NATURAL CONVECTION HEAT EXCHANGERS FOR SOLAR WATER HEATING SYSTEMS

Technical Progress Report
April 1, 1995 to May 31, 1995

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Reporting Period: April 1, 1995 - May 31, 1995

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.

Progress:

This progress report presents analysis of the shear pressure drop in natural convection heat exchangers. Although the experimental data are for only one heat exchanger, much of what was learned can be applied to other designs of this type of heat exchanger.

A comparison between the experimental and theoretical pressure drop on the shell side of a two-pass, tube-in-shell heat exchanger was made. The shear pressure drop on the shell side was determined with the heat exchanger in the vertical position. Previous shear pressure drop measurements on the shell side of the heat exchanger were obtained with the heat exchanger in a horizontal position using forced flow under isothermal conditions. Testing in the vertical operating position, with and without heating, is necessary to determine if secondary flows that may result from natural convection currents inside the shell significantly affect the shear pressure drop. The flow on the shell side was forced at flow rates up to 0.03 kg/s (= 0.5 gpm). Based on the hydraulic diameter, \(D_h\), these flow rates correspond to Reynolds numbers (Re) from 0 to 800. At these low values of Re, the flow on the shell side is laminar. Three tests were conducted, one without heating and the other two with the inlet temperature on the tube side set at 60°C and 90°C.

Measured Data:

New pressure ports with a 1 mm diameter were located on the side walls of polycarbonate blocks. Polycarbonate was selected because of the non-reactive nature of the material, low conductivity, and transparency. The new pressure ports have eliminated scale build up near the pressure taps.

A 0.47 cm diameter copper tube was soldered to the outside of the copper shell and runs from the bottom of the heat exchanger to the pressure transducer as shown in Figure 1.
Pressure drop due to shear for the system in Figure 1 is given by

\[ \Delta P_s = \Delta P_m + \Delta P_{\text{tube}} - \int_0^H \rho(z)g dz, \]  

(1)

where \( \Delta P_m \) is the measured value, \( \Delta P_{\text{tube}} \) is the pressure in the small copper tube, and the last term on the right hand side is the hydrostatic pressure in the heat exchanger. Since the temperature distribution, and thus density distribution, in the heat exchanger is unknown, the calculated \( \Delta P_s \) can be grossly over or underestimated. Connecting the copper tube alongside the shell eliminates the problem of calculating the last two terms in eqn. (1). The heat exchanger and tube were then insulated. With this arrangement, the hydrostatic pressure head in the tube is essentially equal to the hydrostatic head in the heat exchanger and the last two terms on the right hand side of eqn. (1) cancel. Thus, \( \Delta P_m = \Delta P_s \). This method does not require an assumed or measured temperature profile in the heat exchanger but may introduce a systematic error resulting from temperature differences in the tube and heat exchanger shell. For example, a 1°C temperature difference could produce an error of 3 or 4 Pa in the shear pressure drop measurement. The accuracy of the differential pressure transducer is ±2.5 Pa.

**Theoretical Pressure Drop**

The heat exchanger was modeled as a combination of individual pipe sections and fittings. The predicted shear pressure loss is the sum of all the individual component losses. Figure 2 shows a diagram of the modeled heat exchanger sections. Pressure losses resulting from any expansions, contractions or bends were determined using documented loss coefficients, \( K \), and the relationship

\[ K = \Delta P / (gV^2). \]  

(2)
Figure 2. Sketch of tube-in-shell heat exchanger with pressure drops.

The heat exchanger shell was modeled as an annulus. Losses in the annulus and connecting piping were determined using the relationship (From Kakac, Shah, and Aung, 1987):

\[ \Delta p = \frac{4(\text{Re})\mu V L}{2D_h^2} \]

where
- \( \mu \) = dynamic viscosity, Pa \cdot s
- \( D_h \) = hydraulic diameter, m
- \( V \) = mean velocity in channel, m/s
- \( L \) = length, m
- \( f \) = Fanning friction factor

(3)
For the annulus:

\[ fRe = \frac{16(1 - r^*)^2}{1 + r^*^2 - 2r_m^2} \]

\[ r^* = \frac{d_i}{d_o} \]

\[ r_m = \left[ \frac{1 - r^*^2}{2\ln(1/r^*)} \right]^{1/2} \]  \hspace{1cm} (4)

For laminar flow, the term, \( fRe \), has a value of 16 for a smooth a circular pipe and a value of 23.9 for an annulus.

Four different configurations of the heat exchanger were studied to isolate the sources of shear pressure losses. The four cases were: 1) losses across only the heat exchanger with no connecting piping to or from the storage tank; 2) losses across the heat exchanger and the piping required for connection to the tank without instrumentation; 3) losses across the heat exchanger, piping required for connection, and additional piping required for full instrumentation; 4) losses across the heat exchanger and only the piping required for instrumentation, for comparison with measured values. Table 1 presents the types and numbers of loss coefficients that were considered for each case.

<table>
<thead>
<tr>
<th>Case</th>
<th>Sharp edge exit, ( K=0.5 )</th>
<th>Sharp edge entrance, ( K=1.0 )</th>
<th>90° Bend, ( K=0.5 )</th>
<th>45° Bend, ( K=0.25 )</th>
<th>Slight Expansion, ( K=0.15 )</th>
<th>Slight contraction, ( K=2 )</th>
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**Results**

A comparison of the measured and predicted shear pressure drops as a function of mass flow rate is shown in Figure 3. The measured value is within 17% of the predicted value for shear pressure drop. The difference can be nearly accounted for by the uncertainty of the instrumentation. Differences could be a result of inaccurate estimation of minor losses in the heat exchanger or connecting piping. The heating tests indicate that any secondary flows that may exist inside the shell do not significantly affect the pressure drop through the heat exchanger at either heating level.
Figure 3. Predicted versus measured shear pressure drop for heating and no heating conditions.

Figure 4 presents a comparison of the shear losses in each component. The data show that most of the shear losses occur in the connecting piping, inlet, and outlet to the heat exchanger shell. The losses in the shell (annulus) section and in the straight connecting piping account for a small percentage of the pressure drop in the heat exchanger. The solid line in Figure 4 was used in the predicted versus measured comparison.

The importance of reducing the shear losses in the connecting is demonstrated in Figure 4, especially as the flow rate increases. Using large diameter connecting piping and minimizing the number of bends will reduce losses. The location of the heat exchanger inlet and outlet ports should be optimized to reduce unnecessary losses in the loop. In our facility, losses due to instrumentation are insignificant.
Figure 4. Predicted sources of shear pressure loss in heat exchanger.

Paper Presentations

The paper "Comparison of Natural Convection Heat Exchangers for Solar Water Heating" will be presented at the ASES conference July 17, 1995.

Future Activity

More fundamental analysis of the thermosyphon heat exchanger is underway and modeling efforts are in the initial stages. We are currently working with two US manufacturers who are planning to market thermosyphon heat exchangers.