NATURAL CONVECTION HEAT EXCHANGERS FOR SOLAR WATER HEATING SYSTEMS

Technical Progress Report
May 15, 1996 to July 14, 1996

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NATURAL CONVECTION HEAT EXCHANGERS FOR SOLAR WATER HEATING SYSTEMS

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Jane H. Davidson
University of Minnesota

Reporting Period: May 15, 1996 - July 14, 1996

Project Personnel: Jane H. Davidson
Scott Dahl, Graduate Research Assistant

Project Objectives:

The goals of this project are 1) to develop guidelines for the design and use of thermosyphon side-arm heat exchangers 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.

The tasks for the project are as follows.

1) Develop a model of the thermal performance of thermosyphon heat exchangers in solar water heating applications. A test protocol will be developed which minimizes the number of tests required to adequately account for mixed convection effects. The TRNSYS component model will be fully integrated in a system component model and will use data acquired with the specified test protocol.

2) Conduct a fundamental study to establish friction and heat transfer correlations for conditions and geometries typical of thermosyphon heat exchangers in solar systems. Data will be obtained as a function of a buoyancy parameter based on Grashof and Reynolds numbers. The experimental domain will encompass the ranges expected in solar water heating systems.

Summary of Results

We have completed experimental analysis of three tube-in-shell thermosyphon heat exchangers and worked with solar system manufacturers to help them incorporate side-arm heat exchangers in systems. Under the conditions expected in a solar water heater, these devices operate in the mixed convection regime on the thermosyphon side. Consequently, future modeling and testing procedures must not assume that thermal performance is a forced convection phenomenon. Prior models based on experimental correlations of either effectiveness or overall-heat-transfer-coefficient-area-product (UA) with thermosyphon flow rate are inadequate. We have developed testing and modeling procedures that rely on a correlation of UA with the relevant mixed convection dimensionless parameters (Re, Gr, Pr).

Current Activities

Task 1
We have begun a collaborative effort with Steven Long of Florida Solar Energy Center to develop and evaluate a TRNSYS model of side-arm heat exchangers. We provided Mr. Long with test data for a two-pass, tube-in-shell heat exchanger proposed by a solar manufacturer as well as a flow chart of the required modification to TRNSYS. The flow chart is shown in Figure
1. The major modification in this model is simulation of the heat transfer process on the thermosyphon side of the heat exchanger. Instead of relying on experimental data for UA as a function of flow rate, the model uses an empirical correlation of UA with Prandtl, Reynolds and Grashof numbers:

\[
UA = 0.12 \text{Pr}^{0.43} \text{Re}^{0.13} \text{Gr}^{0.30} \quad \text{W/C},
\]

where Grashof number is based on inlet temperatures

\[
\text{Gr} = \frac{g\beta D_n^3 (T_{h,i} - T_{c,i})}{\nu^2}.
\]

Fluid properties are evaluated at \((T_{h,i} + T_{c,i})/2\).

**Task 2**

The experimental plan has been developed for the fundamental study of heat transfer in side-arm heat exchangers. The plan is attached.

**Future Activities**

A paper concerning the modeling efforts will be submitted for the 1996 ASES-ASME-SOLTECH meeting to be held April, 1997 in Washington DC. Work on Task 1 should be complete by December 1996.

Modification of the experimental facility is underway. These modifications will permit completion of the outlined experiments by the end of the 18 month contract.
Initial guess for thermosyphon flow rate \( m_t \)
Initial guess for collector outlet temp., \( T_{h,i} \)
Specify inlet temp. from tank, \( T_{c,i} \)
Specify collector flow rate, \( m_c \)

Calculate \( \text{Re}_t, \text{Gr}_{Dh,AT} \) on thermosyphon side using values for \( T_{h,i}, T_{c,i}, m_t \), and heat exchanger geometry

\[
m_t = \rho_A \frac{V_A}{D_t} = \frac{4A_t}{\pi D_t}
\]

**Fluid Properties at \( T = \frac{T_{h,i} + T_{c,i}}{2} \)**

\[
\text{Gr} = g \beta \Delta T D_h \frac{T_{h,i} - T_{c,i}}{D_h^{3/4}}
\]

