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**NATURAL CONVECTION HEAT EXCHANGERS FOR SOLAR WATER HEATING
SYSTEMS**

Technical Progress Report
December 1, 1995 to January 31, 1996

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NATURAL CONVECTION HEAT EXCHANGERS FOR SOLAR WATER HEATING SYSTEMS

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Reporting Period: December 1, 1995 to January 31, 1996

Project Personnel: Jane H. Davidson
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Project Objectives:

The goals of this project are 1) to develop guidelines for the design and use of thermosyphon heat exchangers external to the storage tank in solar domestic water heating systems and 2) to establish appropriate modeling and testing criteria for evaluating the performance of systems using this type of heat exchanger.

Experimental Work

Experimental measurements of the performance of the two pass, tube-in-shell heat exchanger show that UA is not solely dependent on thermosyphon and collector flow rates. Because the flow is in the mixed convection regime, models which are based on UA or effectiveness as functions of flow rates will not accurately predict performance.

The inappropriateness of predicting UA on the basis of thermosyphon flow rate (for a fixed collector flow rate) is illustrated with results from a test with a collector flow rate of 0.03 kg/s. The test consisted of two parts. During the first part of the test, the tank temperature was maintained at a uniform temperature of 25 degC while the temperature of the collector fluid was increased every 30 minutes in increments of 15 degC from 25 degC to 95 degC. The tank was held isothermal by the slow addition of cold mains water to the bottom of the tank as warm water from the top of the tank was dumped. The second part of the test began after 200 minutes when the collector fluid was 95 degC. At this point in the test, the cold mains water was disconnected from the tank and the tank was allowed to heat up. Collector outlet temperature, $T_{h,i}$, was held constant. Temperatures at the inlets and outlets of the heat exchanger are plotted over the duration of the test in Fig. 1.

Measured as well as predicted values of UA for this test are plotted versus thermosyphon flow rate in Fig. 2. Figure 3 compares measured and predicted values over the duration of the test. Also shown in Fig. 3 are predicted values of Nusselt numbers on both sides of the heat exchanger. The measured value of UA is determined from,

$$UA = \frac{\dot{m}c_p(T_{h,i} - T_{h,o})}{LMTD} \quad (1)$$

Fig. 1. Measured temperatures at the inlet and outlets of the two-pass, tube-in-shell heat exchanger plotted versus test duration. Collector flow rate is 0.03 kg/s.

Fig. 2. Measured and predicted UA versus thermosyphon flow rate.

Fig. 3. Measured UA and predicted values of Nu and UA plotted versus test duration.

Predicted values of UA are based on published heat transfer correlations. Fully developed forced flow correlations were used on the glycol side (pumped collector side) of the heat exchanger,

$$Nu_F = 4.364 \text{ for } Re < 2300 \quad (2a)$$

from Shah and London (1976) and

$$Nu_F = \frac{(f/2)(Re-1000)Pr}{1+12.7(f/2)^{1/2}(Pr^{2/3}-1)} \text{ for } Re \geq 2300 \quad (2b)$$

from Gnielinski (1976), where f is derived by Churchill (1977a),

$$f = 2 \left[\left(\frac{8}{Re} \right)^{12} + \frac{1}{\left(\{A_1\}^{16} + \{37530/Re\}^{16} \right)^{3/2}} \right]^{1/12} \quad (2c)$$

and

$$A_1 = 2.2088 + 2.547 \ln(42.683 / \text{Re}^{0.9}). \quad (2d)$$

The turbulent heat transfer coefficient determined by eqn. (2b) was doubled to account for the enhanced heat transfer surface of the two-pass tube.

Selection of the appropriate heat transfer correlation on the water (thermosyphon) side of the heat exchanger was somewhat arbitrary since no correlation specific to this geometry is available. A reasonable fit to the measured UA values was obtained by combining forced and natural convection Nusselt numbers, Nu_F and Nu_N , respectively. The shell was modeled as a concentric annulus. Nu_N was determined using a correlation proposed by Keyhani *et al.* (1985) for natural convection from rod bundles inside a shell.

$$\text{Nu}_N = 0.188 K^{0.0442} \eta^{-0.238} (P/d)^{0.045N+0.541} \text{Ra}^{0.322}, \quad (3a)$$

where $P/d=1$. The forced flow value determined from Shah and London (1978) is,

$$\text{Nu}_F = 6.2 \text{ for } \text{Re} < 2300 \text{ (inside an annulus with } K=2.0). \quad (3b)$$

The mixed convection coefficient was obtained using (Churchill, 1977b),

$$\text{Nu}_{\text{total}} = (\text{Nu}_N^3 + \text{Nu}_F^3)^{1/3}. \quad (3c)$$

Both Figs. 2 and 3 show that during the first 200 minutes of testing when thermosyphon flow rate increased due to a decrease in water density in the heat exchanger with no change in the isothermal tank, UA increased as temperature differences in the heat exchanger increased. After 200 minutes, the glycol temperature was held constant and the tank was allowed to heat up. As a result of a decrease in the pressure in the water tank, thermosyphon flow rate decreased but with no appreciable decrease in Nu or UA. Under these operating conditions, heat transfer coefficients did not change significantly because both the temperature difference controlling natural convection and Re of the glycol flow remained nearly constant.

The plot of predicted Nu numbers versus time in Fig. 3 indicates that in this heat exchanger, once the glycol flow becomes turbulent, the greatest resistance to heat transfer is on the thermosyphon side of the heat exchanger. The change from laminar to turbulent flow on the collector side occurred as Reynolds number increased due to lowered glycol viscosity as the temperature was raised. Because thermosyphon flow was laminar, heat transfer is limited by natural convection in the shell. The under-prediction of actual UA is attributed to an under-prediction of natural convection heat transfer.

Clearly, UA is not a unique function of thermosyphon flow rate. The same is true of effectiveness and modified effectiveness. Models based on the assumption that for a given collector flow rate, UA or ϵ is determined by thermosyphon flow rate will produce erroneous predictions of system performance. Other researchers have not uncovered this problem because their experiments were conducted with an isothermal tank. In that situation, pressure difference in the water loop (and thus thermosyphon flow rate) and heat exchanger performance are controlled by collector temperature. Thus for a fixed collector flow rate, effectiveness and UA appear related to thermosyphon flow rate. With conditions like the ones measured in this heat exchanger, UA is tied more strongly to changes in the natural convection heat transfer coefficient. In addition, because viscosity is a strong function of temperature in antifreeze mixtures, UA depends on the values of Re and Pr rather than mass flow rate on the collector side of the heat exchanger.

A paper to be presented at the 1996 ASME International Solar Energy Conference in San Antonio, March 30- April 3, is attached.

Future Activity

Continued fundamental analysis of the thermosyphon heat exchanger is underway and modeling efforts have begun. Scott Dahl is preparing his proposal for his doctoral dissertation.