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**COLORADO STATE UNIVERSITY PROGRAM FOR
DEVELOPING, TESTING, EVALUATING
AND OPTIMIZING
SOLAR HEATING SYSTEMS**

**PROJECT STATUS REPORT FOR THE MONTHS OF
DECEMBER 1995 AND JANUARY 1996**

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MASTER

RATING AND CERTIFICATION OF DOMESTIC WATER HEATING SYSTEMS

December/January 1996

Current work involves experimental indoor testing of the tank/heat exchanger portion of the system under investigation. A straightforward and relatively simple method for calibration of the computer model used to simulate thermosyphoning solar domestic hot water systems is outlined below. The indoor experimental set-up used for testing is then described. The goal of this work is to simplify the rating and certification of such systems while at the same time significantly improve the accuracy of the rating.

PROPOSED MODEL CALIBRATION METHOD

Background

The complexity of thermosyphoning solar domestic hot water systems makes them difficult to accurately model. Models that are used must often be adjusted in some fashion to match experimental data. Previous model adjustments have been too simplistic and confidence in the resulting adjusted model's accuracy is questionable. This work seeks to improve the accuracy of the simulation by systematically identifying and adjusting key parameter inputs and requiring that the model output better match the time evolution of selected temperatures and flow rates. (See the November/December 1995 Progress Report for further discussion)

Parameter Adjustments Required

Previous work [November/December 1995 Progress Report] identified three key parameters in the simulation that need to be adjusted. They are: the heat exchanger overall conductance (UA), the equivalent conductivity term between adjacent water layers and along the tank wall, and the total friction factor in the collector fluid loop.

To adjust these accurately, measurement of the collector fluid flow rate is required (along with several temperature measurements). Measurement of the closed loop flow rate is difficult because the total pressure drop in the loop is very low (on the order of 40 Pascals). Standard turbine flow meters cannot be used due to their relatively large pressure drops. This makes accurate measurement of the flow rate complicated or expensive. By forcing the flow rate in the collector loop and simulating the collector with an indoor heating apparatus, the low pressure drop requirement for the flow meter may be eliminated, simplifying the calibration process.

Proposed Calibration Method

The heat exchanger overall conductance and equivalent conductivity term adjustments may be obtained from a relatively simple indoor test. Following this, the total friction factor in the collector loop may be adjusted with a short outdoor test.

Use of a heating apparatus in an indoor test to simulate the collector both simplifies the calibration and also may improve the accuracy by eliminating the coupling between the collector and tank/heat exchanger. To obtain the overall heat exchanger conductance adjustment, the heating apparatus is used to provide a constant heat flux to the collector fluid and a constant flow rate through the system for a period of time. Glycol inlet and exit temperatures from the heat exchanger are recorded. The computer model is then forced to follow the experimental flow rate and inlet temperature profile and the heat exchanger conductance is modified to match the glycol exit temperature as closely as possible. Following this, temperatures from a hot water draw made on the heated system are used to adjust the equivalent conductivity term by matching the draw temperature profile as closely as possible.

Initial indoor tests will begin with uniform temperature water in the tank and will consist of a four hour constant heat flux warm-up period followed by a complete (three volume) energy draw of water out of the tank at a flow rate of approximately 0.2 liters per second. Changes to the testing conditions will be made as necessary, including the addition of an auxiliary heating element in the tank.

The outdoor test will consist of installing the entire system outside as if for normal operation and measuring the collector inlet and outlet temperatures along with appropriate water temperatures and solar radiation. Since the model accurately predicts the collector useful energy [November/December 1995 Progress Report], the collector flow rate can be inferred from the measured temperatures. All that remains is to force the model with the measured collector inlet temperature profile and radiation data and adjust the total friction factor in the collector loop until the predicted collector outlet temperature profile matches the experimental profile as closely as possible. When this is complete, the predicted and experimental collector flow rate and temperature rise profiles should coincide.

This calibration procedure is straightforward and should be able to be completed in a relatively short period of time at any testing facility with the capability to simulate a collector. Improved confidence in model predictions should result.

INDOOR EXPERIMENTAL ARRANGEMENT

System Instrumentation

The tank/heat exchanger portion of the system has been moved inside. Instrumentation includes a turbine flow meter to measure draw flow and four thermistors to measure water temperature in and out of the tank and glycol temperature in and out of the heat exchanger. In addition, two special-limit T-type thermocouples have been attached to the outside of the pipes entering and leaving the heat exchanger using thermocouple epoxy. The temperatures from these thermocouples will be used with the temperatures from the corresponding thermistors to investigate the accuracy of using non-invasive temperature measurements for the closed glycol loop.

