klein et al., 1994), a numerical simulation software package widely used to model solar systems. Braun and Fanney (1983) describe the model. It is similar to Close's model, but uses collector parameters obtained from standardized tests (Hottel-Whiller-Bliss model) and experimental pressure drop versus flow data for the collector. It divides the tank and collector into equally sized uniform temperature nodes. Fairly good results are obtained with this model. Huang and Hsieh (1985) used Close's tank model, the Hottel-Whiller-Bliss collector model, and a semi-empirical loop resistance relation and found good agreement with experimental data.

Hobson and Norton (1988) developed a detailed two-dimensional finite difference model of a vertical tank thermosyphon system. They also used semi-empirical loop resistance calculations and found good agreement with experimental data. The complexity of their model limits the practicality of long-term simulations. Norton et al. (1992) later

developed another model to simulate a vertical tank with an annular heat exchanger used in thermosyphon flow, but did not compare the model with experimental results.

The basis for the current thermosyphon system model in TRNSYS (Version 14) is developed in a series of papers by Morrison et al. (1984, 1985, 1994). It includes such things as developing flow in pipes, convective mixing in the storage tank due to collector and load flows, auxiliary heating, and several types of heat exchangers. This model is described in detail in Chapter Three.

1.4 Project Overview

The Solar Rating and Certification Corporation (SRCC) has chosen to use the TRNSYS (Klein et al., 1994) simulation software to rate solar domestic hot water systems sold in the United States under its OG300 Protocol (SRCC, 1995). These ratings are used by utilities, governmental agencies, and consumers to evaluate the thermal and economic value of different systems. Thus, ensuring that simulated performance matches experimental results is an important and ongoing area of research (Burch et al. 1993).

As discussed above, model development and calibration becomes especially difficult for thermosyphon SDHW systems with heat exchangers. The accuracy of past ratings of such systems is suspect as the models often do not match the time evolution of the collector and heat exchanger/tank temperature and flow rate profiles.

This work seeks to improve the accuracy of a numerical simulation of a horizontal tank thermosyphon solar domestic hot water heating system with a tank-in-tank heat exchanger by developing a model calibration procedure for such systems. The calibration procedure differs from past procedures in that it is accomplished by systematically

identifying key parameter inputs and requiring that the model better match the time evolution of system temperatures and flow rates. The calibration procedure is carried out on the system under investigation, and the calibrated model, including improved heat transfer correlations, is compared with additional data sets.

CHAPTER TWO

SYSTEM DESCRIPTION AND EXPERIMENTAL FACILITY

This chapter describes the thermosyphon solar domestic hot water system under investigation and the indoor and outdoor experimental facilities.

2.1 System Description

The system under investigation is a horizontal tank solar domestic hot water system with a tank-in-tank heat exchanger and a standard flat plate collector. An overview of the system is shown in Figure 2.1.

Figure 2.1 Thermosyphon System Overview

2.1.1 Collector

The flat plate collector consists of parallel vertical risers created by joining two stamped steel plates that are connected to headers at the top and bottom. The absorber surface is coated with a selective Chromonyx coating. Table 2.1 lists several pertinent collector parameters. Other parameters are provided in the sample TRNSYS input decks in Appendix D.

Collector Area, A_c	1.86 m ²
Number of Risers	36
Riser Hydraulic Diameter	4.25 mm
Energy Absorption Parameter, $F_{av}(\tau\alpha)$	0.85
Energy Loss Parameter, $F_{av}U_L$	6.50 W/m ² °C

Table 2.1 Collector Parameters

The collector energy absorption, $F_{av}(\tau\alpha)$, and energy loss, $F_{av}U_L$, parameters were previously determined experimentally using a straight line fit to the efficiency relation

$$\eta = \frac{\dot{m}C_p(T_{c,o} - T_{c,i})}{A_c G_T} = F_{av}(\tau\alpha) - F_{av}U_L \frac{(T_{c,av} - T_a)}{G_T} \quad (2.1)$$

where the temperatures, flow rate, and incident radiation are measured experimentally.

The intercept of this straight line equation is the energy absorption parameter $F_{av}(\tau\alpha)$ and

the slope is the negative of the energy loss parameter $F_{av}U_L$. Bickford (1994) describes the determination of these parameters.

2.1.2 Tank/Heat Exchanger

The tank/heat exchanger unit consists of two horizontal steel containers, one inside the other so that an average annular space of five millimeters exists between the tanks. Figure 2.2 shows the tank/heat exchanger with approximate locations of potable water and heat exchanger connections. Thermocouple locations for the indoor experimental facility are also shown. A mixture of twenty-five percent propylene glycol by volume with water is used in the closed heat exchanger annulus/collector loop. The total closed circuit volume is 12.7 liters. The annulus of the heat exchanger holds about nine liters and the remainder is distributed between the collector and piping. Hot fluid from the collector enters the annulus at the bottom on the left side of the tank and returns to the collector at the same level on the right side. The outside of the shell is insulated with polyurethane insulation.

The inner tank holds 177.9 liters of potable water and is coated with an enamel to prevent corrosion. Cold water is supplied to the bottom of the tank through a small diffuser and hot water is drawn off with a curved pipe reaching near the top of the tank. The potable water inlet and outlet are on the left side of the tank. The auxiliary element is inserted into the middle of the left-side of the tank as seen in Figure 2.2. The overall heat loss coefficient for the tank, determined previously by Bickford (1994), is 3.04 W/K. Additional heat exchanger and tank parameters are provided in the sample TRNSYS input decks in Appendix D.

Figure 2.2 Heat Exchanger/Tank

2.2 Indoor Experimental Facility

2.2.1 System Instrumentation

The indoor experimental facility is designed to test the tank/heat exchanger system independent of the collector. The facility consists of a water conditioning system, a boiler apparatus, and the tank/heat exchanger portion of the system under investigation. The water conditioning system consists of two nominally 300 liter water tanks instrumented with special-limit copper-constantan Type-T thermocouples (± 0.5 °C), a visual flow meter, and appropriate valves and plumbing. The system is configured and controlled such that nearly 600 liters of water may be preconditioned to a selected constant temperature before experimental use. A solenoid-controlled drain valve permits computer-controlled load draws downstream of the installed tank/heat exchanger apparatus. Load flow rate is measured with an EG&G Technologies turbine flow meter (± 0.01 l/s). Appendix A describes the calibration of the meter.

The boiler apparatus, shown on the right in Figure 2.3, is used to simulate the solar collector during indoor testing. The system contains two individually-controlled 4500 Watt electric heating elements, a pump, a Coriolis-effect mass flow meter, and associated valves and plumbing. The Coriolis-effect meter consists of an electronically vibrated U-tube with no internal obstructions. Fluid passing through the tube induces a twist that is directly proportional to the mass flow rate through the tube. The meter electronics convert the signal to a 4-20 mA output that is readable by standard data acquisition systems. The meter is calibrated at the factory and meter uncertainty is taken to be ± 1 kg/hr. Temperatures into and out of the boiler are recorded with 2252-Ohm thermistor probes with a rated accuracy of ± 0.1 °C.

The tank/heat exchanger is plumbed to both the water conditioning system and the boiler apparatus as shown in Figure 2.3. Flow directions are as shown in Figure 2.2.

Figure 2.3 Indoor Experimental Arrangement

Temperature instrumentation consists of four 2252-Ohm thermistor probes used to measure temperatures into and out of the annular heat exchanger and the potable water tank. The probes are inserted pointing into the flow using brass elbow fittings. In addition, there are four special-limit copper-constantan Type-T thermocouples situated in a vertical tree at the center of the tank and twelve thermocouples attached to the outside of the annulus through the insulation. The exterior thermocouples give an indication of fluid temperatures inside the jacket at four levels in the tank. If there is no flow in the heat exchanger, they also give an indication of the temperatures inside the tank. The thermocouple tree and exterior thermocouples are described more fully by Bickford (1994). The voltage across the auxiliary heating element is recorded when it is enabled. This provides a record of the on time for the heater. During one test, an AEMC Clamp-On RMS Power Transducer Model 700 was used to measure the auxiliary power level during heating. The average power level was 1717 ± 154 Watts.

2.2.2 Data Acquisition and Control

Indoor data acquisition and control are accomplished with a Hewlett Packard (HP) System 10 using an E1326A Multimeter and 3 HP E1345-66201 Relay Multiplexer cards and HPE1347A Thermocouple Multiplexer cards. An HP1330A Quad 8-Bit Digital I/O card and an HP1328A 4 Channel D/A Converter are used for control outputs. An IBM compatible 286 computer with Labtech Notebook software is interfaced to the Hewlett Packard hardware to control the system hardware and record data.

The system allows for precise control of the boiler heating rate, on/off control of the water conditioning system heating elements, solenoid control of load draws, and on/off

control of circulating pumps. In addition, the software has a built-in PID controller that can be used in conjunction with the boiler control to perform constant temperature input tests with the heat exchanger. Sampling rates vary from 5 to 15 seconds depending on the specific test being performed. Data are block-averaged to larger time intervals when used as inputs for the TRNSYS program.

2.3 Outdoor Experimental Facility

2.3.1 System Instrumentation

Figure 2.4 show the system as it is installed on the Colorado State University outdoor experimental test bed. Instrumentation includes four special-limit copper-constantan Type-T thermocouples to measure collector and tank inlet and outlet temperatures, an EG&G Technologies turbine flow meter to measure load flow rate, one copper-constantan Type-T thermocouple housed in a ventilated housing to measure ambient temperature, and an Eppley PSP Pyranometer to measure total horizontal radiation ($\pm 1\%$). Collector thermocouples are inserted into brass wyes installed into the collector piping and are pointed to face into the flow near the collector. Tank thermocouples are also installed in appropriate fittings and point into the flow.

2.3.2 Data Acquisition and Control

Outdoor data acquisition is accomplished using an IBM compatible 386 computer with a Keithley Metrabyte MDB-64 driver card and a DAS-1602 8-bit A/D card installed inside the computer. The MDB-64 card controls an external 12 bit Mtherm20 thermocouple card that is used to measure temperatures and the pyranometer output. The

load flow rate is measured with the DAS-1602 card and the turbine flow meter. Load draws are performed manually using a valve placed outdoors near the system. A visual flowmeter allows accurate selection of flow rate while opening the valve.

Figure 2.4 Outdoor Experimental Arrangement

CHAPTER THREE

NUMERICAL SIMULATION MODEL

This chapter briefly describes TRNSYS, the numerical simulation software package used to model the thermosyphon hot water system. A modified version of the standard TRNSYS thermosyphon subsystem that is used for the present study is described in some detail.

3.1 TRNSYS Version 14.1

TRNSYS (Klein et al., 1994), short for Transient System Simulation, is a modular system simulation program distributed by the Solar Energy Laboratory at the University of Wisconsin in Madison. A system is defined as a set of separate components that are interconnected to accomplish a specific task. Each component is modeled in a separate FORTRAN subroutine and given a specific Type number (for example, Type 1 is a solar collector). This facilitates user modification of and addition to the provided components. TRNSYS includes facilities to interconnect components in any desired manner, solve differential equations, and output information.

TRNSYS Version 14.1 includes a Type 45 Thermosyphon Collector-Storage Subsystem that includes a flat-plate collector, a stratified vertical or horizontal storage tank, a check valve to prevent reverse flow, and water as the working fluid. No provisions are made in the subroutine for a heat exchanger in the system.

3.2 Modified Type 45 Thermosyphon Subsystem

A thermosyphon subsystem and tank model routine written by Graham Morrison at the University of New South Wales, Australia (Morrison, 1994) is used in this study. This code was written to expand the capabilities of the Type 45 Subroutine supplied with TRNSYS. The code was written for TRNSYS Version 13.1 and minor modifications have been made so that it can be used with TRNSYS Version 14.1. (The code is actually supplied as two subroutines, a Type 45 Thermosyphon System and a Type 38 Stratified Storage Tank, but they are used together in this study.)

The subroutine allows for seven different tank/heat exchanger modes, but only the horizontal tank-in-tank heat exchanger mode will be discussed here. There are four main simulation tasks in the model: prediction of the collector performance, calculation of the thermosyphon flow rate, prediction of the heat exchanger performance, and handling of tank stratification and load flows. These will be discussed in the following four sections.

3.2.1 Collector Model

The solar collector useful energy output is obtained from the Hottel-Whillier-Bliss relation

$$\dot{Q}_u = \dot{m}C_p(T_{c,o} - T_{c,i}) = A_c G_T F_{av}(\tau\alpha) - A_c F_{av} U_L (T_{c,av} - T_a) \quad (3.1)$$

where the collector energy absorption ($F_{av}(\tau\alpha)$) and energy loss ($F_{av}U_L$) parameters used in the simulation were determined experimentally as described in Chapter Two.

In the model, the collector is divided into N_c (typically ten) equally sized uniform temperature nodes. An energy balance on each of the nodes combined with the Hottel-Whillier-Bliss relation yields the temperature of any node K

$$T_K = T_a + \frac{G_T F_{av}(\tau\alpha)}{F_R U_L} + \left(T_i - T_a - \frac{G_T F_{av}(\tau\alpha)}{F_R U_L} \right) \exp\left(-\frac{F U_L A_c}{\dot{m} C_p} \frac{(K - 1/2)}{N_c} \right) \quad (3.2)$$

where the overall collector loss coefficient-collector efficiency factor is given by

$$F U_L = \frac{-\dot{m}_{test} C_p}{A_c} \ln\left(1 - \frac{F_{av} U_L A_c}{\dot{m}_{test} C_p} \right). \quad (3.3)$$

A quadratic efficiency mode and several modes for evacuated tube collectors are also provided. Equation 3.2 is used iteratively in the model to calculate the collector outlet temperature. It is also used to calculate the thermal head in the collector as discussed in the following section.