Determine \( \text{UA} \) utilizing experimentally obtained relationship for \( \text{UA} \)

\[\text{UA} = f(G_{\text{Gr}}, \text{Re}_t, \text{Re}_p, \text{Pr})\]

Calculate outlet temperatures \( T_{c,o} \) and \( T_{h,o} \) using:

\[
Q = \text{UA} \Delta T_{\text{LMTD}} = m_c \rho_{c,p} (T_{h,i} - T_{c,i}) = m_c \rho_{c,p} (T_{c,o} - T_{c,i})
\]

\[
\Delta T_{\text{LMTD}} = \frac{m_c \rho_{c,p} (T_{h,i} - T_{c,i})}{\ln((T_{h,i} - T_{c,i})/(T_{c,o} - T_{c,i}))}
\]

Calculate shear pressure drop across heat exchanger from pressure balance around thermosyphon loop using \( T_{h,i}, T_{h,o}, T_{c,i}, T_{c,o}, \) and \( m_t \).

*Assume linear temperature profile in heat exchanger as 1st approximation

Compare calculated pressure drop with experimentally obtained relationship for heat exchanger pressure drop to obtain \( m_t \)

\[f_{\text{Re}} = f(G_{\text{Gr}}, \text{Re}_t, \text{Re}_p, \text{Pr}) \text{ or } \Delta P = f(G_{\text{Gr}}, \text{Re}_t, \text{Re}_p, \text{Pr})\]

Does new value of \( m_t \) match initial guess

No

New guess for \( m_t \)

Yes

Collector Loop Energy Balance

Using calculated value of \( T_{h,o} \) and \( Q_u \) determine \( T_{h,i} \)

\[
Q_u = m_c \rho_{c,p} (T_{h,i} - T_{h,o}) = f(G_{\text{Gr}}, T_{h,o}, T_{\text{amb}}, F_R(\tau_{\alpha}), F_R U_L)
\]

Does new value for \( T_{h,i} \) match previous value for \( T_{h,i} \)

No

Yes

Done

Figure 1. Flow chart of the TRNSYS simulation of systems with a side-arm heat exchanger.
Objective

The purpose of this study is to understand the effects of natural convection on the thermal and hydraulic performance of thermosyphon heat exchangers used in solar water heating systems. The influence of three parameters will be investigated: the Reynolds and Grashof numbers on the thermosyphon flow side of the heat exchanger, and the flow rate of the fluid on the forced flow side of the heat exchanger. Laminar mixed convection data will be obtained for three tube-in-shell heat exchanger designs for which no data currently exist. Four, seven, and nine tube-in-shell heat exchangers will be studied. Detailed temperature, flow rate, and pressure measurements will be obtained for each design. The measurements will allow for the determination of the following quantities on the thermosyphon side of each heat exchanger:

1. Vertical temperature distribution
2. Local heat transfer coefficients
3. Average heat transfer coefficient
4. Heat transfer coefficient-area product (UA)
5. Total shear pressure drop and friction coefficient.

Correlations will be developed for the heat transfer and friction coefficients which consider forced convection, natural convection, and fluid property effects.

In addition to the experimental work, a semi-empirical model will be developed to predict the performance of thermosyphon heat exchangers in solar water heaters. Previous analyses assumed forced flow correlations alone can be used to predict the performance of thermosyphon heat exchangers (e.g., Avina (1994); Bergelt et al. (1993); Fraser (1992); Fraser et al. (1992); Fraser et al. (1993); and Parent et al. (1990)). Natural convection effects were neglected or ignored.

A testing protocol for thermosyphon heat exchangers will also be conceived which minimizes the number of tests while still providing the necessary data to implement the predictive model.

Introduction

The performance of a thermosyphon heat exchangers in solar water heating systems is affected by the following operating conditions and parameters:

A) Heat exchanger geometry (e.g. number of tubes, spacing of tubes, and heat transfer area)
B) Heat exchanger flow configuration (e.g. counterflow or parallel flow)
C) Collector side flow rate
D) Thermosyphon side flow rate
E) Natural convection on both the collector and thermosyphon side
F) Storage tank stratification
G) Fluid property effects.