All connecting piping has been insulated and the system is attached to a two-tank water conditioning system so that water temperature entering the tank can be controlled. The closed loop glycol side of the system is attached to the boiler apparatus described below.

Boiler Apparatus

The boiler apparatus used to simulate the collector consists of a circulator pump, a direct-reading in-line flow meter, a low pressure drop Coriolis mass flow meter, and a hydronic heater with two 4500 Watt heating elements all mounted on a semi-portable uni-strut frame. Data measurements include mass flow rate in the loop, power input to the heating elements, and temperatures into and out of the heating block. Control of the system is performed either manually or by computer. Computer control allows automatic, timed switching of the pump and heating elements and control of the water temperature and/or heating rate via an electronic controller.

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

UNIQUE SOLAR SYSTEM COMPONENTS

INTEGRATED TANK/HEAT EXCHANGER MODELING/EXPERIMENTS

Work continues on the development of an improved wrap-around heat exchanger/tank model for use in TRNSYS 14. Current efforts have focused on developing a two layer model of the natural convection in the tank similar to that introduced by Drake (1966, and Evans et. al., 1968). In this approach, the natural convection boundary layer on the wall of the tank is modeled in detail. The results of the boundary layer calculations are then coupled to a more simple control volume model of the core region of the tank. This approach was used successfully to model natural convection processes in large oil filled underground caverns of the Strategic Petroleum Reserve (SPR) (Webb, 1988).

The most recent version of the "old" CSU wrap-around heat exchanger model using the new version of the TRNSYS Type 4 Thermal Storage Tank Model developed by B. J. Newton (1995) at the University of Wisconsin for TRNSYS 14 has not yet been tested. Final documentation of all experimental results obtained to date is waiting on the completion of the new version of the wrap-around tank model.

References

- Drake, E. M., "Transient Natural Convection of Fluids Within Vertical Cylinders," Sc.D. Thesis, MIT, Cambridge, MA, 1966.
- Evans, L. B., R. C. Reid, and E. M. Drake., "Transient Natural Convection in a Vertical Cylinder," *A.I.Ch.E. Journal*, Vol. 14, No. 2, pp. 251-259, March, 1968.
- Webb, S. W., *Development and Validation of SPR Cavern Fluid Velocity Model*, Sandia National Laboratories Report SAND88-2711, 1988.

ADVANCED RESIDENTIAL SOLAR DOMESTIC HOT WATER (SDHW) SYSTEMS

This report is for December, 1995 and January, 1996. Experimental tests are being conducted on three side by side systems: ASN, NEG and Thermodynamics.

Experimental Work

Work continues on analyzing the data collected from the 5 day normal operation tests, as well as the continuous operation tests. As of yet, no other experimental problems have been discovered (like the missing "dip-tube" described in the last report).

A "dip-tube" was installed on the Thermo Dynamics system. A tube for a standard 40 gallon tank was bought and altered to fit the larger 60 gallon solar storage tank. It seems to be working with no leaks. During a day of good weather, an attempt was made to repeat a test for the Thermo Dynamics system, but it was discovered that the draw flow meter was no longer working, the electronics were operating sporadically. The problem was not located, so we will rely on the supply flow meter for the flow rate measurement (since a factory repair would take six weeks). Three tests have been completed during the stretches of sunny weather. These tests were performed without the Micro Motion flow meter (an extremely low pressure drop meter), used to measure the thermosyphon flow rate. The Micro Motion now needs to be re-installed. The tests are performed with and without the Micro Motion flow meter so that we can have one set of measurements with a measured natural convection flow rate to compare against the other set (to verify the procedure for calculating the flow rate). It is currently being used on a different experimental set-up, it will be installed as soon as the other person's test is completed.

A paper titled "Analysis of a Load-Side Heat Exchanger for a Solar Domestic Hot Water Heating System," was submitted to the ASME transactions Journal of Solar Energy Engineering for review and possible publication. A copy of the paper is attached.

Work continues towards completion of Toni Smith's dissertation. The literature search is still in progress.

Optimization of Evacuated Tube Collectors

This effort involves an array of evacuated tube collectors, which is being tested experimentally and simulated numerically. The approach for testing and analysis follows.

The initial experimental approach is to fill the collectors, which are being tested outdoors, in the morning with constant temperature water. The collectors are then allowed to respond to ambient weather conditions, which are carefully monitored, for about four hours under a no-flow condition. This eliminates the complexities due to flow. The collectors are then purged with constant temperature water to determine the net energy gain during the test period. An IAM program, developed by K. Knappmiller, is then run, and the resulting IAM's are used in TRNSYS to model system performance.