3.2.2 Thermosyphon Flow Rate Prediction

The thermosyphon flow rate prediction through the collector-tank loop is similar to that reported by Close (1962). The system is divided into N segments normal to the direction of flow. For stability, the sum of the pressure drop changes around the loop due to buoyancy forces and frictional losses must be zero

$$\sum_{i=1}^N \rho_i F_i + \sum_{i=1}^N \rho_i H_i = 0. \quad (3.4)$$

where F_i denotes the head losses (frictional and/or fittings) for the current element i and H_i denotes the thermal head in the current element.

Total pressure drop in the loop is calculated based on the collector flow rate from the previous time step. If the total is not zero, a new flow rate is determined based upon the previous flow rate and total pressure drop. Successive iterations occur until an acceptable solution is obtained. Calculation of the buoyancy force and the head loss are described below for each section of the system.

3.2.2.1 Calculation of Buoyancy (Thermal) Pressure Head

Calculation of the thermal head (in Pascals) in the collector risers is made by summing the buoyancy forces calculated for each collector node

$$H_c = \sum_{k=1}^{N_c} SG(T_k) \frac{L_r \sin(\alpha)}{N_c} \rho_g g \quad (3.5)$$

where L_r is the vertical height of the collector risers, α is the collector tilt angle, the specific gravity (SG) of the glycol solution is a quadratic function of temperature, the solution density (ρ_g) is taken as constant, and the node temperature (T_k) is calculated using eqn. 3.2. There is no thermal head in the horizontal collector headers.

The thermal head in the connecting pipes is given by

$$H_p = SG(T_{av}) H \rho_g g \quad (3.6)$$

where H is the vertical height of the pipe (inlet or outlet) and the average temperature, T_{av} , is given by

$$T_{av} = T_a + \frac{(T_i - T_a)\dot{m}C_{p,g}}{UA} \left[1 - \exp\left(\frac{-UA_{pipe}}{\dot{m}C_{p,g}}\right) \right] \quad (3.7)$$

which is the average result for mean fluid temperature in a pipe of given length with constant surface temperature. The overall heat transfer coefficient for the pipe, UA_{pipe} , is an input parameter to the model based on pipe and insulation properties. Note that the pipes are modeled as a single node with negligible thermal capacitance.

The thermal head in the tank is zero because the collector inlet and outlet are at the same level on the tank.

3.2.2.2 Calculation of Friction Drop

The pressure drop due to friction and velocity head in the collector headers is calculated by

$$F_h = \frac{1}{2} \left(32S_1 \frac{L_h}{D_h^2} \mu_g V + S_2 \rho_g SG(T) \frac{V^2}{2} \right) \quad (3.8)$$

where S_1 and S_2 are constants that depend on the number of collector risers, μ_g is the viscosity, ρ_g is the density, V is the velocity, and L_h and D_h are the length and diameter of the header.

The pressure drop due to friction in the risers is calculated by

$$F_r = 32 \frac{L_r}{D_r^2} \mu_g V (DEV) \quad (3.9)$$

where the increase in pressure drop due to developing flow, DEV , by Langhaar as reported by Morrison and Ranatunga (1980) is given by

$$DEV = 1 + \frac{0.038}{(L_r / D_r / Re_r)^{0.964}} \quad (3.10)$$

where Re_r is the Reynolds number in the risers.

The friction pressure drop in the annulus of the heat exchanger is calculated using a laminar friction factor for a concentric annulus

$$F_{HX} = 12 \frac{L_{HX}}{gap^2} \mu_g V \quad (3.11)$$

where L_{HX} is the length of the heat exchanger and gap is defined as the distance between the inner and outer tank walls.

Laminar losses are also added for contractions, bends, and wyes using minor loss factors (k) and equivalent lengths (L_e) so that the friction loss for fittings and pipes is

$$F_{pipes, fittings} = \left(f \frac{L_e}{D} + k \right) \frac{1}{2} \rho_g V^2. \quad (3.12)$$

where f is the Darcy friction factor and D is the diameter of a given element.

3.2.3 Heat Exchanger Model

In this system, the heat exchanger is incorporated with the tank as a shell surrounding the potable water tank. Heat is transferred from the glycol solution to the potable water. To begin, an understanding of the tank geometry used in the model is

Figure 3.1 End View of Tank Geometry and Heat Transfer Area (Shaded)

useful. Figure 3.1 shows an end view of the tank and several associated quantities. The same geometry is used for the outer and inner tanks with appropriate dimensions. The tank is divided into horizontal uniform temperature nodes that are distinguished by the angle θ from the top of the tank. For example, the bottom of a node with an angle of 90 degrees would be at the centerline of the tank and the bottom node of the tank corresponds to an angle of 180 degrees. The model has a separate subroutine to keep track of the positions (angles) and relative sizes of the nodes.

To begin the heat exchanger calculation, hot side (glycol) and cold side (water) heat exchanger temperatures are required. The hot side temperature is assumed to be the temperature supplied from the outlet pipe of the collector, calculated assuming a constant exterior pipe temperature as

$$T_c = T_a + (T_i - T_a) \exp\left(\frac{-UA_{pipe}}{\dot{m}C_{p,g}}\right). \quad (3.13)$$

Heat transfer is assumed to take place over the surface of the inner tank where the water temperature is less than the hot side temperature calculated in eqn. 3.13. The cold side temperature is then calculated as a simple volume weighted average of the nodes with a temperature below the hot side temperature as

$$T_w = \frac{\sum_{T < T_c}^N T_N V_N}{V_{T < T_c}} \quad (3.14)$$

where N is the number of nodes with temperature less than T_c and the summation is divided by the total volume of the N nodes. The shaded region in Figure 3.1 illustrates this concept. Heat transfer from hot water at the top of the tank to cooler glycol at the top of the annulus (unshaded region in Figure 3.1) is not included in the model, although during energy collection, T_c is often higher than the temperature at the top of the tank. After hot and cold side temperatures are calculated, an initial guess is made for the heat exchanger wall temperature and the inside and outside heat transfer coefficients are calculated.

The inside heat transfer coefficient is based on Nusselt number correlations from Kreith as reported by Burmeister (1993) for natural convection inside a spherical cavity

$$\begin{aligned} \overline{Nu}_d &= 0.59 Ra_d^{1/4} \quad (Ra_d < 10^9) \\ \overline{Nu}_d &= 0.10 Ra_d^{1/3} \quad (Ra_d \geq 10^9) \end{aligned} \quad (3.15)$$

where the Rayleigh number, Ra , is the product of the Grashof and Prandtl numbers and is calculated with temperature dependent fluid properties, the temperature difference between the heat exchanger wall and the water, and the depth of the heat transfer portion of the tank (d). The outside (collector fluid) heat transfer coefficient is calculated assuming fully developed, constant property, laminar flow in an annulus with one surface insulated and the other at constant temperature. This results in a constant Nusselt number (Kay and Perkins, 1972)

$$\overline{Nu}_L = 4.86. \quad (3.16)$$

The heat transfer convection coefficient for each surface is then calculated as

$$h = \frac{\overline{Nu}_L k}{L} \quad (3.17)$$

where the conductivity (k) is a polynomial function of temperature for both the water and glycol solution and the length (L) is the heat transfer vertical depth for the water side and is twice the annular gap (the hydraulic diameter) for the shell side. The overall heat transfer coefficient-area product is then

$$UA_{HX} = \frac{1}{\frac{1}{h_{in}A} + \frac{1}{h_{out}A} + R_{wall}} \quad (3.18)$$

where R_{wall} is the tank wall resistance.

The effectiveness and total heat transfer are then calculated using a NTU (number of transfer units) relation assuming the heat capacity of the water is much larger than that of the glycol solution

$$\varepsilon = 1 - \exp\left(\frac{-UA_{HX}}{\dot{m}C_{p,g}}\right), \text{ and} \quad (3.19)$$

$$\dot{q}_{in} = \varepsilon \dot{m}C_{p,w}(T_c - T_w). \quad (3.20)$$

The above procedure is repeated, varying the heat exchanger wall temperature until two successive calculations of the heat transfer, $\dot{q}_{in}$, are approximately equal. The new tank node temperatures are then calculated on a node-by-node basis by solving the first order linear differential equation

$$(mC_{p,w})_{elem} \frac{dT_{elem}}{dt} = (UA)_{elem}(T_{c,mean} - T_{elem}) \quad (3.21)$$

where T_{elem} is the current tank element temperature and $T_{c,mean}$ is the mean temperature of the collector fluid in contact with the current tank element. The subscript *elem* means that the overall value of mC_p or UA_{HX} (as calculated in the single node approximation) has been multiplied by the fractional volume or area of the current element. The left-hand side of eqn. 3.21 represents the change in internal energy of the current element and the right hand side is the energy flow into the element from the heat exchanger. Equation 3.21 is iterated first at the highest tank node where $T_{elem} < T_c$. Iterations continue until the mean collector fluid temperature in contact with the node essentially stops changing. A new

mean collector fluid temperature is then calculated and used with the next lower node in the tank and the process continues to the bottom of the tank.

The overall energy transferred into the tank becomes

$$\dot{q}_{in} = \sum_1^N \dot{m} C_{p,g} (T_{N,i} - T_{N,f}) \quad (3.22)$$

where the subscripts i and f represent the initial and final mean temperature of the collector fluid in contact with the tank nodes. Note that the energy quantity calculated in eqn. 3.20, used iteratively to determine the overall heat transfer-area product (UA_{HX}) of the heat exchanger, is reassigned in this step.

The temperature returned to the collector is the temperature of the collector fluid in contact with the last (bottom) tank node after heat transfer has occurred

$$T_{ret} = T_i - \frac{(mC_{p,w})_{elem} (\Delta T)_{water}}{\dot{m} C_{p,g} \Delta t} \quad (3.23)$$

where T_i is the temperature of the collector fluid when the heat transfer iterations reach the bottom node in the tank.

3.2.4 Tank Model

This section describes treatment of the potable water in the inner tank of the tank/heat exchanger. The model assumes a one-dimensional thermally stratified tank by dividing the tank into uniform temperature horizontal nodes that move with any load flow. By formulating energy balances for the moving segments of fluid, the relatively rapid convective transfer between nodes may be handled by a record keeping process thus

allowing larger timesteps to be used in the model (Morrison and Braun, 1985). Treatment of conduction between adjacent water layers, tank losses to ambient temperature, load flows, and auxiliary heating are discussed here.

Conduction between adjacent water layers and energy losses to the environment are handled using a heat balance on each tank element or moving plug

$$\left(mC_p\right)_i \frac{dT_i}{dt} = (UA)_i (T_a - T_i) + \left(\frac{kA}{\Delta Z}\right)_{i+1} (T_{i+1} - T_i) + \left(\frac{kA}{\Delta Z}\right)_{i-1} (T_{i-1} - T_i) \quad (3.24)$$

where the index i refers to the current element and $i-1$ and $i+1$ refer to the elements above and below, respectively. Figure 3.2 illustrates this energy balance on a tank with three nodes. Note that energy must first pass through the glycol in the annulus before reaching ambient conditions. This is not included in the model.

Figure 3.2 Tank Energy Balance

The overall heat loss coefficient-area product for the tank, $(UA)_i$, is calculated for each node based on the outer area of the sides and ends of the node. The effective conductivity

term, $kA/\Delta Z$, for each element includes conduction between layers of water and along the tank wall

$$\frac{kA}{\Delta Z} = \frac{k_{wall}A_{wall}}{\Delta Z} + \frac{k_{water}A_{elem}}{\Delta Z} \quad (3.25)$$

where A_{wall} is the cross sectional area of the tank wall in contact with the current element, A_{elem} is the area of contact between two elements, and ΔZ is the conduction path length.

Load flow into and out of the tank is handled as illustrated in Figure 3.3. A segment of fluid corresponding to the correct load volume is added at the bottom of the tank (gray area in Figure 3.3). Tank nodes are moved upward to accommodate the new element and the delivered temperature is the temperature of the portion above the top of the tank (volume weighted average of the temperature if more than one node is involved).

Figure 3.3 Load Flow Illustration

When the auxiliary heater is enabled, the tank node at the heating element is divided into two nodes if necessary at the actual position of the auxiliary. The node containing the heating element is heated until it reaches the temperature of the node above it. These two nodes are heated together and the process continues until the maximum heater input is used or the set point temperature is reached.

The tank model also includes routines to split large tank elements into two when conduction is being considered and to aggregate small tank elements to avoid convergence problems and keep the number of elements below a preset maximum. Temperature inversions due to top loss, load draw, auxiliary heating, and heat transfer from the collector fluid are corrected by mixing the new element with elements above it to eliminate the inversion.

3.2.5 Model Solution Procedure

Figure 3.4 shows a simplified flow chart of the model solution procedure which starts by guessing the collector flow rate. The temperature distribution around the thermosyphon loop is known from the previous timestep.

