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

**PROJECT STATUS REPORT FOR THE MONTHS OF
APRIL AND MAY 1996**

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MASTER

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).

The natural convection boundary layer on the wall of the tank will be modeled using an approximate method to solve the boundary layer equations for natural convection on a vertical surface in a stratified environment. When coupled to the equations governing the performance of the external heat exchanger, the total heat transfer to the tank and the distribution of the heat flux along the tank wall can be estimated. Inside of the tank, the boundary layer solution also provides an estimate of the rate of entrainment of fluid from the central core region of the tank as the natural convection boundary layer grows along the tank wall. Coupling the rate of entrainment from the boundary layer calculations to a traditional, one dimensional, control volume model of the core region of the tank provides a more accurate estimate of the convective motion of the water in the tank. In this way, the model provides an estimate not only of the overall heat transfer to the tank, but also the flow of mass and heat inside of the tank and therefore, the development of the vertical tank temperature profile over time.

Drake employed integral methods to model the natural convection boundary layer in her investigations of cylindrical fluid storage tanks with uniform wall heat flux. Webb (1988, 1989, 1990) developed this idea further to investigate the natural convection processes in large oil filled underground caverns of the Strategic Petroleum Reserve (SPR) over long periods of time (up to 30 years). In the case of Webb's model, the natural convection boundary layer solutions were calculated using a modified local similarity method.

Several approximate techniques for solving the natural convection boundary layer equations are currently being evaluated, including Webb's modified local similarity method. Regardless of the method chosen, the similar or pseudo-similar boundary layer equations must be solved numerically. Computer programs for two commonly used numerical solution techniques have been developed. The first is a shooting method outlined by Nachtsheim and Swigert (1965). The second is an implicit finite difference method developed by Keller (1971, Cebeci 1988) to solve the boundary layer equations and other parabolic partial difference equations.

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RATING AND CERTIFICATION OF DOMESTIC WATER HEATING SYSTEMS

April/May 1996

Indoor testing of the horizontal-tank thermosyphon system under investigation has been completed. Current work involves outdoor testing of the system and further analysis of the heat exchanger calculations used in the TRNSYS model of the system.

OUTDOOR TESTING

The complete system has been installed on the Outdoor Test Bed at the CSU Solar Energy Applications Laboratory and data is being collected to finish the calibration procedure outlined in the December/January 1996 Progress Report.

HEAT EXCHANGER ANALYSIS

While performing the model calibration using the technique outlined in the December/January 1996 Progress Report, it was discovered that the heat exchanger overall conductance (UA) adjustment factor required to match the temperature profile exiting the heat exchanger depended on the operating conditions of the heat exchanger/boiler system. As a result of this, time has been spent examining the operation of the heat exchanger and the assumptions made in the heat transfer correlations in the model. The last report (February/March 1996) summarized the heat transfer relations used in the model and reported on an attempt to calculate the heat transfer coefficient on the inside of the tank by forcing the outside heat transfer coefficient to be large. This report continues the analysis.

High Flow Constant Temperature Test

We wish to determine the inside heat transfer coefficient as a function of the driving temperature difference (or appropriate dimensionless parameter). To isolate the inside heat transfer coefficient, a large flow rate on the shell side (outside) of the heat exchanger is required so that the outside heat transfer coefficient becomes very large and the inside resistance to heat transfer become dominant (April/May 1996 Progress Report). By running a forced-flow test at a constant flow rate of 300 kg/h and a constant heat exchanger inlet temperature of approximately 60°C, a range of operating conditions may be obtained. This is performed using a proportional-integral (PI) controller built into the experimental control and data acquisition software being used in the study. 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 will reduce the energy input to the heat exchanger and thus provide a range of operating conditions necessary for the desired correlation.

Figure 1 shows the overall heat transfer coefficient which is the same as the inside heat transfer coefficient under the assumption made above during the four hour test. Figure 2 shows the coefficient as a function of driving temperature during the test. Larger driving temperatures lead to larger heat transfer coefficients. The mean temperature difference described in the previous progress report (February/March 1996) was selected as the driving temperature difference since that is the closest equivalent to the temperature difference used in the model. The model actually uses the difference between the tank 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/m²C near the beginning of the test (after the initial transient) to about 90 W/m²C near the end.

Figure 3 shows the TRNSYS 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 overall heat transfer coefficient ranges from a peak of about 150 W/m²C near the start of the test to about 125 W/m²C at the end of the test which is about ten to thirty percent above the calculated values. The model predicts a nearly constant outside coefficient due to the assumption of fully developed forced laminar flow in the annulus. This assumption has been questioned previously, but if the outside heat transfer coefficient is less than the inside heat transfer coefficient, the assumption that the inside resistance is dominant is incorrect. However, the analysis is continued assuming that the assumption is valid.

Inside and Outside Heat Transfer Coefficient Correlations

The model determines the Nusselt number, and thus heat transfer coefficient, using

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

where eqn. 1a is for laminar convection and eqn. 1b is for turbulent natural convection. The Rayleigh Number, Ra , is the product of the Grashof and Prandtl numbers and is a measure of the buoyant forces present.