Our initial analysis has focused on using TRNSYS to assess the split between beam and diffuse radiation, as this is very important in determining the absorbed energy for an evacuated tubular collector. Using the Erbs model, we get agreement between the total horizontal and tilt (at 45 degrees) radiation by using a ground reflectance of greater than 1.0, which is unreasonably high. To eliminate experimental error, we have sent an Eppley PSP pyranometer back to Eppley for calibration.

We are also continuing the rewriting of the IAM program to make it more user friendly. We have found that the Constructive Solid Geometry (CSG) model used in the original program by Knappmiller, and the associated non-linear transformations required to construct a final geometry, are too difficult to implement. Phase I of the rewrite, with black tubes on a flat backplane is complete, and is being tested. Phase II, almost complete, will add reflection to all surfaces, including diffuse, specular and semi-specular components of reflectance. Phase III will add the glass covers.

Off-line Testing of a Natural Convection Heat Exchanger

The testing and analysis of a natural convection heat exchanger system has been completed. An MS Thesis for the work will be available soon, along with files of raw data, which we shall make available via anonymous ftp.

The system consists of a lower, unpressurized, custom primary tank, and a standard 40 gallon auxiliary tank situated on top of the primary tank. Water may be circulated through the primary tank and to a solar collector (although we did not implement a full system with a collector). The 40 gallon tank is plumbed to a 120 ft long copper coil immersed in the primary tank. Water circulates via natural convection from this immersed copper coil through the auxiliary tank.

Testing was accomplished by: 1) Natural heating - filling the lower tank with constant temperature (hot) water, and the upper tank with (cold) mains water, and 2) Forced heating - starting both tanks at the same temperature, and heating the lower tank using an external boiler. The heat input was controlled at 1, 2, 3 and 4 kW. Eight vertical temperature measurements were taken in each tank, together with flow rate measurements using a low pressure drop Micro Motion flow meter. Measurements were taken every 30 seconds, and averaged to yield 5 minute data. The duration of each test was about four hours.

Data analysis of the heat transfer and fluid flow rates were accomplished for all tests. The dominant thermal resistance observed was due to the natural convection outside of the coil - the internal thermal resistance was small. The fluid flow resistance was observed to be about a factor of two greater than for standard, laminar flow. This is perhaps due to the secondary motion introduced by centripetal forces.

TRNSYS runs were made to simulate the two-tank system. Fairly good agreement on overall energy transfer was obtained, but the stratification in both tanks is not well modeled. By adjusting the number of nodes in the lower tank, either the stratification in the upper tank, or the stratification in the lower tank, could be matched, but not both simultaneously.

Experimental results indicate very good performance of the system. However it was found from TRNSYS simulations that the coil could have been decreased in length by one-half, with no observable change in system performance. The reduction in heat exchanger area was almost exactly compensated for by an increase in flow rate due to reduced resistance to flow.

ANALYSIS OF A LOAD-SIDE HEAT EXCHANGER FOR A SOLAR DOMESTIC HOT WATER HEATING SYSTEM

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ABSTRACT

The Colorado State University Solar Energy Applications Laboratory is currently testing several solar domestic hot water heating systems. The experimental systems are fully instrumented to yield data appropriate for in-depth analyses of performance. One of these systems is an unpressurized drainback system with a load-side heat exchanger. An analysis of the performance of this heat exchanger is the focus of this paper. Analytical calculations for the effectiveness and convective heat transfer coefficients from correlations are compared against the experimental data. Three models for the convective heat transfer coefficients are analyzed and yield results that underpredict the measured heat transfer from 23% to 72%. TRNSYS simulations were performed using the average effectiveness (the calculated effectiveness varies from 0.68 to 0.95); the results compare favorably with experimental results, indicating that a constant effectiveness is an adequate model for the system.

INTRODUCTION

Load-side heat exchangers are used in solar domestic hot water heating systems for several reasons. One such reason is to allow the use of an unpressurized storage tank; thus, a drainback system (to avoid freezing of the lines to and from the collector) can be used. Also, if the system is unpressurized, then the collectors can be designed for lower pressures. Furthermore, the use of a load-side heat exchanger eliminates the need for a pump on the load delivery side. The system

tested is shown schematically in Figure 1. To transfer the heat from the solar storage tank to the load, water mains supply, shown as point A, travels downward to point B, and then circulates upward through the coil that is submerged in the solar storage tank. The water exits the solar storage tank and then enters the auxiliary tank, via a dip tube, at point C. This arrangement may promote stratification, depending on the location of the coil, which can increase collector performance [Farrington and Bingham, 1987]. This type of heat exchanger operates only when a draw is occurring and is a "one-pass" system. In support of the Solar Ratings and Certification Corporation's (SRCC) ongoing work, validation of the model of the load-side heat exchanger is important.