Figure 3.4 Model Solution Procedure (One Time Step)

CHAPTER FOUR

A PROPOSED SIMULATION CALIBRATION METHOD

This chapter details the development of a proposed simulation calibration method. Previous experimental data collected using a solar simulator are described. These data are then used to calibrate a simulation of the experiments to arrive at the proposed calibration method.

4.1 Previous Experimental Data and Simulation

Bickford and Hittle (1995) recently reported experimental measurements on the performance of the thermosyphon SDHW system used in this study. Their experiments were performed using the indoor solar simulator at the Solar Energy Applications Laboratory (SEAL) at Colorado State University. The simulator is a unique facility that allows for full-size SDHW systems to be tested under constant irradiation conditions over extended periods of time (Kenny, 1993). Instrumentation for the simulator tests was essentially the same as that described for the outdoor tests in Chapter Two with the

addition of a low-pressure-drop Coriolis-effect mass flow meter in the inlet pipe to the collector so that the thermosyphon mass flow rate could be measured.

A variety of four and eight hour constant irradiation tests with and without load draws were performed. A four hour test with a 420 second draw at an average flow rate of 0.21 liters per second at two hours and an eight hour test with no draw are used for this study. The draw volume for the four hour test was about one-half of the total tank volume.

Bickford and Hittle simulated the results of their experiments using the modified thermosyphon system model described in Chapter Three. Figure 4.1 shows a comparison of the collector inlet and outlet temperatures during the four hour test. The predicted temperature rise across the collector is larger than in the experiment.

Figure 4.1 Collector Inlet (Bottom) and Outlet (Top) Temperature Profiles During Four Hour Test With Draw

Figure 4.2 shows the predicted and actual collector flow rate. The qualitative shapes of these curves are discussed by Bickford and Hittle, 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.

Figure 4.2 Collector Flow Rate Profiles During Four Hour Test With Draw

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. Although this agreement is reasonable, it arises due to compensating errors in the model.

Figure 4.3 compares the average tank temperature as a function of time over the four hour test. 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. The TRNSYS average temperature is computed as a weighted average of the tank node temperatures. The simulation predicts a higher average tank temperature before the draw with a larger drop during the draw.

Figure 4.3 Average Tank Temperature Profiles During Four Hour Test With Draw

Figure 4.4 shows the tank draw temperature as a function of fractional tank volume during the 420 second draw. The predicted draw temperature starts off higher and then drops faster than the corresponding experimental draw temperature. This indicates that there is a higher degree of vertical stratification in the model tank than in the experimental tank. The measured energy delivered during the 420 second draw is 7412 kJ. The model predicts an energy delivery of 9503 kJ which is 28 percent larger.

With no adjustment, the TRNSYS Thermosyphon System model does not effectively simulate this four hour solar-only constant irradiation test with a fixed volume draw.

Figure 4.4 Draw Temperature Profiles

4.2 Improving the Simulation

4.2.1 Possible Means of Simulation Calibration

By adjusting one parameter such as the tank heat loss (UA) value or the collector heat loss parameter ($F_{av}U_L$), the simulation model could be "calibrated" to match any single experimental outcome, such as the draw energy. However, if the model does not match the details of the development of the system temperatures and thermosyphon flow

rate there can be little confidence in the simulation model's ability to predict system performance under widely varying conditions. More likely, several parameters will have to be adjusted in order to obtain a more accurate match between the simulation and the experiment.

In this case, the model consists of a standard flat plate collector that is coupled to a heat exchanger and a thermosyphon calculation. The performance of the individual components is highly coupled and modification of any one of them will affect the others making multiple parameter adjustment difficult. Therefore, in the procedure used here, the thermosyphon flow calculation in the collector loop is disabled and the collector model is forced to follow exactly the measured collector inlet temperature and flow as shown in Figures 4.1 and 4.2. 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.

4.2.2 Decoupled Model Simulation Results

Figure 4.5 shows the predicted and experimental collector outlet temperatures when the measured experimental flow rate and collector inlet temperature shown in Figures 4.1 and 4.2 are used as inputs to the model (this is called the "forced model" hereafter). With two exceptions, the predicted collector outlet temperature is within about one half of one degree of the experimental collector outlet temperature. The discrepancies at the beginning of the test and during the draw occur because the TRNSYS model does

not account for the thermal capacitance of the system. The experimental flow rate, being forced into the model, is very low at the start of the test (Figure 4.2), which causes the model immediately to predict a very large collector temperature rise. In reality, the collector takes some finite time to respond to the step change in irradiation and does not experience a large temperature rise immediately.

Figure 4.5 Collector Outlet Temperature Profiles With Forced Collector Flow Rate and Inlet Temperature

Examination of Figure 4.2 shows that the measured (and predicted) thermosyphon flow increases as the average temperature of the fluid in the collector/heat exchanger loop increases with time (due to a decrease in fluid viscosity). During the draw, the average temperature in the collector/heat exchanger loop will drop as the cold water enters the tank. The thermosyphon flow predicted by the model immediately predicts a

corresponding drop in flow. In the experiment, the flow first responds to only a drop in the collector inlet temperature which causes the flow rate to increase. The eventual drop in the average loop temperature and corresponding drop in flow rate is delayed to well beyond the end of the draw. The current model (and TRNSYS in general) will not reproduce this same trend.

Even with the discrepancies mentioned above, the experimental and predicted total collector useful energy differ by less than one percent indicating that 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 4.6 compares the collector inlet temperature predicted by the forced model with the experiment.

Figure 4.6 Collector Inlet (Heat Exchanger Outlet) Temperature Profiles With Forced Collector Flow Rate and Inlet Temperature

The predicted temperature out of the heat exchanger (into the collector) is significantly lower than the measured value throughout the test. Since the measured and predicted temperature and flow rate entering the heat exchanger are essentially the same, this result implies that the predicted heat transfer to the tank water is larger than in the experiment. This suggests that the model overestimates the performance of the heat exchanger and a modification of the heat exchanger overall heat transfer coefficient (UA_{HX}) calculation is required.

Figure 4.7 shows the average tank temperature for the model and the experiment.

Figure 4.7 Average Tank Temperature Profiles With Forced Collector Flow Rate and Inlet Temperature

As discussed in relation to Figure 4.3, 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 conductance.

Figure 4.8 shows the predicted and actual draw temperature during the 420 second draw. The model predicts higher draw temperatures and higher stratification as discussed in relation to Figure 4.4. 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.

Figure 4.8 Draw Temperature Profiles With Forced Collector Flow Rate and Inlet Temperature

4.2.3 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.

Since the predicted inlet temperature profile to the heat exchanger (collector outlet) is very close to the experimental profile, adjusting UA_{HX} 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 during the draw. The eight hour test is modeled by forcing the flow rate and collector inlet temperature as with the four hour test. UA_{HX} is adjusted until the predicted collector inlet temperature profile matches the experimental profile as closely as possible. A multiplicative factor of 0.70 applied to the UA_{HX} calculation in the model minimizes the RMS error in the prediction.

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 as described in Chapter Three. 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 if the top of the tank is cooler than the collector fluid. 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).

4.2.4 Final Simulation Results

The modifications calculated using the forced model are applied to the non-forced model. The heat exchanger conductance (UA_{HX}) value is multiplied by 0.70 and the equivalent conductivity term is multiplied by 27 throughout the simulation. When this is done, the collector energy is within four percent of the experimental value and the draw energy is within one percent of the experimental value, but the collector flow rate is still underpredicted and the temperature rise is overpredicted (resulting in good agreement in 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 multiplied by a factor of 0.60. About eighty percent of the predicted total pressure drop is in the collector risers. Since the collector is made with two stamped plates joined together, the uncertainty in the effective riser diameter is large. This may account for a large portion of the error in the pressure drop calculation.

Figure 4.9 shows the collector inlet and outlet temperatures for the final model and the experiment.

Figure 4.9 Collector Inlet (Bottom) and Outlet (Top) Temperature Profiles For Calibrated Model

The profiles match within two degrees except at the start of the experiment and during the draw. Figure 4.10 shows the collector flow rate. Again the profiles match very well. Figure 4.11 shows the predicted and actual draw temperature profiles. The predicted temperatures in the profile have an RMS error of 1.9 °C compared to the original RMS error of 5.6 °C (Figure 4.4). The energy results are summarized in Table 4.1.

Figure 4.10 Collector Flow Rate Profiles for Calibrated Model

Figure 4.11 Draw Temperature Profiles for Calibrated Model

	Experiment	Original Model	Calibrated Model	% Error Original Model	% Error Calibrated Model
Collector Energy (kJ)	19626	21053	19828	+7	+1
Draw Energy (kJ)	7412	9503	7445	+28	+0.4

Table 4.1 Integrated Energy Results

4.3 A Proposed Simulation Calibration Method

Based upon the results obtained above, a simulation calibration method may be proposed for thermosyphon systems of this type (and perhaps other types). The three adjustments described in the above sections may be made with a simple indoor test using a simulated collector followed by a simple outdoor test.

4.3.1 Indoor Tests

Indoor testing involves connecting the tank/heat exchanger apparatus to a water source and a boiler apparatus. The boiler apparatus, used to simulate the solar collector, will consist of a heating unit of sufficient power, a pump, a flow meter capable of accurately measuring low flow rates, a valve to control flow rate, and associated temperature sensors and control hardware. The water source will consist of a sufficiently large water conditioning system so that the tank/heat exchanger apparatus may be filled with constant temperature water. The boiler apparatus, water conditioning system, and experimental arrangement are described in Chapter Two.

The goal of the indoor tests with a simulated collector is to adjust the heat exchanger overall heat transfer coefficient-area product (UA_{HX}) and the effective conductivity term (eqn. 3.25). To accomplish this, the following procedures are suggested; the first is to be done with the auxiliary element disabled and the second is to be done with the auxiliary element enabled.

4.3.1.1 Indoor Test Without Auxiliary

1. Create A TRNSYS input deck using measured and specified system parameters for the system under study. System orientation should be the same as for the outdoor test. It is suggested that the system faces south and has a tilt angle close to the latitude of the testing location. Initialize the one-day simulations with an isothermal tank at 20°C.
2. Using the TRNSYS deck and weather data files from at least two locations, simulate the performance of the system on the equinoxes and the solstices after making sure that the four days in question are clear ($K_T > 0.7$). If a day is not clear, use a day before or after the day in question. Record the peak useful power and corresponding thermosyphon flowrate from the four one-day simulations and use the largest values for the experimental test. The purpose of this step is to determine the approximate peak operating conditions of the system.
3. Using the boiler apparatus, perform a test with the flow rate and useful energy determined in the previous step starting with an isothermal 20°C tank. Heat the system for four hours with the boiler and then perform a complete energy (approximately three volume) draw at a flow rate of approximately 0.19 liters per second (3.0 GPM). For the tank in this system, this corresponds to a draw of approximately 45 minutes.

4. Modify the TRNSYS model being used to accept flow rate and heat exchanger inlet temperature as two additional inputs (this eliminates the thermosyphon flow rate calculation from the model). Also add multiplicative adjustment factors to the UA_{HX} calculation, the effective conductivity calculation, and the total closed-loop frictional pressure drop calculation. In this study, this was accomplished by adding three additional parameters to the Type 45 Thermosyphon System model.
5. Using the modified (forced) model, simulate the experimental data and adjust the UA_{HX} factor to minimize the RMS error between the experimental and simulated temperature profiles leaving the heat exchanger. For the remainder of the procedure, leave the calculated adjustment factor at the value calculated in this step.
6. Adjust the conductivity factor to minimize the RMS error between the experimental and simulated draw temperature profiles at the end of the test. For the remainder of the procedure, leave the calculated adjustment factor at the value calculated in this step.

4.3.1.2 Indoor Test With Auxiliary

When the auxiliary element is enabled, the solar storage volume is reduced by about one-half, significantly affecting system performance. Thus, a separate indoor test is performed with the auxiliary heating element enabled. The suggested procedure is outlined below.

1. Follow steps 1 and 2 from section 4.3.1.1 for the non-auxiliary tests except initialize the bottom of the tank to 20°C and the top of the tank to the selected setpoint (55°C

in this study). Note that in the TRNSYS Type 45 subroutine, the top portion of the tank is initialized to the setpoint temperature by default.

2. Starting with an isothermal 20°C tank, heat the system using only the auxiliary element until it is switched off by the thermostat. Then heat the system for four hours using the boiler at the flow rate and power input determined above for four hours. Following the two heating stages, perform a complete energy draw at a flow rate of approximately 0.19 liters per second (3.0 GPM).
3. Modify the TRNSYS model being used as in step 4 from section 4.3.1.1. In addition, modify the model so that the portion of the tank above the heating element is initialized to the same temperature as the lower half of the tank instead of the setpoint temperature. This allows use of the model to simulate auxiliary heating to the setpoint and simulated solar heating using the boiler.
4. Confirm that the model is correctly predicting the integrated auxiliary energy input and then follow steps 5 and 6 from the procedure in section 4.3.1.1.