Most experimental studies involving fluid flow through rod bundles have focused on heat transfer at high Reynolds numbers (Re>6000). In this regime, the forced convection dominates over the effects of natural convection. However, as the Reynolds number decreases the influence of natural convection on the friction and heat transfer coefficients becomes more apparent. Both the heat transfer and friction coefficients increase when natural convection effects are significant.

In previous studies involving mixed convection inside circular tubes, annular channels, and tube bundles, the heat transfer and pressure drop have been correlated as a function of Gr/Re^5. In circular and annular passages, mixed convection begins to affect pressure drop and heat transfer when the value of Gr/Re is in the range of 300 to 1000. The limited studies in rod bundles have shown a mixed convection begins to significantly affect heat transfer at Gr/Re^5 values greater than 2. The
values of $Gr/Re$ and $Gr/Re^2$ on the thermosyphon side of the heat exchanger are expected to range from 384 to $4.8 \times 10^6$ and from 0.3 to $1.92 \times 10^5$ in this study, respectively.

Few workers have investigated the heat transfer and friction coefficients in the laminar mixed convection flow regime. El-Genk et al. (1990), El-Genk et al. (1992), El-Genk et al. (1993), and Kim and El-Genk (1989) have collected and correlated heat transfer data for forced turbulent, forced laminar, combined and natural convection of water in uniformly heated seven rod triangular arrays and nine rod square arrays with pitch to diameter ratios ($P/D$) of 1.25, 1.38, and 1.5. The total mixed convection Nusselt number was expressed in a form first proposed by Churchill (1977), which combines the natural convection and forced convection components of the flow according to Eq. (1). No pressure drop data were obtained in these studies.

$$Nu_t = Nu_f + Nu_N$$  \hspace{1cm} (1)

Experimental studies using water naturally circulating through a triangular array seven rod bundle and a twenty-one bundle were conducted by Gruszczynski and Viskanta (1983) and Hallinan and Viskanta (1985), respectively. Heat transfer correlations were expressed in terms of Reynolds, Grashof, and Prandtl numbers.

An experimental study of the performance of coil-in-shell heat exchangers in solar water heating system by Allen and Ajele (1994) yielded heat transfer correlations in terms of the Rayleigh number. The influences of forced convection were not considered in their analysis.

Pressure drop data in rod bundles under mixed convection are very limited. A predictive model and experimental pressure drop data for a 19-rod bundle in a hexagon array with a $P/D$ of 1.25 are presented by Suh et al. (1989a) and Suh et al. (1989b).

The present experimental study will add to the limited data base of heat transfer and pressure drop coefficients for three rod bundle configurations. Heat transfer and friction coefficients for a 4, 7, and 9 rod bundle configurations will be obtained for thermosyphon flows with Reynolds numbers less than 1500. The Grashof number is expected to range up to $1.2 \times 10^8$. These values may prove to be an important contribution in many engineering applications, including design and operation of other tube-in-shell heat exchanger designs and operation and safety of nuclear reactors. The primary objective is to utilize the data to develop a model and testing protocol for thermosyphon heat exchangers used in solar water heating applications.

**Experimental Apparatus**

Table 1 presents a description of the three multiple tube-in-shell heat exchanger configurations to be studied. The thermosyphon (or mixed convection) flow is on the shell side in all cases. Schematics of each heat exchanger cross-section are presented in Figure 1. The length of each heat exchanger is 0.889 m (~35 inches). The three geometries were chosen because they are representative of the type of heat exchangers that are expected to be used in solar water heating applications and no heat transfer or pressure drop data exist for flow in the mixed convection regime.

Table 1. Geometric description of the heat exchanger designs to be tested.