SYSTEM DESCRIPTION AND INSTRUMENTATION

A comprehensive system instrumentation plan has been designed to effect system measurements of temperatures, electrical power input, flow rate, and differential pressure across the pumps. In addition, a weather station has been designed and installed specifically for these tests. Measurements from the weather station constitute inputs while system measurements constitute system states and outputs. A detailed test of the systems under actual operating conditions is the intent of these experiments.

Outdoor and system measurements are made as indicated in Figure 1. Measurements are taken every 8 seconds. Three hot water draws are accomplished each day at 8 a.m., noon, and 4 p.m., as per ASHRAE standard 95-85 [1985]. The test data were acquired for the time period of July 24 - 28, 1994. The data were reduced to 5 minute averaged intervals, which is well within the time constant of the systems, except the energy calculated during the seven minute draws is determined using the 8 second intervals.

Fig. 1 Weather station and system instrumentation

All temperature measurements are made with high-grade type T (Copper/Constantin) thermocouples, with an accuracy of ± 0.1 °C. It is essential that extremely accurate temperature measurements be made, as the temperature differences are directly proportional to the energy

transfer rates. All thermocouples are shielded using electrical conduit, and electrically isolated from the shields to prevent ground loops. Thermocouple arrays have been designed to measure tank temperatures at eight heights, where the spacing between thermocouples in the tank is uniform at one-eighth of the height of the tank.

The return flow from the drainback collector enters a custom 0.35 m^3 (91 gal) unpressurized solar storage tank (refer to Figure 1) through a stratification manifold. The pump circulates the solar storage tank water to the collectors and back to the solar storage tank at a flow rate of $3.9 \times 10^{-5} \text{ m}^3/\text{s}$ (0.6 gpm). A turbine flow meter with an accuracy of $\pm 0.05\%$ was used to measure the flow. The pump motor is rated at 29.8 W (1/25 hp). Fluid pumping of the system is accomplished using a bang-bang controller, with operation controlled by the temperature difference between the collector outlet and a point on the lower wall of the solar storage tank.

The load-side heat exchanger coil is 36.58 m (120 ft) of 0.0191 m (3/4 in) copper tubing coiled through the entire height of the solar storage tank. The coil is situated near the outer edge of the inside of the tank. The draw flow rate is $1.96 \times 10^{-4} \text{ m}^3/\text{s}$ (3.1 gpm), a turbine flow meter with an accuracy of $\pm 0.5\%$ was used to measure the flow. During a draw potable water from main supply flows to the bottom of the solar storage tank, then spirals upward through the heat exchanger coil. The water then flows into the bottom of a standard 0.14 m^3 (36 gal) auxiliary storage tank. Energy is transferred between the tanks only when there is a hot water draw.

EXPERIMENTAL RESULTS

Due to space limitations, only two days of the five days of test data will be presented. The two days are July 26 and July 27, 1995. These days were chosen as representative of the entire data set. July 26 was a fairly cloudy day and July 27 was a clear day with the highest solar insola-

tion of the five days. As an indirect example of the heat exchanger performance, the solar storage tank temperatures are shown versus time in Figure 2. The top node of the tank is denoted by N=1, while the bottom node of the tank is denoted by N=8. The times when draws are done are denoted on the figures with a "D."

An average stratification of 10-15 °C is observed in the tank, with greater stratification in the afternoon, due to the solar contribution. The very top node of the tank is cooler than several of the nodes below it. This is due mostly to an air gap which exists at the top of the tank when the collector is being pumped, and in part to the heat loss out the top of the tank. The stratification becomes more pronounced during a draw because the temperature in the bottom of the tank decreases by as much as 15 °C while the temperature in the top of the tank decreases by only 3-5 °C. This suggests that the mains supply water gains most of its energy from the bottom of the tank. During the day, between draws, the tank water temperature increases due to the solar contribution. This is especially noticeable in the bottom of the tank (probably due to the stratification manifold). During the night, the bottom of the tank actually warms due to the conductive heat transfer from the rest of the tank.

Figure 3 shows the time histories of the temperature rise across the heat exchanger. The water mains supply temperature remains fairly constant at 16 °C. As seen from Figure 3, initially the energy from the heat exchanger is very high. This is due to the fact that the fluid in the heat exchanger has had the time between draws (at least four hours) to come into equilibrium with the temperature of the fluid in the tank. During a draw, the residence time of water in the coil is 53 seconds at a draw flow rate of $1.96 \times 10^{-4} \text{ m}^3/\text{s}$ (3.1 gpm). Thus, after about 1 minute, the heat exchange should begin to be well represented by traditional correlations. The additional amount of energy in the "humps" in Figure 3, totaled over the day, is only three percent of the total daily

energy drawn off.