4.3.2 Outdoor Test

Outdoor testing involves installing the system, including the collector, outdoors and performing a one-day test to determine the adjustment factor required for the frictional pressure drop calculation in the collector loop. In this case, only one test is performed since the use of the auxiliary element will not directly affect the pressure drop in the closed glycol loop. The system must be instrumented to measure appropriate

temperatures, total horizontal radiation, and draw flow rate. The outdoor experimental facility is described in Chapter Two. The following procedure is suggested:

1. Starting with an isothermal 20°C tank before sunrise, collect temperature and radiation data on a day with at least three consecutive hours of clear sunshine. Perform a complete energy draw at the end of the test.
2. Using the thermosyphon system model with the TRNSYS radiation processor, simulate the experiment (leaving the UA_{HX} and conductivity adjustments at the values calculated from the procedure described in section 4.3.1.1). Adjust the total closed-loop frictional pressure drop calculation (sum of all head losses) to minimize the RMS error between the experimental and predicted collector temperature rise during solar heating.

Note that no direct flow measurement in the collector loop is required for this adjustment. The flow rate is inferred from the collector useful energy gain which is accurate after the heat exchanger and conductivity adjustments as discussed in section 4.2.4.

CHAPTER FIVE

MODEL CALIBRATION RESULTS

This chapter reports model calibration results using the calibration technique described in Chapter Four.

5.1 Selection of Calibration Operating Conditions

The operating conditions are selected as recommended in Chapter Four by simulating the system under investigation using the TRNSYS model described in Chapter Three. Data from Denver, Colorado and Sacramento, California are used for the simulation. Resulting daily peak power and flow rate data are given in Table 5.1 (auxiliary off) and Table 5.2 (auxiliary on). Based upon this data, the selected test point is approximately 1400 Watts at a flow rate of 38 kg/hr with no auxiliary and approximately 1200 Watts at a flow rate of 42 kg/hr with the auxiliary.

	Sacramento, CA		Denver, CO	
Day Near	Power (W)	Flow (kg/hr)	Power (W)	Flow (kg/hr)
March 21	1400	38	1111	30
June 21	1280	36	1170	33
September 21	1380	38	1350	38
December 21	1020	29	780	24

Table 5.1 Daily Simulated Peak Power and Flow Rate Without Auxiliary Heater

	Sacramento, CA		Denver, CO	
Day Near	Power (W)	Flow (kg/hr)	Power (W)	Flow (kg/hr)
March 21	1130	40	1140	40
September 21	1160	42	1100	39

Table 5.2 Daily Simulated Peak Power and Flow Rate With Auxiliary Heater

5.2 Indoor Testing and Model Adjustments

Indoor testing consists of constant-flow, constant heat-flux tests for calibration purposes both with and without the auxiliary heating element.

5.2.1 Without Auxiliary

Testing consists of filling the tank with constant temperature water at 20°C, turning on the boiler at the flow rate and heat input determined above, letting the system run for four hours, disabling the boiler, and performing an extended draw on the water in the tank. The flow rate and heat exchanger inlet glycol temperature data are then forced into the TRNSYS model and the appropriate adjustments are made.

Figure 5.1 shows the experimental heat exchanger inlet and outlet glycol temperatures and the outlet temperature predicted by the model when it is forced as described in Chapter Four. The inlet temperature is the top line on the graph. Note that the model predicts a lower temperature leaving the heat exchanger suggesting that more heat is being transferred in the model than in the experiment. Also note that quasi-steady-state is reached at about 2500 seconds after the associated hardware heats up. To avoid start-up capacitance effects, model adjustments are made during the period from 3600 to 14400

seconds after the start of the test. The RMS error between the TRNSYS and experimental outlet temperature profiles is 5.3°C .

Figure 5.1 Heat Exchanger Inlet and Outlet Temperature Profiles With No Auxiliary Input Before Calibration

The overall heat exchanger conductance (UA_{HX}) is then adjusted until the RMS error between the outlet temperature profiles is minimized. This requires that the conductance be multiplied by a factor of 0.59 or reduced by approximately 40 percent. The RMS error between the two profiles is reduced to 0.37°C . Figure 5.2 shows the predicted temperature profile along with the same two experimental profiles as in Figure 5.1. The new predicted profile lies almost exactly on top of the experimental heat exchanger outlet profile. Table 5.3 shows the amount of energy transferred to the tank

water for the experiment, the unadjusted model, and the calibrated model. The calibrated model matches the experimental result to well within the experimental uncertainty.

Figure 5.2 Heat Exchanger Inlet and Outlet Temperature Profiles With No Auxiliary Input After Calibration

	Energy (kJ)	Percent Error (%)
Experiment	20890 ± 540	0.0
Unadjusted Model	24230	+ 16.0
Calibrated Model	20840	- 0.24

Table 5.3 Energy Transferred Through Heat Exchanger

After modification of the UA_{HX} calculation, the draw at the end of the test is studied to adjust the effective conductivity term in the model. Figure 5.3 shows the experimental water draw temperature and supply temperature along with the model's

prediction for the draw temperature. The experimental water supply temperature profile is input to the model. It can be seen that the model predicts an outlet temperature that is about 16°C larger than the experimental outlet temperature at the start of the draw. From Table 5.3, the correct amount of energy is being transferred to the tank. Thus, the model predicts more tank stratification than in the experiment. The RMS error between the two profiles is 6.2 °C for the three volume draw.

Figure 5.3 Draw and Supply Water Temperature Profiles With No Auxiliary Input Before Calibration

Figure 5.4 shows the same temperatures in Figure 5.3 after the effective conductivity term has been adjusted to minimize the RMS error between the draw temperature profiles. The effective conductivity term is multiplied by a factor of forty-four throughout the test. Possible reasons for the large size of this adjustment are discussed in

Chapter Four. The RMS error between the temperature profiles is reduced to 1.1 °C. The predicted profile now starts about a degree below the experimental profile. Table 5.4 compares draw energy values. The integrated energy error is decreased to well within the experimental uncertainty.

Figure 5.4 Draw and Supply Water Temperature Profiles With No Auxiliary Input After Calibration

	Energy (kJ)	Percent Error (%)
Experiment	20600 ± 1070	0.0
Unadjusted Model	26000	+ 26
Calibrated Model	21060	+ 2.2

Table 5.4 Draw Energy Values

5.2.2 With Auxiliary

Testing with the auxiliary element involves filling the tank with constant temperature water at 20°C and turning on the heating element with no load or heat exchanger flow. After the thermostat turns the auxiliary heater off (about four hours in this situation), signifying that the preheat portion of the tank is at the setpoint temperature, the boiler is switched on at the flow rate and heat input determined in section 5.1. The system is then run for four hours, the boiler is disabled, and a draw is performed as before.

TRNSYS is used to simulate the auxiliary energy input by enabling the auxiliary element in the model for the same length of time it is on during the experiment. However, the model predicts an integrated auxiliary energy of 17,430 kJ while the actual energy input is 23,100 kJ. Figure 5.5 shows the average exterior jacket thermocouple temperatures at four levels in the tank during auxiliary heating. The third line from the top in the figure is at a point three-quarters of the way down the tank, well below the heating element. Significant heating, seen as the temperature rises above 30°C, is occurring in this zone. The top of the tank is also significantly hotter than the 55°C thermostat setpoint.

The heat exchanger adjustment during simulated solar heating should be made with the same energy in the tank (even if it is distributed differently), so the setpoint in the model is adjusted until the integrated auxiliary energy inputs are essentially the same before making the heat exchanger adjustment. This requires raising the model setpoint to 67.6 °C. Table 5.5 summarizes this result. Following this adjustment, model calibration is continued as before.

Figure 5.5 Exterior Jacket Average Thermocouple Temperatures

	Setpoint (°C)	Energy (kJ)	Percent Error (%)
Experiment	55	23100 ± 2070	0.0
Model	55	17430	- 25
Model	67.6	23050	- 0.2

Table 5.5 Auxiliary Energy Input Prior to Calibration

Figure 5.6 shows the experimental heat exchanger inlet and outlet glycol temperatures and the outlet temperature predicted by the forced model. The RMS error between the outlet profiles is 3.0 °C. Figure 5.7 shows the two experimental profiles with the predicted profile after the overall heat exchanger conductance is multiplied by a factor of 0.63. The RMS error between the outlet profiles is reduced to 1.8 °C. Table 5.6

summarizes the integrated energy transfer during simulated solar heating. In the auxiliary case, the heat exchanger conductance adjustment does not improve model performance as well as in the non-auxiliary case.

Figure 5.6 Heat Exchanger Inlet and Outlet Temperature Profiles With Auxiliary Before Calibration

	Energy (kJ)	Percent Error (%)
Experiment	15191 ± 367	0.0
Unadjusted Model	17180	+ 13.0
Calibrated Model	15990	+ 5.3

Table 5.6 Energy Transferred Through Heat Exchanger After Auxiliary Heating

Figure 5.7 Adjusted Heat Exchanger Inlet and Outlet Temperature Profiles With Auxiliary After Calibration

Following the heat exchanger adjustment, the effective conductivity term is adjusted to minimize the error between the experimental and predicted draw temperature profiles. This requires that the effective conductivity be multiplied by a factor of fourteen throughout the test. After calibration with the raised setpoint, the model predicts a large auxiliary energy input of 26,740 kJ. This is expected due to the increase in conduction, and thus heating required, to the lower portion of the tank after the conductivity adjustment. To correct this, the setpoint is returned to the experimental value of 55 °C resulting in an auxiliary energy input of 24,530 kJ which is about six percent larger than the experimental value, but within the experimental uncertainty.

Figures 5.8 and 5.9 show the experimental and predicted draw temperature profiles and experimental supply temperature profile before and after the calibration. The RMS error decreases from 5.3 °C to 1.8 °C. Tables 5.7 and 5.8 compare the auxiliary energy input and the integrated draw energies, respectively. Note in Table 5.8 that the error in the integrated draw energy increases after adjustment even though the RMS temperature error decreases considerably. However, both integrated draw energy values are within experimental uncertainty.

Figure 5.8 Draw and Supply Water Temperature Profiles With Auxiliary Before Calibration

	Setpoint (°C)	Energy (kJ)	Percent Error (%)
Experiment	55	23100 ± 2070	0.0
Cal. Model	67.6	26740	+ 16
Cal. Model	55	24530	+ 6.6

Table 5.7 Auxiliary Energy Input After Calibration

Figure 5.9 Draw and Supply Water Temperature Profiles With Auxiliary After Calibration

	Energy (kJ)	Percent Error (%)
Experiment	38034 ± 2040	0.0
Unadjusted Model	39226	+ 3.1
Calibrated Model	36205	- 4.8

Table 5.8 Draw Energy Values

5.2.3 Additional Indoor Tests

While performing simulated solar input tests with the boiler apparatus, it was discovered that the required heat exchanger conductance (UA_{HX}) adjustment to minimize the RMS error between the predicted and experimental heat exchanger outlet temperature profiles depends on the operating conditions of the system. Additional indoor tests are performed to study this effect. Initially, several constant flux tests at varying flow rates,

fluxes, and initial temperatures are performed. Table 5.9 summarizes the results from these tests.

Tests 1-3 vary the flow rate at a nearly constant power input, tests 4 and 5 vary the flow rate at similar power levels but start with a higher tank temperature, and test 6 is performed at a lower power input. Tests 1-3 show that, as the flow rate is decreased at a given power, the required adjustment increases (UA_{HX} is modified to a greater extent). Tests 4 and 5 indicate that initial tank temperature has little effect on the adjustment. Test 6 indicates that at this relatively low power level, the heat exchanger equations work well.

Test #	Initial Tank Temperature	Average HX Power Input	Average Flow Rate	UA_{HX} Adj. Factor
	°C	W	kg/hr	
1	14.5	1154	48.3	0.75
2	14.3	1052	33.2	0.52
3	14.9	860	20.9	0.32
4	29.1	1122	50.2	0.79
5	29.2	1030	36.1	0.53
6	15.1	548	50.3	1.01

Table 5.9 Heat Exchanger UA_{HX} Adjustment Under Different Operating Conditions

After these tests are completed it is decided that a more in-depth analysis of the heat exchanger performance calculation would be beneficial. A successful attempt to determine improved empirical heat transfer correlations in the heat exchanger is discussed in the following chapter.

5.3 Outdoor Testing and Model Adjustment

The outdoor test consists of using data from a reasonably clear day to adjust the sum of all the pressure drops in the closed loop. Figure 5.10 shows the total horizontal

radiation during the selected day. The morning is very clear and the afternoon is partly cloudy. Model calibration is performed using data between the hours of 7:30 and 11:30 a.m..

Figure 5.10 Horizontal Radiation

Figures 5.11 and 5.12 show the experimental and predicted collector temperature rise during the morning hours before and after calibration. A multiplicative factor of 0.41 is applied to the closed-loop friction pressure drop to minimize the RMS error between the temperature profiles. Again the large uncertainty in the actual hydraulic diameter of the collector risers may explain the size of this adjustment.

Figure 5.11 Collector Temperature Rise Before Friction Calibration

Figure 5.12 Collector Temperature Rise After Friction Calibration

5.4 Summary of Suggested Model Adjustments

Table 5.10 summarizes suggested model adjustments for no auxiliary input and for auxiliary input for the SDHW system under investigation. Note that use of the auxiliary element significantly decreases the required adjustment to the effective conductivity term. This is due to the reduced size of the solar storage portion of the tank when the auxiliary heating element is used.