We wish to obtain a correlation of the form

$$Nu = f(Ra)\tag{2}$$

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 in the form of eqn. 1, namely

$$Nu = C(Ra)^n\tag{3}$$

where the constants C and n are determined by a fit to the experimental data. Figure 4 shows a plot of the Nusselt number as a function of the Rayleigh number and the fitted equation

$$Nu = 0.15(Ra)^{0.26}\tag{4}$$

The correlation coefficient is 0.94 ($R^2=0.88$) and the standard error is 6.1. Further tests using eqn. 4 did not provide useful results, so the data were refitted without constraining to the form of eqn. 3.

Figure 5 also shows the Nusselt Number as a function of the Rayleigh Number, but the fit has been changed to a quadratic form and is

$$Nu = 93.67 + 6.138 \cdot 10^{-11}(Ra) + 7.464 \cdot 10^{-22}(Ra)^2. \quad (5)$$

The correlation coefficient is 0.97 ($R^2=0.94$) and the standard error is 3.9. Note that the first and second order terms in eqn. 5 are small when the Rayleigh number is on the order of the transitional Rayleigh number of 1×10^9 , so that eqn. 5 predicts a nearly constant Nusselt number of 94 at the lower end of the turbulent regime.

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 Figures 4 and 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 the unrealistically large heat inputs near the beginning of the constant temperature test.

In an attempt to improve model performance, eqn. 5 is substituted for eqn. 1b in the model and simulations of several low-flow experiments are performed to determine the outside Nusselt number required to predict performance accurately. Extrapolation of the curve fit beyond the experimental data is questionable, but as a rough check, eqn. 1b predicts a Nusselt number of 100 at the transitional Rayleigh number and the new correlation, eqn. 5, predicts a nearly constant value of 94. Thus, no large discontinuity will be introduced. Figure 6 shows the 'old' and 'new' heat transfer correlations. The 'old' correlation is on the top. Below the transitional Rayleigh number of 1×10^9 , the correlations are the same. Note the nearly constant turbulent Nusselt number prediction of the new correlation until large Rayleigh numbers are reached.

Table 1 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 model is forced with the heat exchanger inlet temperature and flow rate profiles for constant heat input low-flow rate tests. The Nusselt number decreases with decreasing flow rate at nearly constant heat input and increases with decreasing heat input at nearly constant flow rate.

Average Energy Input (W)	Average Mass Flow Rate (kg/hr)	Nu
1100	43	2.3
1050	28	1.6
860	16	0.92
548	45	3.1

Table 1. Required Nusselt Number for Several Low-Flow Tests

Conclusions

The uncertainty in this analysis is the principal assumption that the inside resistance to heat transfer is much larger than the outside. If true, the analysis might serve to improve the heat transfer predictions in the model after determining an appropriate relation for the outside Nusselt number. If not, the calculation of the overall heat transfer coefficient will have been incorrectly divided between the inside and outside coefficients. More analysis needs to be done to determine, if possible with the given data, the validity of the assumptions that have been made. Without further justification of the principal assumption, which is questionable at this point, adjustment of the overall heat transfer coefficient at a single appropriately selected operating condition may be the best solution.

Figure 1. Overall Heat Transfer Coefficient vs. Time

Figure 2. Overall Heat Transfer Coefficient vs. Driving Temperature Difference

Figure 3. Heat Transfer Coefficient vs. Time

Figure 4. Nusselt Number vs. Rayleigh Number (Power Fit)

Figure 5. Nusselt Number vs. Rayleigh Number (Quadratic Fit)

Figure 6. Nusselt Number Correlations. Old Correlation on Top.

ADVANCED RESIDENTIAL SOLAR DOMESTIC HOT WATER (SDHW) SYSTEMS

This report is for April and May, 1996.

Experimental Work

As indicated in the last progress report, the tools and techniques for data analysis of the collector data were being developed. Almost all of the effort during April and May has been expended on data analysis of the solar collectors. Analyses conducted encompassed:

- Using different solar radiation models: 1) measured total radiation on the tilt, using TRNSYS to predict: 2) beam and diffuse on the tilt, and 3) beam, sky diffuse and ground diffuse on the tilt. Various models were used to split the total horizontal radiation into horizontal beam and diffuse, and other models were used to project the sky diffuse onto the tilt.
- Using different optical models: 1) average $(\tau\alpha)$ for total incident radiation, 2-4) $(\tau\alpha)_n$ using both first through fourth-order IAM's.
- Also, regressions were performed on both collector efficiency vs. time, and Q_{useful} vs. time.
- Also, using FR_{UL} measured at night, extrapolated to daytime, as a known with the daytime data to estimate the optical parameters.

Toni Smith's PhD dissertation should be completed during the next month, and available for distribution.

Optimization of Evacuated Tube Collectors

This effort involves an array of evacuated tube collectors, which is being tested experimentally and simulated numerically. In the last progress report, the progress to date was given encompassing calibrations, and data collection using the specular backplane.

Specific tasks accomplished during these two months include:

- Detailed testing on the new array of NEG collectors, with different backplanes, proceeded during this period. Here, diffuse white and black backplanes were used. All tests to date have been for the eight-tube system. Additional tests are planned using all of these backplanes, where every other tube will be removed, resulting in a four-tube system, with significant spaces in between the tubes. This should allow us to assess the effect of the backplane reflectance and spacing.
- Phase III of rewriting the IAM program, with the addition of refraction to the optical model of the glass covers, is nearly complete. Photons are being emitted and traced accurately. The program has been handed off to Joe Ryan for detailed testing of correctness, further development, and general clean-up.

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.