Fig. 2 Main solar storage tank temperatures

Fig. 3 Heat exchanger inlet and outlet versus time

ANALYSIS

The errors reported in the text are computed in quadrature as recommended by Holman [1978]. The fluid properties are assumed to be known with no associated errors.

Effectiveness

In this section, the performance of the heat exchanger is considered based on heat exchanger design theory. The average heat exchanger effectiveness is calculated from the experimental data and a representative correlation that best predicts the experimental effectiveness is sought.

The heat exchanger effectiveness is calculated at each data point using equation (1).

$$(1) \quad \varepsilon = \frac{\Delta Q}{\Delta Q_{\max}} = \frac{\dot{m}_{\text{drw}} C_p (T_{\text{HX, out}} - T_{\text{HX, in}})}{\dot{m}_{\text{drw}} C_p (T_{\text{TnkMax}} - T_{\text{HX, in}})}$$

Figure 4 presents the calculated effectiveness for over two days of draw data. The average effectiveness is obtained by summing all of the calculated effectivenesses and dividing by the total number of data points, the average effectiveness for all data is 0.78, +/- 0.5%. This excludes the first minute of data when the heat exchanger is being purged of the warmed coil water. The disparity in the curves is due to the dependence of the outside convective heat transfer coefficient on the outside temperature difference. Notice on Figure 2 that the tank temperature does not rise significantly between the 8 a.m. draw and the noon draw on July 26, 1995, and that this temperature is the lowest on the plot. This was a cloudy day and the pump did not operate much; hence the temperature difference is lower, resulting in a reduced Raleigh number, and thus a lower effectiveness. Also, it was found that the effectiveness is not significantly affected by whether or not the

pump is on or off during the actual draw. This is not surprising since the tank is so large.

Fig. 4 Heat exchanger effectiveness

The effectiveness is predicted using various effectiveness correlations for different types of heat exchangers in order to best match the experimental effectiveness. The data indicate that the operation of this heat exchanger most resembles a parallel, counterflow shell and tube, shell fluid mixed; or a cross flow, one fluid mixed type heat exchanger. The minimum capacity rate is based on the draw flow (817 W/°C) and the maximum capacity rate is based on the tank capacitance (3428 W/°C). Since $C_{\max}$ is much greater than $C_{\min}$, the flow arrangement configuration becomes less important in the effectiveness calculation. Equation (2) [Kays and London, 1964] is for a cross flow heat exchanger with one fluid mixed. The average overall heat transfer coefficient used is 1500 W/°C, calculated from experimental data (discussed later). The resulting calculated effectiveness is 0.76.

$$(2) \quad \epsilon = \frac{C_{\max}}{C_{\min}} \left(1 - e^{-\Gamma \frac{C_{\max}}{C_{\min}}} \right), \quad \Gamma = \left(1 - e^{-N_{TU}} \right), \quad N_{TU} = \frac{UA}{C_{\min}}$$

Heat Transfer Coefficient Models

In this section, the heat exchanger performance is considered based on the overall heat transfer coefficient. Four different models are used to predict the outside heat transfer coefficient of the heat exchanger. None of these models strictly apply to this heat exchanger because of the different assumptions and configurations of the models. These are then used to calculate an overall heat transfer coefficient. The predicted coefficients are then compared to the experimentally determined average values.

The overall heat transfer resistance of the heat exchanger is comprised of three terms. The first is the inside convective coefficient due to forced flow through the coil (draw flow), the second is the conductance through the copper tubing, and the last is the outside convective heat transfer coefficient due to the natural convection flow around the coil.

The inside convective coefficient, h_i , is determined using the Dittus and Boelter equation, equation (3), [Welty,1976], for the fluid being heated. The Reynold's number is 19,104 (the flow is turbulent), yielding a Nusselt number of 109.

$$(3) \text{Nu}_i = \frac{h_i D}{k} = 0.023 \text{Re}_D^{0.8} \text{Pr}^{0.4}$$

The corresponding convective coefficient is 3,576 W/m²°C (630 Btu/hrft²°F), +/- 7%. This correlation is well documented in the literature, but it is for a horizontal straight coil. The curvature will increase the convective coefficient by a factor of 1.16 +/- 10%, Chauvet, et. al., [1994]. The conductance through the copper tubing is negligible. Therefore, only the outside convective coefficient remains to be calculated. Once the outside convective coefficient is determined, based on the outside surface area of the coil, the overall heat transfer resistance is calculated from equation (4).

$$(4) \quad U_{\text{tot}} = \left[A_{\text{coil}} \left(\frac{1}{h_o A_{D_o}} + \frac{1}{h_i A_{D_i}} \right) \right]^{-1}$$