	Heat Transfer Coefficient - Area Product (UA_{HX})	Effective Conductivity	Closed Loop Friction Pressure Drop
No Auxiliary	0.59	44	0.41
Auxiliary	0.63	14	0.41

Table 5.10 Summary of Suggested Model Adjustments

The following chapter describes an experimental determination of new heat transfer correlations in the heat exchanger that may be used as an alternative to the UA_{HX} calibration factor determined in this chapter.

CHAPTER SIX

HEAT EXCHANGER ANALYSIS

Calculation of the overall heat transfer coefficient in the model is described in Chapter Three. This chapter examines the heat transfer correlations used in this calculation and describes the development of new correlations.

6.1 Outside (Shell) Heat Transfer Coefficient

The heat transfer coefficient on the annulus side of the heat exchanger is

$$\overline{Nu_L} = 4.86. \quad (6.1)$$

The assumptions made in reaching this result are: laminar flow, fully-developed flow, constant properties, one surface insulated and the other at constant temperature, and forced convection. Each of these assumptions is discussed below.

6.1.1 Laminar Flow

In fully developed flow, the onset of turbulence occurs at a Reynolds number of approximately 2300. Thus, if the Reynolds number in the annular gap is well below this for any realistic operating conditions, laminar flow may be assumed. The Reynolds number based on the hydraulic diameter at the largest experimentally observed flow rate in the solar simulator (45 kg/hr) is about thirty-five which is certainly in the laminar regime.

Even if the flow in the annulus did not completely envelop the inner tank and flowed through only a fraction of the annulus, the Reynolds number would remain in the laminar regime.

6.1.2 Fully Developed Flow

Since the annular gap in the heat exchanger is small compared to the radius of the tank, flow in the annular gap may be approximated as flow between parallel plates. Schlichting (1979) gives the entrance length for pressure driven uniform initial velocity flow between parallel plates as

$$l_e = 0.04dRe_d \quad (6.2)$$

where d is the channel diameter, or annulus gap in this case, and Re_d is the Reynolds number based on the channel diameter. Table 6.1 shows the ratio of the entrance length to the tank length at several typical operating conditions.

Flow Rate [kg/hr]	Glycol Temperature [C]	Entrance Length/ Tank Length
20	30	0.001
45	30	0.003
45	80	0.010

Table 6.1 Heat Exchanger Entrance Length

The entrance length is always less than one percent of the overall length. The last two rows in the table show that entrance length is sensitive to the fluid temperature due to viscosity dependence in the Reynolds number.

Values in Table 6.1 assume that the flow enters the annulus and immediately rises to pass through the entire annulus. This is likely not the case. From eqn. 6.2, the entrance length is linearly dependent on the Reynolds number which is linearly dependent on the fluid velocity in the annulus. Thus, if the fluid were to pass through only the lower quarter of the annulus, the velocity would increase by a factor of four and the ratio in the last column of the last row would increase by a factor of four to become 0.04 or four percent of the tank length. Still, the assumption of fully developed flow seems reasonable.

Even if the flow is fully developed in its velocity profile, it may not be fully developed thermally. Burmeister (1993) gives the thermal entrance region for laminar heat transfer in ducts as

$$0 \leq \frac{L}{D_h \text{Re}_{D_h} \text{Pr}} = \frac{1}{Gz} \leq 0.05 \quad (6.3)$$

where L is the tank length, D_h is the hydraulic diameter, Pr is the Prandtl number, and Gz is the Graetz number. The second column in Table 6.2 gives the reciprocal of the Graetz number at several typical operating conditions. Even at the lowest flow rate, the flow is still thermally developing at the end of the tank. The third column in Table 6.2 gives the limiting Nusselt number for laminar flow in a concentric-circular annulus with constant wall heat flux from Kays as reported by Burmeister (1993). Note that values in Table 6.2 are after the fluid has passed through the full length of the tank. Nusselt numbers in the earlier region of the tank will be even larger.

Flow Rate [kg/hr]	1/Gz	Nu
20	0.021	5.5
30	0.014	6.5
45	0.0093	8.0

Table 6.2 Heat Exchanger Thermal Entrance Length and Nusselt Number

6.1.3 Constant Property Flow

The assumption of constant properties implies that at any given moment the temperature ranges in the heat exchanger annulus are not large enough to affect fluid property values significantly. The largest difference between the heat exchanger inlet and exit temperatures that has been observed is about 35 °C. Table 6.3 lists relevant properties of a 25 percent propylene glycol-water solution and their approximate percent variance over a 35 °C temperature range.

Property	Percent Change
Density (ρ)	2
Dynamic Viscosity (μ)	20
Specific Heat (C_p)	2
Thermal Conductivity (k)	5

Table 6.3 Glycol Property Variations

The only significant change in property value occurs in the viscosity. Burmeister (1993) describes a technique to incorporate property variation effects on the Nusselt number using an equation of the form

$$\frac{Nu}{Nu_m} = \left(\frac{\mu}{\mu_m} \right)^{0.5} \quad (6.4)$$

where the subscript m denotes the Nusselt number or viscosity at a reference temperature such as the mixing cup or surface temperature and the exponent arises due to laminar flow. The exponent reduces the error in the Nusselt number prediction to about ten percent in this case. Incorporation of this technique into the model might serve to improve performance.

6.1.4 One Surface Insulated, One at Constant Temperature

The outer surface of the heat exchanger is insulated, but does have a small amount of heat transfer across it. The initial heat transfer calculation is based on a calculated average temperature of water in the heat transfer region of the tank. The assumption that the tank water is isothermal in the heat transfer region is not easily confirmed or denied without a large number of interior thermocouples or other flow visualization.

6.1.5 Forced Convection

Since the flow in the annulus occurs with low Reynolds numbers, there is a possibility that combined forced and natural (free) convection is occurring. The ratio of buoyancy to inertial forces serves as an indicator of this ratio (Burmeister, 1993)

$$\frac{F_b}{F_i} \approx \frac{Gr_{D_h}}{Re_{D_h}^2} \quad (6.5)$$

where the left-hand side of eqn. 6.5 is the ratio of buoyancy to inertial forces, Gr is the Grashof number and Re is the Reynolds number in the annulus. If this ratio is on the order of unity, both free and forced convection are important. If the ratio is much larger than one, free convection is dominant and if the ratio is much less than one, forced convection is dominant. The Reynolds number calculation is straightforward and its magnitude has been discussed above. The Grashof number is given by

$$Gr = \frac{g\beta(\Delta T)D_h^3}{\nu^2} \quad (6.6)$$

where ΔT is the temperature difference between the tank wall and the mean glycol temperature in the annulus, ν is the kinematic viscosity, β is the coefficient of volumetric expansion, and g is the acceleration due to gravity. The uncertainty in this analysis arises when choosing the temperature difference. Since the annular gap is very small, it seems likely that there will be a relatively small temperature difference between the tank wall and the fluid plug adjacent to it, perhaps one degree Celsius or less. Table 6.4 shows the resulting ratio for two flow rates under typical operating conditions.

Flow Rate [kg/hr]	Temperature Difference [C]	$\frac{Gr}{Re^2}$
20	1	12.5
45	1	2.5
45	0.1	0.23

Table 6.4 Ratio of Buoyancy to Inertial Forces

Table 6.4 indicates that even with a temperature difference of only 0.1 °C, forced convection does not dominate in this calculation and free convection should probably be included in the analysis.

6.2 Inside Heat Transfer Coefficient

The natural convection heat transfer coefficient inside the tank is calculated using the Nusselt number correlation

$$\begin{aligned}\overline{Nu} &= 0.59 Ra^{1/4} \quad (Ra < 10^9) \\ \overline{Nu} &= 0.10 Ra^{1/3} \quad (Ra > 10^9)\end{aligned}\tag{6.7}$$

where eqn. 6.7a is for laminar convection and eqn. 6.7b is for turbulent natural convection. The Rayleigh number, Ra , is the product of the Grashof and Prandtl numbers and is an indicator of the buoyant forces present. Equation 6.7 is based on empirical correlations from Kreith as reported by Burmeister (1993) for natural convection inside a spherical cavity.

6.3 Testing of Inside Heat Transfer Coefficient

The overall heat transfer coefficient-area product, UA_{HX} , for the heat exchanger is given in eqn. 3.18. Since the wall resistance is small (less than five percent of the total resistance under normal operating conditions), eqn. 3.18 can be approximately reduced to

$$U_{HX} \cong \frac{1}{\frac{1}{h_{in}} + \frac{1}{h_{out}}}\tag{6.8}$$

In an effort to isolate the inside heat transfer coefficient, h_{in} , the collector flow rate is increased in successive tests at constant useful energy input. As the flow rate increases, the heat transfer coefficient in the annulus, h_{out} , should increase until the inside heat transfer coefficient becomes the dominant resistance to heat transfer. U_{HX} then approaches an asymptote and assuming the outside resistance to be small, eqn. 6.8 reduces to

$$U_{HX} \cong h_{in}. \quad (6.9)$$

The experimental overall heat transfer coefficient, U_{HX} , is determined by

$$U_{HX} = \frac{\dot{m}C_{p,g}(T_{hx,in} - T_{hx,out})}{A(\Delta T)} \quad (6.10)$$

where the numerator is the useful energy delivered from the simulated 'collector', A is the surface area of the tank separating the annulus from the water, and ΔT is a temperature difference across the heat exchanger.

Selection of an appropriate temperature difference across the heat exchanger is difficult in this case due to limited temperature data and the fact that natural convection is the only heat transfer mechanism inside the tank. Two possibilities are investigated for the temperature difference. The first is simply the average of the glycol inlet and outlet temperatures less the volume-weighted average tank temperature as measured by four thermocouples inside the tank given by

$$\Delta T = \frac{(T_{hx,in} + T_{hx,out})}{2} - T_{tank,ave}. \quad (6.11)$$

The second temperature difference corresponds roughly to the definition of the log mean temperature difference for a counter-flow heat exchanger and is given by

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \quad (6.12)$$

where

$$\begin{aligned} \Delta T_1 &= T_{hx,in} - T_{tnk,top} \\ \Delta T_2 &= T_{hx,out} - T_{tnk,bot} \end{aligned} \quad (6.13)$$

Note that the log mean temperature difference in eqn. 6.12 is based on the top and bottom tank temperatures as measured by thermocouples near the top and bottom of the center of the tank. These temperatures are not exactly the 'cold outlet' and 'cold inlet' temperatures that are typically used for a counter-flow heat exchanger, and the flow of fluid in the heat exchanger is not actually 'counter-flow'. Use of parallel-flow definitions for temperature difference results in negative temperature differences and cannot be used with a log mean temperature difference.

Figures 6.1 and 6.2 show the overall heat transfer coefficient as the flow rate is increased using equations 6.11 and 6.12 at a constant useful energy input of about 1300 W. Using mean temperatures, Fig. 6.1 shows the overall heat transfer coefficient approaching 90 W/m²K as the flow rate increases and using the log mean temperature difference, Fig. 6.2 shows the overall heat transfer coefficient approaching about 95 W/m²K.

Figure 6.1 Overall Heat Transfer Coefficient at Constant Flux Using Mean Temperatures

Figure 6.2 Overall Heat Transfer Coefficient at Constant Flux Using Log Mean Temperature Difference

Thus, on the basis of eqn. 6.9 and the assumptions made in reaching it, it is concluded that the inside heat transfer coefficient is on the order of $100 \text{ W/m}^2\text{K}$ at the selected useful energy of about 1300 W. Examination of the heat transfer coefficient predicted by the TRNSYS model shows values greater than $200 \text{ W/m}^2\text{K}$ indicating that the inside heat transfer coefficient calculation is a possible source of error in the model.

6.4 Towards Improved Heat Transfer Correlations

6.4.1 Inside

By running a forced-flow test at a constant flow rate of 300 kg/h (large outside heat transfer coefficient) and a constant heat exchanger inlet temperature of approximately 60°C , a range of operating power levels may be obtained. This is performed using a proportional-integral (PI) controller built into the experimental control and data acquisition software. The test starts with an isothermal tank that is heated for four hours followed by a complete energy draw after the heating has been completed. As the tank fluid heats up, the controller reduces the energy input to the heat exchanger and thus provides a range of operating conditions necessary to create a new correlation.

Figure 6.3 shows the overall heat transfer coefficient which is the same as the inside heat transfer coefficient if eqn. 6.9 is assumed to be valid. The mean temperature difference in eqn. 6.11 is selected as the driving temperature difference since it is the closest equivalent to the temperature difference used in the model. The model uses the difference between the wall temperature and the mean water temperature as the driving potential, but the assumption of negligible wall resistance in this analysis leads to the mean

temperature difference that is used. The values range from about 135 W/m²K near the beginning of the test (after the initial transient) to about 90 W/m²K near the end.

Figure 6.3 Overall Heat Transfer Coefficient vs. Time

Figure 6.4 shows the model's predictions for the inside, outside, and overall heat transfer coefficient for the same test. The inside heat transfer coefficient is predicted to be considerably higher than the outside heat transfer coefficient until near the end of the test. The model overall heat transfer coefficient ranges from a peak of about 150 W/m²K near the start of the test to about 125 W/m²K at the end of the test which is about ten to thirty percent above the measured values. The model predicts a nearly constant outside coefficient due to the questionable assumption of fully developed forced laminar flow in the annulus.

Figure 6.4 Predicted Heat Transfer Coefficient vs. Time

We wish to obtain a correlation of the form

$$Nu = f(Ra) \quad (6.14)$$

from the experimental data to predict the Nusselt number and thus the heat transfer coefficient for the water inside the tank. Typically, the correlation is of the form

$$Nu = C(Ra)^n \quad (6.15)$$

where the constants C and n are determined by a fit to experimental data. Figure 6.5 shows a plot of the Nusselt Number as a function of the Rayleigh Number for the experiment. The line shows the fitted equation

$$Nu = 0.15(Ra)^{0.26} \quad (6.16)$$

The square of the correlation coefficient is 0.88.

Figure 6.5 Nusselt Number vs. Rayleigh Number

The Rayleigh number predicted by the model for the same test ranges from about 2×10^{10} to 8×10^{10} which is in the lower region of the experimental data. Examination of other indoor forced-flow tests show model predictions in the lower region of the data in Figure 6.5 or below. The presence of large Rayleigh numbers in the constant temperature test and not in the low-flow tests is likely due to unrealistically large heat inputs near the beginning of the constant temperature test.

In an attempt to improve model performance, eqn. 6.16 is substituted for eqn. 6.7b (turbulent) in the model and eqn. 6.7a (laminar) is modified to

$$Nu = 0.18Ra^{1/4} \quad (6.17)$$

to avoid a large discontinuity at the transition. Extrapolation of the experimental curve fit beyond the experimental data is questionable and arbitrarily reducing the constant coefficient in eqn. 6.7a is suspect, but is attempted here for investigational purposes. Figure 6.6 shows the 'old' and 'new' heat transfer correlations. The transition from laminar natural convection to turbulent natural convection is seen as the slight discontinuity at a Rayleigh number of 1×10^9 in both curves. Following the modification to the model, simulations of several low-flow experiments are performed to determine the outside Nusselt number required to predict performance accurately.

Figure 6.6 Nusselt Number Correlations

6.4.2 Outside

Table 6.5 shows the Nusselt number on the outside (shell side) of the heat exchanger required to minimize the RMS error between the experimental and predicted heat exchanger outlet temperature profiles when the forced model is modified using equations 6.16 and 6.17 for constant heat input low-flow rate tests.

The first three rows in the table show the Nusselt number decreasing with decreasing flow rate at nearly constant heat input. This is expected based on the calculation in section 6.1.2. The fourth and fifth row in the table are tests that are started with higher temperature water in the tank (about 30 °C compared to the usual 15 °C). No large effect is seen with this change. The sixth test has a reduced heat input and the seventh test has an increased heat input.

Average Energy Input (W)	Average Mass Flow Rate (kg/hr)	Nu
1100	48	7.0
1050	33	2.8
860	21	1.03
1030	36	2.87
898	25	1.26
401	19	1.22
1392	43	4.13

Table 6.5 Required Nusselt Number for Several Low-Flow Tests

Figure 6.7 shows a plot of the Nusselt number as a function of the Reynolds number for the tests in Table 6.5. The fitted equation

$$Nu = 0.035 Re^{1.94} \quad (6.18)$$

is also shown in the figure. The square of the correlation coefficient is 0.94.

Figure 6.7 Nusselt Number vs. Reynolds Number

Equation 6.18 is substituted for eqn. 6.1 in the model and simulations are performed with the new inside and outside heat transfer correlations. Table 6.6 lists the same tests as Table 6.5 with the required heat exchanger UA_{HX} adjustment to minimize the RMS error between the predicted and experimental heat exchanger outlet temperature profiles using the forced model with the new correlations. The adjustments are close to one (no adjustment) with the largest deviation being fifteen percent for the low energy test.

Average Energy Input (W)	Average Mass Flow Rate (kg/hr)	UA_{HX} Adjustment Factor
1100	48	1.12
1050	33	1.01
860	21	0.95
1030	36	0.96
898	25	0.89
401	19	1.15
1392	43	0.98

Table 6.6 Required Heat Exchanger Adjustment Factor Using New HT Correlations

6.5 Model Calibration Using Improved Heat Transfer Correlations

Equations 6.16 through 6.18 are used in the model and the calibration procedure is continued to find the required conductivity and pressure drop adjustments. Table 6.7 compares the required adjustments to the results from Chapter Five. The only significant change in the remaining two adjustments is the change from 44 to 36 of the effective conductivity adjustment with no auxiliary heating. Differences in modeling results arising due to use of the UA_{HX} adjustment from Chapter Five or the correlations developed in this chapter are discussed in the following chapter along with the results of additional outdoor testing.

		Heat Transfer Coefficient - Area Product (UA_{HX})	Effective Conductivity	Closed Loop Friction Pressure Drop
Old HT Correlations	Non-Auxiliary	0.59	44	0.41
	Auxiliary	0.63	14	0.41
New HT Correlations	Non-Auxiliary	1.00	36	0.42
	Auxiliary	1.00	13	0.42

Table 6.7 Suggested Model Adjustments With New HT Correlations

CHAPTER SEVEN

ADDITIONAL OUTDOOR DATA

Several additional outdoor tests are performed to assess the success of the model calibration results in Chapter Five and the heat transfer correlations developed in Chapter Six. This chapter summarizes experimental and modeling results for these tests.

Calibrated results in this chapter use the heat transfer correlations from Chapter Six. Differences caused by using the new heat transfer correlations as opposed to the heat exchanger UA_{HX} adjustment are discussed.

7.1 Non-Auxiliary Tests

Table 7.1 lists additional outdoor tests with no auxiliary heating. The testing procedure involves filling the tank with isothermal water just before sunrise, performing load draws at selected times, and performing a complete (approximately three volume) energy draw at the end of the test. Test #1 has only a complete energy draw at the end of the test.

Test #	Length (Days)	# Draws	Weather Comments
1	1	1	Mostly Clear
2	1	2	Clear
3	3	3	Mostly Cloudy/Rain

Table 7.1 Additional Outdoor Tests Without Auxiliary

Figures 7.1 and 7.2 show the experimental draw and supply temperature profiles and the predicted draw temperature profile for Test #1. Table 7.2 shows the integrated energy results with experimental uncertainties for the same test. The predicted draw temperature profile is improved significantly, from an RMS error of 3.5 °C to 1.5 °C. From Table 7.2, the error in the total energy prediction when all of the energy is drawn out of the tank is improved after calibration, and is within the experimental uncertainty before calibration.

Figure 7.1 Draw and Supply Water Temperature Profiles For Test #1 Using Uncalibrated Model

Figure 7.2 Draw and Supply Water Temperature Profiles For Test #1 Using Calibrated Model

	Energy (kJ)	Percent Error (%)
Experiment	20460 $\pm$ 2300	0.0
Unadjusted Model	20930	+ 2.3
Calibrated Model	20580	+ 0.59

Table 7.2 Energy Results For Outdoor Test #1

Figures 7.3 through 7.6 show the draw and supply temperatures for Test #2 with a ten minute draw near solar noon and a complete energy draw at the end of the day. Table 7.3 shows integrated energy results for the test. The uncalibrated model overpredicts the temperature at the beginning of the short draw by more than 16 °C while the calibrated model predicts it to within 1 °C. The error in the energy drawn during the short draw is improved by forty-five percent. This large improvement is due to the large degree of

stratification predicted in the tank before calibration. The prediction for the long draw temperature profile is also improved significantly.

Figure 7.3 Draw and Supply Water Temperature Profiles For Short Draw of Test #2 Using Uncalibrated Model

Figure 7.4 Draw and Supply Water Temperature Profiles For Short Draw of Test #2 Using Calibrated Model

Figure 7.5 Draw and Supply Water Temperature Profiles For Long Draw of Test #2 Using Uncalibrated Model

Figure 7.6 Draw and Supply Water Temperature Profiles For Long Draw of Test #2 Using Calibrated Model

	Short Draw Energy (kJ)	Percent Error (%)	Long Draw Energy (kJ)	Percent Error (%)	Total Energy (kJ)	Percent Error (%)
Experiment	5664 ± 450	0.0	23350 ± 2600	0.0	29014	0.0
Unadjusted Model	9291	+ 64	22279	- 4.6	31570	+ 8.8
Calibrated Model	7032	+ 19	23258	- 0.40	30290	+ 4.4

Table 7.3 Energy Results For Outdoor Test #2

Figures 7.7 through 7.12 show the draw and supply temperatures for the three day test (#3) with three draws. The first draw was close to noon on the first day, the second was in the afternoon on the second day, and the final draw was during the evening of the

third day. The weather was often cloudy and it rained several times during the test. Table 7.4 shows the integrated energy results for the draws. Table 7.5 shows the RMS error between the temperature profiles for the draws. All of the temperature profiles are greatly improved and the integrated energy predictions are considerably better after calibration. The total draw energy after calibration is within the experimental uncertainty.

Figure 7.7 Draw and Supply Water Temperature Profiles For First Draw of Test #3 Using Uncalibrated Model

Figure 7.8 Draw and Supply Water Temperature Profiles For First Draw of Test #3 Using Calibrated Model

Figure 7.9 Draw and Supply Water Temperature Profiles For Second Draw of Test #3 Using Uncalibrated Model

Figure 7.10 Draw and Supply Water Temperature Profiles For Second Draw of Test #3 Using Calibrated Model

Figure 7.11 Draw and Supply Water Temperature Profiles For Final Draw of Test #3 Using Uncalibrated Model

Figure 7.12 Draw and Supply Water Temperature Profiles For Final Draw of Test #3 Using Calibrated Model

	Experiment	Unadjusted Model	Calibrated Model
Draw #1	775 ± 140	2445	1025
% Error	0.0	+ 215	+ 32
Draw #2	5930 ± 407	7328	5909
% Error	0.0	+ 24	- 0.35
Draw #3	24190 ± 2870	25357	25506
% Error	0.0	+ 4.8	+ 5.4
Total Energy (kJ)	30895	35130	32440
% Error	0.0	+ 13.7	+ 5.0

Table 7.4 Energy Results For Outdoor Test #3

	Unadjusted Model RMS Error (°C)	Calibrated Model RMS Error (°C)
Draw #1	9.2	0.43
Draw #2	8.0	1.8
Draw #3	3.5	1.8

Table 7.5 RMS Error Between Draw Temperature Profiles For Outdoor Test #3

7.2 Auxiliary Tests

Table 7.6 lists additional outdoor tests with auxiliary heating. The testing procedure involves filling the tank with isothermal water in the evening, switching on the auxiliary element so the preheat portion of the tank will arrive at the selected setpoint temperature (55 °C) before sunrise, performing load draws at selected times, and performing a complete (approximately three volume) energy draw at the end of the test. Simulation is performed using a modified model to initialize the tank to a uniform temperature so that initial auxiliary heating may be simulated.

Test #	Length (Days)	# Draws	Weather Comments
4	1	4	Partly Cloudy
5	1	3	Partly Cloudy
6	3	9	Day 1: Sunny Day 2: Partly Cloudy Day 3: Mostly Clear

Table 7.6 Additional Outdoor Tests With Auxiliary

Figures 7.13 through 7.16 show the second and fourth (final) draw and supply temperature profiles for Test #4. Table 7.7 lists integrated energy results for the test. Improvements in temperature profile prediction during this test are not as significant as in the non-auxiliary tests. The slope of the temperature profile for the second draw is

improved significantly (Figure 7.14), but the energy prediction is only improved slightly. The overall error in the predicted draw energy is decreased from seventeen percent to about seven percent and the error in the integrated auxiliary energy input is decreased from twenty-nine to about three percent. The calibrated model significantly improves total energy values.

Figure 7.13 Draw and Supply Water Temperature Profiles For Second Draw of Test #4 Using Uncalibrated Model

Figure 7.14 Draw and Supply Water Temperature Profiles For Second Draw of Test #4 Using Calibrated Model

Figure 7.15 Draw and Supply Water Temperature Profiles For Final Draw of Test #4 Using Uncalibrated Model

Figure 7.16 Draw and Supply Water Temperature Profiles For Final Draw of Test #4 Using Calibrated Model

	Experiment	Unadjusted Model	Calibrated Model
Draw #1	10550 ± 645	8824	8237
% Error	0.0	- 16	- 22
Draw #2	12250 ± 743	12776	12723
% Error	0.0	+ 4.3	+ 3.9
Draw #3	9620 ± 562	8800	9160
% Error	0.0	- 8.5	- 4.8
Draw #4	28595 ± 3092	20130	26480
% Error	0.0	- 30	- 7.4
Auxiliary	50120 ± 4500	35840	48520
% Error	0.0	- 29	- 3.2
Total Draw Energy (kJ)	61015	50650	56600
% Error	0.0	- 17	- 7.2

Table 7.7 Energy Results For Outdoor Test #4

Figures 7.17 and 7.18 show the draw and supply temperature profiles for the final draw of Test #5. Table 7.8 shows integrated energy results for the test. The draw temperature profile for the first draw (not shown) is worsened to a slight degree and the draw temperature profile for the second draw (not shown) is improved to a slight degree. The RMS error in the profile for the final draw shown in Figures 7.17 and 7.18 is increased from 3.7 °C to 4.0 °C, even though the error in the prediction of the draw energy is improved from twenty-three percent to twelve percent. The prediction of the total auxiliary energy input to the tank is improved to within experimental uncertainty.