The outside convective coefficient is predicted using four methods. Method (a) uses a correlation for a vertical cylinder, given as equation (5) from Kakac [1987], which is valid for all Ra_D . The large coil diameter (0.66 m, 26") is used as the characteristic length. This correlation actually was developed for an isothermal solid wall in an infinite fluid, with no influence from the surrounding wall; none of these restrictions apply to this case.

$$(5) \quad Nu_o = \frac{4}{3} \left[\frac{7 Gr Pr^2}{5 (20 + 21 Pr)} \right]^{\frac{1}{4}} + \frac{4 (272 + 315 Pr) L}{35 (64 + 63 Pr) D}$$

Method (b) assumes that the coil is a vertical flat plate (front and back). This could be inferred since $D/L > 35/Gr_D^{1/4}$ [Kakac, 1987], with less than 5% error. The correlation, given as equation (6), is based on the height of the tank, L (1.4 m, 55"). This method does not account for the interaction of the fluid leaving one coil loop and interfering with the next coil loop, nor for the effect of flow over a cylinder.

$$(6) \quad Nu_o = \left\{ 0.825 + \frac{0.387 Ra^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right\}^2$$

Method (c) is for flow over a horizontal cylinder, this method also does not account for the interactions between coil loops. The Ra number is based on the diameter of the tubing (0.019 m, 0.75"). The McAdams correlation [Welty, et. al., 1976], which holds for $10^4 < Ra_D < 10^9$, is used, equation (7). The exponent of $1/4$ is for laminar flow.

$$(7) \quad Nu_o = 0.53 (Gr_D Pr)^{1/4}$$

All three of the above methods use an average representative temperature difference of 14.7 °C (the minimum was 10.8 °C and the maximum was 22.5 °C, the temperatures during the purging time are not included). This is an average of the difference between the average tank temperature and the coil temperature $(T_{out}/2 + T_{in}/2)$, calculated at each point over the course of a draw. This temperature difference is named "coil dT."

Method (d) uses the correlation from Farrington and Bingham [1987], equation (8). Note that they recommend using a Log Mean Temperature Difference (LMTD), given as equation (9), as the driving potential. The calculated average LMTD is 9.8 °C (the minimum was 6.5 °C and the maximum was 14.1 °C, the LMTD's during the purging time are not included). Further, they recommend $C=0.9$ and $m=1/4$ (for laminar flow).

$$(8) \text{Nu}_o = C \text{Ra}_D^m$$

$$(9) \text{LMTD} = \frac{(T_{\text{TnkBot}} - T_{\text{HX, in}}) - (T_{\text{TnkMax}} - T_{\text{HX, out}})}{\ln \left(\frac{T_{\text{TnkBot}} - T_{\text{HX, in}}}{T_{\text{TnkMax}} - T_{\text{HX, out}}} \right)}$$

The results are presented in Table 1, along with the results from the next section.

Experimental Data Reduction

Finally, the test results are reduced using three different methods. Equation (10), method (e), uses the measured values to do an energy balance around the coil. The calculations are completed for each data point, and an average is obtained, M is the total number of data points. Since three draws are performed each of the 5 days, for a total of seven minutes per draw, and the data is taken every 8 seconds, the average is obtained for about 262 data points. Method (f) uses the same equation, except the temperature difference, (coil dT), is replaced with the LMTD. In all cases, the area, A , is the surface area of the coil and is equal to 1.86 m² (20 ft²). The average calculated value

is 1,500 W, +/- 2.5%, using the coil dT. Once the overall heat transfer coefficient is known, the average convective heat transfer coefficient can be determined, +/- 5%.

$$(10) U_{\text{meas}} = \left\langle \sum_1^M \frac{\dot{Q}_{\text{drw}}}{A (\text{coil } dT)} \right\rangle / M = \left\langle \sum_1^M \frac{(\dot{m} c_p)_{\text{drw}} (T_{\text{HX, out}} - T_{\text{HX, in}})}{A (\text{coil } dT)} \right\rangle / M$$

Equation (11), method (g), solves the heat balance equation on a nodal basis. The heat exchanger coil is treated as a 36.58 m (120 ft) long tube. The distance traveled along the tube in the direction of flow is designated as x , and P is the perimeter of the tube. The temperature of the fluid inside the tube is $T_{x,N}$ and $T_{\text{Tank},N}$ is assumed to be the temperature of the surrounding tank water at each node N . To solve this equation, the tank and tube are divided into eight equal segments to correspond to the eight tank temperature measurements. A constant overall heat transfer resistance value, U_{tot} , is then assumed. Applying equation (11) for $N=1$ yields the inlet temperature to node 2. This procedure is done successively until the outlet temperature of the coil, $N=8$, is obtained. The process is repeated using different U_{tot} values until the final calculated coil outlet temperature predicted from equation (11) matches the experimentally measured temperature.

$$(11) \ln \theta_N = \ln \theta_0 - \frac{U_{\text{tot}} P}{\dot{m} C_p} x, \quad \theta_N = T_{x,N} - T_{\text{Tank},N}$$

The results of the calculations are provided in Table 1. The numbers in the % diff. column are based on the experimentally determined average heat transfer, equation (12), and the heat transfer calculated using the correlations. The experimental average heat transfer value is 22,099 W, +/- 1%. Note that the horizontal cylinder correlation, method (c), agrees with the measured data to within 23%. Since the correlation for a single horizontal cylinder underpredicts the heat transfer, stacking the coils seems to enhance the heat transfer. The coils within the tank are stacked

on top of one another with almost no gap in between each successive coil. The enhanced heat transfer may be attributed to vortex shedding off of each coil, which means the flow is probably turbulent over each coil. This local turbulence would increase the heat transfer.

TABLE 1: HEAT TRANSFER PROPERTIES

Method	Eqn	Ra	h_o (W/m ² C)	UA (W/C)	Q (W)	% diff.
(a) vert. cyl.	(2)	1.5×10^{11}	315	414	6,105	+72
(b) vert. plate	(3)	1.4×10^{12}	672	825	12,162	+45
(c) horiz. cyl.	(4)	3.6×10^6	756	1,159	17,082	+23
(d) Farr&Bing	(5)	3.4×10^5	716	1,113	10,909	+51
(e) $Q_{ave}/A(\text{coil } dT)$	(7)	3.4×10^6	1,042	1,504	22,173	-0.3
(f) $Q_{ave}/A(\text{LMTD})$	-	3.4×10^5	1,852	2,264	22,188	-0.4
(g) exponential	(8)	3.4×10^6	1,031	1,493	22,004	+0.4

$$(12) \dot{Q}_{\text{experimental, ave}} = \left(\sum_{1}^M \dot{m} C_p (T_{\text{HX, out}} - T_{\text{HX, in}}) \right) / M$$

Also, note the poor agreement (51%) from Farrington and Bingham's correlation, method (d). This is perhaps due to the short duration of our draws (7 minutes) relative to the long duration of Farrington and Bingham's tests (3 to 4 hours). Only a small fraction of energy is extracted from the tank over 7 minutes, a "shallow" draw, while in contrast in Farrington and Bingham's tests the tank was nearly completely purged of energy over 4 hours, a "deep" draw. Thus, the performance of our system, where the tank temperature changes little, is not modeled well using the Farrington and Bingham correlation. A load-side heat exchanger of this design appears to be relatively ineffective at extracting energy from the tank over "shallow" draw profiles. Therefore, it probably leads to sub-optimal performance of the system.

Curve Fits

The generalized form of equation (8) was re-evaluated using the experimental data to investigate the Nusselt and Raleigh numbers to determine trends. The correlation is considered for

both average representative temperature differences, the coil dT (14.7 °C) and the LMTD (9.8 °C). The logarithm of the average Nusselt number per draw is shown in Figure 5 plotted against the logarithm of the average Raleigh number per draw. Each line represents different values for C and m, as given in equation (13). For one case, the coefficient m is forced to be 1/4, for laminar flow, and the average C value is then determined. For the second case, a two-parameter curve fit is performed, thus allowing both m and C to be determined. The results for Farrington and Bingham are also presented.

$$(13) \text{Log}_{10}(\text{Nu}) = \text{Log}_{10}(C) + m\text{Log}_{10}(\text{Ra})$$

As can be seen in Figure 5, using the LMTD as the representative temperature difference results in much greater spread of the data. Hence, the coefficient of determination, r^2 , for the fits have a lower value than when using the coil temperature difference. The calculated slopes, m, for both models are lower than for laminar flow (0.104 using the coil temperature and 0.188 using the LMTD). The calculated heat transfers using these results are shown in Table 2. By using the coil temperature difference the final result is within 2% of the actual, regardless of whether m is forced to be 1/4 or not. If the LMTD is used, the difference increases to 8%. Therefore, it is recommended that the coil temperature difference be used for calculations.