Figure 7.17 Draw and Supply Water Temperature Profiles For Final Draw of Test #5 Using Uncalibrated Model

Figure 7.18 Draw and Supply Water Temperature Profiles For Final Draw of Test #5 Using Calibrated Model

	Experiment	Unadjusted Model	Calibrated Model
Draw #1	10630 ± 645	8920	8165
% Error	0.0	- 16	- 23
Draw #2	12202 ± 715	11690	11925
% Error	0.0	- 4.2	- 2.3
Draw #3	32178 ± 2980	24740	28310
% Error	0.0	- 23	- 12
Auxiliary	46410 ± 4160	33380	42610
% Error	0.0	- 28	- 8.2
Total Draw Energy (kJ)	55010	45350	48400
% Error	0.0	- 18	- 12

Table 7.8 Energy Results For Outdoor Test #5

Figures 7.19 through 7.22 show the draw and supply temperature profiles for the second and final draws of Test #6. Table 7.9 compares energy values during the test.

Figures 7.19 and 7.20 show that for the relatively small volume second draw, the slope of the draw temperature profile is improved after calibration, even though from Table 7.9 the draw energy value is not significantly changed. In Figures 7.21 and 7.22, the draw temperature profile is not visibly improved near the beginning of the draw, although the integrated energy for the draw is improved significantly. The prediction in the integrated auxiliary energy is again greatly improved.

Figure 7.19 Draw and Supply Water Temperature Profiles For Second Draw of Test #6 Using Uncalibrated Model

Figure 7.20 Draw and Supply Water Temperature Profiles For Second Draw of Test #6 Using Calibrated Model

Figure 7.21 Draw and Supply Water Temperature Profiles For Final Draw of Test #6 Using Uncalibrated Model

Figure 7.22 Draw and Supply Water Temperature Profiles For Final Draw of Test #6 Using Calibrated Model

7.3 Alternate Use of UA_{HX} Adjustment

Use of the UA_{HX} adjustment calculated in Chapter Five rather than the new heat transfer correlations changes the above results to a small degree. Table 7.10 shows comparisons of several energy values. The error in three of the energy values in the table increases and the error in one decreases when the UA_{HX} adjustment is used. None of the changes are large.

	Experiment	Unadjusted Model	Calibrated Model
Draw #1	13207 ± 776	10540	9706
% Error	0.0	- 20	- 26
Draw #2	12591 ± 734	12850	12894
% Error	0.0	+ 2.1	+ 2.4
Draw #3	13776 ± 770	14490	14390
% Error	0.0	+ 5.2	+ 4.4
Draw #4	10066 ± 626	8600	8570
% Error	0.0	- 15	- 15
Draw #5	18333 ± 1103	18450	19330
% Error	0.0	+ 0.64	+ 5.4
Draw #6	6214 ± 417	4990	4610
% Error	0.0	- 20	- 26
Draw #7	10381 ± 624	10850	10980
% Error	0.0	+ 4.5	+ 5.8
Draw #8	11331 ± 679	12050	11400
% Error	0.0	+ 6.3	+ 0.61
Draw #9	29535 ± 3005	22980	26220
% Error	0.0	- 23	- 12
Auxiliary	88960 ± 7980	63880	83280
% Error	0.0	- 28	- 6.4
Total Draw Energy (kJ)	125434	115800	118100
% Error	0.0	- 7.7	- 5.8

Table 7.9 Energy Results For Outdoor Test #6

Test # and Draw #	Using New HT Correlations		Using UA _{HX} Adjustment	
	% Energy Error (kJ)	RMS °C Error	% Energy Error (kJ)	RMS °C Error
Test #3 Draw #2	+ 19	2.8	+ 24	6.7
Test #4 Draw #3	+ 5.4	1.8	+ 6.2	1.7
Test #6 Draw #2	- 4.2	2.4	+ 1.2	1.9
Test #7 - Auxiliary Energy	- 6.4	N/A	- 9.3	N/A

Table 7.10 Comparison of Heat Exchanger Calibration Methods

CHAPTER EIGHT

CONCLUSIONS AND RECOMMENDATIONS

This chapter summarizes significant conclusions and recommends areas for further research.

8.1 Conclusions

The thermosyphon system model in TRNSYS Version 14.1 does not effectively simulate the short-term performance of the SDHW system under investigation. Collector flow rate is underpredicted, heat transfer in the heat exchanger is overpredicted, and tank stratification is overpredicted. These errors result in relatively large discrepancies in temperature profile predictions and integrated energy values in the collector loop and during load draws. Auxiliary energy input is underestimated to a large degree.

Several weaknesses in the TRNSYS model related to these deficiencies have been identified:

- In the closed collector loop, the pressure drop is predicted incorrectly. This causes an underprediction of the collector loop flow rate and an overprediction of the collector temperature rise. This error may be explained by the fact that the pressure drop in the collector risers is the largest component of the head loss and the value for the hydraulic diameter of the risers is largely uncertain due to the construction of the collector.

- The heat exchanger UA product is overestimated in most situations due to heat transfer correlations that overpredict inside and outside convection coefficients.
- Tank stratification overprediction is likely due to a convective destratification mechanism that is not currently simulated. During experiments with auxiliary heating, significant heating of the lower solar preheat portion of the storage tank occurs. This is not predicted in the model and results in significantly underestimated auxiliary energy input.

A systematic calibration of the model, including improved heat transfer correlations in the heat exchanger, has been developed based on experimental data. Model performance improves in most respects after calibration:

- In almost all cases, the RMS error between experimental and predicted temperature profiles decreases.
- With the exception of short draws near the beginning of tests with the auxiliary element enabled, the accuracy of energy predictions for short draws increases dramatically. Errors often are reduced by more than forty percent. This is due to more accurate prediction of the degree of thermal stratification of tank water after calibration.
- Results vary when all of the energy is drawn out of the tank in a multiple volume draw. With no auxiliary heating, the integrated energy prediction is usually not changed significantly after model calibration. When auxiliary heating is used, the accuracy of the prediction improves significantly.

- In all tests, the accuracy of the predicted total energy drawn from the system improves after model calibration.
- Auxiliary energy input is always underpredicted by the uncalibrated model. Calibration improves the prediction to a large degree in all tests.

8.2 Recommendations

Based on this work, several recommendations can be made:

- Use of current TRNSYS models in the rating of thermosyphon SDHW systems requires experimental data for calibration.
- Improved heat transfer correlations for this particular tank/heat exchanger combination are required. Detailed temperature measurements in several locations in the heat exchanger and flow visualization would facilitate this study.
- Destratification mechanisms in the tank need further study. Flow visualization would allow direct examination of convective destratification mechanisms.
- Closed circuit pressure drop measurements with varying flow rates would determine the accuracy of the model's pressure drop calculation and help identify possible sources of error.

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APPENDIX A

CALIBRATION OF TURBINE FLOW METER

The EG&G Technologies turbine flow meter operates by using a sensor that counts pulses caused by a vane inserted into the flow. These pulses are converted by an EG&G Pulse Rate Converter to a DC voltage output. Calibration is performed using a calibrated venturi meter connected to graduated manometer tubes. Height differences from manometer readings are converted to volumetric flow rates which are calibrated against voltages read by the data acquisition system. Uncertainty in reading the manometer scales is estimated to be significantly less than one percent. Figure A.1 shows the experimental data points and the line

$$\left[\frac{l}{s} \right] = 0.1602[Volts] - 6.346 \cdot 10^{-4}. \quad (A.1)$$

The square of the correlation coefficient (R^2) for the fitted line is 0.99996 which is essentially one.

Figure A.1 Turbine Flow Meter Calibration

APPENDIX B

EXPERIMENTAL UNCERTAINTY ANALYSIS

B.1 Measurement Uncertainties

Uncertainties in individual experimental measurements are obtained from manufacturers specifications and/or calibration results. Table B.1 lists experimental uncertainties. Thermocouple and thermistor measurements use reference junctions and compensations internal to the hardware and software. The relatively large uncertainty in the heat exchanger flow rate read by the Coriolis meter is due to an offset error that was corrected after the data had been collected.

Measurement	Sensor	Uncertainty
Temperature	Type-T Thermocouple	$\pm 0.5^{\circ}\text{C}$
Temperature	2252 Ω Thermistor Probe	$\pm 0.1^{\circ}\text{C}$
Water Flow Rate	Turbine Flow Meter	$\pm 0.01\text{ l/s}$
Heat Exchanger Flow Rate	Coriolis Meter	$\pm (1\text{ kg/hr} + 0.2\%)$
Auxiliary Power	AEMC Clamp-On RMS Power Transducer (M700)	$1717 \pm 154\text{ W}$
Insolation	Eppley PSP Pyranometer	$\pm 1\%$

Table B.1 Experimental Measurement Uncertainties

B.2 Calculated Quantity Uncertainties

Calculated quantities include temperature differences and energy values. Temperature differences are calculated between two thermocouple or two thermistor measurements as

$$\Delta T = T_1 - T_2 \quad (\text{B.1})$$

for any two temperatures. The combined uncertainty is expressed as

$$U_{\Delta T} = \sqrt{\left(U_{T_1} \frac{\partial \Delta T}{\partial T_1}\right)^2 + \left(U_{T_2} \frac{\partial \Delta T}{\partial T_2}\right)^2} \quad (\text{B.2})$$

where U denotes the absolute uncertainty in the subscripted quantity. However, both of the partial derivatives in eqn. B.2 reduce to an absolute value of one so that the uncertainty becomes

$$U_{\Delta T} = \sqrt{U_{T_1}^2 + U_{T_2}^2} \quad (\text{B.3})$$

or

$$U_{\Delta T} = \sqrt{2U_T^2} \quad (\text{B.4})$$

if both temperature measurements are made with the same type of device. This results in an uncertainty of 0.7 °C for thermocouple measurements and 0.14 °C for thermistor measurements.

Energy values for collector and load flows are calculated as

$$Q = \dot{m} C_p \Delta T \Delta t \quad (\text{B.5})$$

where ΔT denotes a temperature difference and Δt denotes the timestep. Uncertainties in the timestep and specific heat (C_p) are considered negligible compared to uncertainties in flow rate and temperature difference, thus the overall uncertainty can be represented as

$$U_Q = \sqrt{\left(U_{\dot{m}} \frac{\partial Q}{\partial \dot{m}} \right)^2 + \left(U_{\Delta T} \frac{\partial Q}{\partial \Delta T} \right)^2} \quad (\text{B.4})$$

Taking the partial derivatives and simplifying, the uncertainty in any energy calculation is expressed as

$$U_Q = C_p \Delta t \sqrt{(U_{\dot{m}} \Delta T)^2 + (U_{\Delta T} \dot{m})^2} \quad (\text{B.5})$$

Auxiliary energy values are calculated as

$$Q_{AUX} = P_{AUX} \Delta t \quad (\text{B.6})$$

where P_{AUX} is the experimentally measured auxiliary power input. Assuming negligible uncertainty in the timestep and simplifying the resulting expression, the uncertainty in the auxiliary energy input becomes simply

$$U_{Q_{AUX}} = U_{P_{AUX}} \Delta t \quad (\text{B.7})$$

APPENDIX C

SAMPLE TRNSYS INPUT DECKS

C.1 Indoor TRNSYS Deck

```
ASSIGN AUX6B.LST 6
ASSIGN AUX6B.OUT 10
ASSIGN AUX6B.DAT 15
*
*   This deck uses a modified TRNSYS that initializes top
*   and bottom of tank to same temperature.
*
*   *****
*   **** Solahart 180JK Thermosyphon System ****
*   *****
*
*   Inside warm-up with 45 Min. draw following.
*   Includes Aux. Heating.
*
*   Deck by: Todd Swift
*   Date: 7/10/96
*
*   ***** Simulation Parameters *****
EQUATIONS 6
START = 0
STOP = 8.73
STEP = 1/60
HXUAKN = 1
KAKNOB = 1
FFKNOB = 1
*
*   *****Collector Parameters*****
EQUATIONS 4
A = 0.846
*   Efficiency Intercept (Fav(TA))
B = 6.5 * 3.6
*   (Tw - Ta + R) Coefficient [kJ/hr-m2-K]
C = 0.0 * 3.6
*   (Tw - Ta + R)^2 Coefficient [kJ/hr-m2-K]
R = 0
*   Sky Temperature Allowance [K]
*
*   *****Thermosyphon Unit Parameters*****
EQUATIONS 25
AREAPAN = 1.860
*   Single panel net area [m2]
NPANEL = 1.0
*   Total number of panels
AREA = AREAPAN*NPANEL
*   Total Area
SLOPE = 45
*   Test Stand Slope [deg]
RHOG = .000001
*   Ground Reflectance
RHOTNK = 1.000000E+03
*   Collector fluid density [Kg/m3]
CPTNK = 4.186728E+00
```



```

* Collector Fluid Specific Heat [KJ/kg-k]
NR = 36
* Number of parallel collector risers
DR = 4.25E-03
* Riser Diameter [m]
LR = 1.843
* Riser Length [m]
DH = 2.412882E-02
* Header Diameter [m]
LH = 0.990
* Header Length [m]
HC = 1.320
* HC from Figure [m]
HCOLD = 0.03
* Ht. of Cold In Above Tank Bottom [m]
HO = 1.405
* HO from Figure [m]
HR = 0.000
* HR from Figure [m]
NX = 10
* Nodes for Thermal Head Calculations
DI = 0.0169
DO = DI
* Diameter of Collector Inlet and Outlet Pipes [m]
LI = 2.87
* Length of Thermosyphon Inlet Pipe [m]
NB1 = 7.0
* # of Right Angle Bends in Inlet Pipe
NB2 = 2.0
* # of Right Angle Bends in Outlet Pipe
UI = 17.6
UO=UI
* Loss Coefficient, Collector Pipes [kJ/h-m2-K]
LO = 0.42
* Length of Thermosyphon Outlet Pipe [m]
*
*          *****Solar Storage Tank*****
EQUATIONS 15
VOLSOL = 0.1779
* Solar tank volume [m3]
HT = 0.4259
* Inner tank diameter [m]
UASOL = 10.944
* Solar tank UA-value [kJ/hr-C]
RI = 1
* Insulation Ratio
VMAX = 0.1
* Max Size of Tank Element During Conduction Analysis (Fraction)
TMAX = 95
* Dump Valve Operating Temp [C]
TMIN = 94
* Dump Valve Closing Temp [C]
DT = 0.436
* Outer Tank Diameter [m]
LT = 1.31
* Outer Tank Length [m]
RB = 7.2e-05
* Cond. Thermal Resistance of Tank Wall / Length [K-h-m/kJ]
TWKW = 0.4465
* Tank Thk * Wall Conductivity [kJ/h-K]
*
*          *****Auxiliary Heater*****
HTH = 0.23
* Height of thermostat above bottom of tank [m]
AUX = 6181
* Maximum auxiliary heating rate [kJ/hr]
HHEAT = 0.223
* Height of Aux. heater above bottom of tank [m]
TDBTNK = 5
* Aux. heater deadband [K]
*
*          *****Experimental Variables*****
EQUATIONS 11
TI = 19.79
* Initial Tank Temperature (Avg) [C]