TABLE 2: HEAT EXCHANGER COEFFICIENTS

TEMP. DIFF.	m	C Value (ave)	Ra	h_o (W/m ² C)	UA (W/C)	Q (W)	% diff.
coil dT m forced	0.25	0.75	3.4×10^6	1,067	1,533	22,598	-2
coil dT curve fit	0.104	6.66	3.4×10^6	1,048	1,513	22,300	-1
LMTD m forced	0.25	2.60	3.4×10^5	2,064	2,442	23,934	-8
LMTD curve fit	0.188	5.65	3.4×10^5	2,041	2,425	23,764	-7
LMTD Far&Bing	0.25	0.9	3.4×10^5	716	1,113	10,909	+51

Fig. 5 Log(Ra) vs. Log (Nu)

TRNSYS SIMULATION RESULTS

It is also important to assess the performance of the solar system vis a vis the SRCC efforts, which seek to rate system performance using TRNSYS [Klein et al., 1994] simulations [Burch et al., 1993]. One of the overall goals is to match the model to experimental data by adjusting the model parameters to represent more accurately the physical properties of the system.

A TRNSYS module for modeling a load-side heat exchanger does not exist in version 13.1 (used at the time of this work), so the tank models were “tricked” into simulating one by including a heat exchanger effectiveness parameter. The TRNSYS tank model variables are shown in Figure 6. The measured average heat exchanger effectiveness of 0.78 is used to adjust the load flow out of the solar storage tank, m_{LM} . The temperature of the replacement fluid into the solar storage tank, T_{LM} , is at mains supply temperature since the flow into the main solar storage tank from the “hot source” (collectors), T_{HM} , is not equal to the solar storage tank load flow. The flow from the “hot source” (the solar storage tank) into the auxiliary storage tank, m_{HA} , is zero. The temperature entering the auxiliary storage tank as replacement fluid, T_{LA} , is also adjusted by the effectiveness.

The draw flow from the auxiliary tank, m_{LA} , is the experimental value.

Fig. 6 TRNSYS tank model

The results are shown in Table 3, indicating excellent agreement. The $F_R(\tau\alpha)$ and $F_R U_L$ (first order equation is used) are determined from the experimental data. An energy balance was performed on the system using the measured data and closure was obtained to within two percent. An energy balance using TRNSYS predicted values exhibits closure to within 2.6%. A TRNSYS simulation was used to predict the performance for another set of experimental data and the results were equally good. Therefore, this is deemed an acceptable model for this system.

TABLE 3: MEASURED AND TRNSYS RESULTS

Quantity	Measured ₁	TRNSYS	%Error
Q_{useful} (MJ)	187	186	+0.3
Q_{draw} (MJ)	114	103	+9.6

CONCLUSIONS

The external flow around the coil is laminar, natural convection, while the inside is turbulent, forced convection. The inside heat transfer coefficient contributes little to the overall thermal resistance, while the outside resistance is dominant. The best agreement for the outside heat trans-

fer coefficient is obtained using the correlation for free convection from a horizontal cylinder, with a 23% enhancement most probably due to an increase in the heat transport attributed to turbulence generated from flow over the stacked coils of the heat exchanger. An average effectiveness of 78% was calculated from experimental data. When used in a TRNSYS model, agreement to within 0.5% of collector useful energy is achieved, as well as agreement to within 10% for the energy delivered, indicating that for "shallow" energy draws, a constant effectiveness model yields good agreement with observations.

NOMENCLATURE

A	=	Area, m ²
C _p	=	Specific Heat, J/kgC
D	=	Representative Diameter, m
Gr	=	Grashof Number
h	=	Convective Heat Loss Coefficient, W/m ² C
k	=	Thermal Conductivity, W/mC
L	=	Representative Length, m
LMTD	=	Log Mean Temperature Difference, °C
m	=	Mass Flow Rate, kg/s
Nu	=	Nusselt Number
P	=	Perimeter, m
Pr	=	Prandtl Number
Q	=	Rate of Energy, W
Ra	=	Raleigh Number
Re	=	Reynolds Number
T	=	Temperature, °C
U _{tot}	=	Overall Heat Loss Coefficient, W/m ² C
ε	=	Effectiveness

Subscripts

drw	=	Draw
HA	=	Hot Inlet on Auxiliary Tank
HM	=	Hot Inlet on Main Tank
HX,in	=	Coil Heat Exchanger Inlet
HX,out	=	Coil Heat Exchanger Outlet
i	=	inside
LA	=	Replacement Inlet on Auxiliary Tank
LM	=	Replacement Inlet on Main Tank

NA = Bottom Outlet On Auxiliary Tank
 NH = Bottom Outlet On Main Tank
 o = outside
 TnkBot = Tank Bottom Temperature
 TnkMax = Tank Maximum Temperature
 1A = Draw Outlet on Auxiliary Tank
 1H = Draw Outlet on Main Tank

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MANAGEMENT AND COORDINATION OF COLORADO STATE/DOE PROGRAM

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