```

```

TAMB = 22.93
* Average ambient Temperature during test [C]
TSET = 55
* Set Point Temperature [C]
DUMMY = 0.0
* Value for Dummy Parameters
IREV = 0.0
* Reverse Flow Analysis Option - no rev. flow analysis
TT = 200
* Temp for Flow Restriction (none for TT > 100 C)
IT = 10
* Incident Radiation on Collector Aperture [kJ/hr m^2]
IH = 0
* Horizontal Total Radiation
ID = 0
* Diffuse Radiation
MDOT = 0.200 * 3600
* Av. Draw Flow Rate [kg/h] - adjusted to correct volume
THETA = 0
* Angle of incidence on collector
*
* *****
* SIMULATION START
* *****
*
SIMULATION START STOP STEP
LIMITS 25 5 25
TOLERANCES -0.01 -0.01
*
UNIT 16 TYPE 9 Data Reader
PARAMETERS 29
2 8 STEP -1 1 0 -2 1 0 -3 1 0 -4 1 0 -5 1 0 -6 1 0 -7 1 0 -8 1 0 15 -1
* Time; Flow; GI(tm); GI(tc); GO(tm); GO(tc); TDrw; Tsup
*
UNIT 14 TYPE 14 Draw Profile [kg/h]
PARAMETERS 10
0,0 7.99,0 7.99,MDOT 8.75,MDOT 8.75,0
*
UNIT 15 TYPE 14 AUXILIARY ELEMENT ENABLE
PARAMETERS 8
0,1 8,1 8,0 8.75,0
*
UNIT 45 TYPE 45 G.M. Thermosyphon
PARAMETERS 58
*
*Eff Mode; Ac; Eff; 1st Coeff; 2nd Coeff; Sky Temp.; Slope; LU; # Risers; DR; LR
*DH; LH; NX; Hc; Ho; DI; LI; NBI; UI; DO; LO; NB2; UO; Tank Mode; VOLSOL
*HT; Hr; CPTNK; RHOTNK; twkw; Tank Config; UASOL; RI; TI; HCOLD; AUX
*HHEAT; HTH; TSET; TDBTNK; UAF; VMAX; TMAX; TMIN; DT; LT; RB; Dummy
*Dummy; Optical Mode; Bo; IREV; TT; FLUID (1 for pg); HXUAKnob; KAKnob
*
2 AREA A B C R SLOPE -2 NR DR
LR DH LH NX HC HO DI LI NBI UI
DO LO NB2 UO 6 VOLSOL HT HR CPTNK RHOTNK
TWKW 2 UASOL RI TI HCOLD AUX HHEAT HTH TSET
TDBTNK 0 VMAX TMAX TMIN DT LT RB DUMMY DUMMY
1 0 IREV TT 1 HXUAKN KAKNOB FFKNOB
*
INPUTS 12
*
*It; Ih; Id; Theta; Rhog; Tamb; Tmains; Load Flow; Tenv; Aux Enable; THXIN; mdot
*
0,0 0,0 0,0 0,0 0,0 0,0 16,8 14,1 0,0 15,1 16,3 16,2
IT IH ID THETA RHOG TAMB 0.0 0.0 TAMB 1.0 0.0 0.0
*
* Output
*
UNIT 24 TYPE 24 Integrator
INPUTS 1
45,6
0.0
* Mass(kg)
*
UNIT 25 TYPE 25 Printer
PARAMETERS 4

```

```

STEP 0 STOP 10
INPUTS 8
45,3 45,4 45,1 45,5 16,7 24,1 16,5 45,10
THXOUT MDOT THXIN TDRW TDRWEX VOL TOUTEXP QAUX
*
END

```

C.2 Outdoor TRNSYS Deck

```

ASSIGN OUT6.LST 6
ASSIGN OUT6.OUT 10
ASSIGN OUT6.DAT 11
*
* *****
* **** Solahart 180JK Thermosyphon System ****
* *****
*
* Outdoor auxiliary warm-up with several draws.
*
* Deck by: Todd Swift
* Date: 7/10/96
*
* ***** Simulation Parameters *****
EQUATIONS 7
START = 22.0 - 1
STOP = 46.07 - 1
STEP = 1/60
DAY = 193
HXUAKN = 0.63
KAKNOB = 13
FFKNOB = 0.41
* Adjusted is (0.63,13,0.41)
*
* *****Collector Parameters*****
EQUATIONS 5
A = 0.846
* Efficiency Intercept (Fav(TA))
B = 6.5 * 3.6
* (Tw - Ta + R) Coefficient [kJ/hr-m2-K]
C = 0.0 * 3.6
* (Tw - Ta + R)^2 Coefficient [kJ/hr-m2-K]
R = 0
* Sky Temperature Allowance [K]
IAM = 0.1
* Incident Angle Modifier
*
* *****Thermosyphon Unit Parameters*****
EQUATIONS 25
AREAPAN = 1.860*0.91
* Single panel net area [m2]
NPANEL = 1.0
* Total number of panels
AREA = AREAPAN*NPANEL
* Total Area
SLOPE = 45
* Test Stand Slope [deg]
RHOG = 0.5
* Ground Reflectance
RHOTNK = 1.0000E+03
* Tank fluid density [Kg/m3]
CPTNK = 4.186728
* Tank Fluid Specific Heat [KJ/kg-k]
NR = 36
* Number of parallel collector risers
DR = 4.25E-03
* Riser Diameter [m]
LR = 1.843
* Riser Length [m]
DH = 2.412882E-02
* Header Diameter [m]
LH = 0.990
* Header Length [m]

```

```

HC = 1.320
* HC from Figure [m]
HCOLD = 0.03
* Ht. of Cold In Above Tank Bottom [m]
HO = 1.405
* HO from Figure [m]
HR = 0.040
* HR from Figure [m]
NX = 10
* Nodes for Thermal Head Calculations
DI = 0.0169
DO = DI
* Diameter of Collector Inlet and Outlet Pipes [m]
LI = 2.21
* Length of Thermosyphon Inlet Pipe [m]
NB1 = 2.0
* # of Right Angle Bends in Inlet Pipe
NB2 = 2.0
* # of Right Angle Bends in Outlet Pipe
UI = 17.6
UC=UI
* Loss Coefficient, Collector Pipes [kJ/h-m2-K]
LO = 0.42
* Length of Thermosyphon Outlet Pipe [m]
*
* *****Solar Storage Tank*****
EQUATIONS 15
VOLSOL = 0.1779
* Solar tank volume [m3]
HT = 0.4259
* Inner tank diameter [m]
UASOL = 10.944
* Solar tank UA-value [kJ/hr-C]
RI = 1
* Insulation Ratio
VMAX = 0.1
* Max Size of Tank Element During Conduction Analysis (Fraction)
TMAX = 95
* Dump Valve Operating Temp [C]
TMIN = 94
* Dump Valve Closing Temp [C]
DT = 0.436
* Outer Tank Diameter [m]
LT = 1.31
* Outer Tank Length [m]
RB = 7.2e-5
* Cond. Thermal Resistance of Tank Wall / Length [K-h-m/kJ]
TWKW = 0.4465
* Tank Thk * Wall Conductivity [kJ/h-K] - Todd (Carl had 0.4465)
*
* *****Auxiliary Heater*****
HTH = 0.23
* Height of thermostat above bottom of tank [m]
AUX = 6181
* Maximum auxiliary heating rate [kJ/hr]
HHEAT = 0.223
* Height of Aux. heater above bottom of tank [m]
TDBTNK = 5
* Aux. heater deadband [K]
*
* *****Experimental Variables*****
EQUATIONS 9
TI = 14.35
* Initial Temperature of Preheat portion of tank (Avg) [C]
TSET = 55
* Set Point Temperature [C]
DUMMY = 0.0
* Value for Dummy Parameters
IREV = 0.0
* Reverse Flow Analysis Option - no rev. flow analysis
TT = 200
* Temp for Flow Restriction (none for TT > 100 C)
MDOT1 = 0.185 * 3600
MDOT2 = 0.180 * 3600
MDOT3 = 0.190 * 3600

```

```

MDOT4 = 0.1929 * 3600
* Draw #1- #4 Flow Rate [kg/h]
*
* *****
* SIMULATION START
* *****
*
SIMULATION START STOP STEP
LIMITS 25 5 25
TOLERANCES -0.01 -0.01
*
Unit 9 Type 9 Input Data Reader
Parameters 29
2 8 1.66667E-2 -1 1 0 -2 1 0 -3 1 0 -4 1 0 -5 1 0 -6 1 0
7 3600 0 -8 3.6 0 11 -1
* Time Amb ColIn ColOut Tsup Tdrw mdot Rad
*
Unit 16 Type 16 Solar Processor
Parameters 8
4 1 3 DAY 40 4871 0 2
Inputs 6
9,8 9,19 9,20 0,0 0,0 0,0
0.0 0.0 0.0 RHOG 45 0
*
UNIT 15 TYPE 14 Auxiliary Enable
PARAMETERS 8
START,1 43,1 43,0 STOP,0
*
UNIT 14 TYPE 14 Draw Profile [kg/h]
PARAMETERS 36
START,0 31.19,0 31.19,MDOT1 31.27,MDOT1 31.27,0 36.81,0
36.81,MDOT2 36.91,MDOT2 36.91,0 41.15,0 41.15,MDOT3
41.23,MDOT3 41.23,0 43.98,0 43.98,MDOT4 45.06,MDOT4
45.06,0 STOP,0
*
UNIT 45 TYPE 45 G.M. Thermosyphon
PARAMETERS 58
*
*Eff Mode; Ac; Eff; 1st Coeff; 2nd Coeff; Sky Temp.; Slope; LU; # Risers; DR; LR
*DH; LH; NX; Hc; Ho; DI; LI; NB1; UI; DO; LO; NB2; UO; Tank Mode; VOLSOL
*HT; Hr; CPTNK; RHOTNK; twkw; Tank Config; UASOL; RI; TI; HCOLD; AUX
*HHEAT; HTH; TSET; TDBTNK; UAF; VMAX; TMAX; TMIN; DT; LT; RB; Dummy
*Dummy; Optical Mode; Bo; IREV; TT; FLUID (1 for pg); hxuaknob; KAKnob; ffknob
*
2 AREA A B C R SLOPE -2 NR DR
LR DH LH NX HC HO DI LI NB1 UI
DO LO NB2 UO 6 VOLSOL HT HR CPTNK RHOTNK
TWKW 2 UASOL RI TI HCOLD AUX HHEAT HTH TSET
TDBTNK 0 VMAX TMAX TMIN DT LT RB DUMMY DUMMY
1 IAM IREV TT 1 HXUAKN KAKNOB FFKNOB
*
INPUTS 10
*
*It; Ih; Id; Theta; Rhog; Tamb; Tmains; Load Flow; Tenv; Aux Enable
*
16,6 16,4 16,5 16,9 0,0 9,2 9,5 14,1 9,2 15,1
0.0 0.0 0.0 0.0 RHOG 0.0 0.0 0.0 0.0 1.0
*
* Output
*
UNIT 24 TYPE 24 Integrator
INPUTS 3
45,6 45,8 45,10
0.0 0.0 0.0
* Mass(kg) Draw(kJ) Qaux(kJ)
*
UNIT 25 TYPE 25 Printer
PARAMETERS 4
STEP START STOP 10
INPUTS 10
45,1 45,3 9,4 9,3 45,5 9,6 9,5 24,1 24,2 24,3
TOTRN TITRN TOEXP TIEXP Tdrw TDRWEXP TSUP VOL ENERGY QAUX
END

```