<table>
<thead>
<tr>
<th>Geometry</th>
<th>Tube O.D. (m)</th>
<th>Shell I.D. (m)</th>
<th># tubes</th>
<th>Heat Transfer Area (m$^2$)</th>
<th>Flow Area (m$^2$)</th>
<th>$P_{\text{wetted}}$ (m)</th>
<th>$D_h$ (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 rod 2x2 square array</td>
<td>0.0159</td>
<td>0.0504</td>
<td>4</td>
<td>0.1776</td>
<td>1.201E-03</td>
<td>0.3581</td>
<td>0.0134</td>
</tr>
<tr>
<td>7 rod concentric circle</td>
<td>0.0159</td>
<td>0.0749</td>
<td>7</td>
<td>0.3108</td>
<td>3.016E-03</td>
<td>0.5849</td>
<td>0.0206</td>
</tr>
<tr>
<td>9 rod 3x3 square array</td>
<td>0.0159</td>
<td>0.0749</td>
<td>9</td>
<td>0.3996</td>
<td>2.619E-03</td>
<td>0.6848</td>
<td>0.0153</td>
</tr>
</tbody>
</table>
Figure 1. Schematic of the three heat exchanger designs under investigation. A) 4 rod, 2x2 square array \( P/D = 1.2 \), B) 7 rod, concentric circle array, C) 9 rod, 3x3 square array, \( P/D = 1.2 \).

A schematic of the shell and tube assembly is presented in Figure 2. The heat exchanger assembly consists of brass tubes inside a copper shell. Two brass flanges on each end of the copper shell hold the brass tubes and shell in place. The flanges provide a means to seal the heat exchanger by sandwiching the entire assembly together using four threaded rods that run the length of the heat exchanger. The brass rods fit inside holes bored into brass inserts which are sealed in place using a static O-ring seal. The O-ring seal allows the brass rods to thermally expand in the axial direction, which insures the rods remain vertical and parallel at all operating temperatures.

Figure 2. Schematic of the tube-in-shell assembly.
Two types of tests are planned for each geometry. In the first type of test, electrically heated rods will be inserted into the brass tubes to provide energy to water circulating on the shell side (thermosyphon side) of the heat exchanger. It will be assumed that a constant heat flux boundary condition will exist on the tube walls. In the second type of test, energy to the water on the thermosyphon side will be provided by circulating a hot mixture of ethylene glycol and water inside the tubes. The conditions of the second test mimic those found in solar water heating systems.

The heat exchangers will be tested in the facility shown in Figure 3. The test facility consists of two circuits: a loop in which a 50/50 mixture of ethylene glycol and water is heated with an electric boiler and a cold water supply system. The mains water supply is controlled to a specified temperature (±1 °C) using a computer controlled cold water supply system consisting of two 189 liter tanks with four independently controlled 3900 W (@ 208 VAC) electric heating elements. All connecting piping in the system is insulated (k = 0.040 W/(m·°C)).

![Figure 3. Schematic of test facility.](image)

The ethylene glycol mixture is heated using a 13 kW in-line electric boiler. The boiler is controlled using a microprocessor-based controller that is capable of maintaining either a constant outlet temperature or constant temperature difference (constant heat flux). Temperatures at the boiler inlet and outlet are measured with platinum RTDs (±(0.15 °C + 0.2% of reading)). Temperature control is within ±0.5 °C. Flow rate is controlled by adjusting a globe valve located downstream of a 30 W centrifugal pump.

In the set of tests where the resistive heater cartridges are used, the ethylene glycol mixture loop will not be used. Instead, the heat exchanger will be disconnected from the glycol loop and then the heating rods will be placed inside the tubes. A Variac (voltage regulator) will be used to regulate the power supplied to the resistive heater rods. The electric power supplied will be measured with a Watt transducer to determine the total and local heat flux in each design.

Measured values include flow rates on both sides of the heat exchanger, temperature differences across the heat exchanger, vertical temperature distribution in the storage tank connected to the heat exchanger, vertical water temperature profile in the heat exchanger determined from thermocouple measurements along the shell wall and with thermocouple probes inserted into the fluid, axial wall temperatures of the brass tubes inside the heat exchanger shell, and pressure drop on the
thermosyphon side of the heat exchanger. Temperatures are measured using type T thermocouples (±0.5 °C), represented by solid dots in Figures 3 and 4. Five junction thermopiles (±(0.5% of reading + 0.05 °C)) are used to measure differential temperatures at the heat exchanger inlets and outlets.

The test facility is monitored and controlled using a data acquisition/control program written in C++ programming language, a channel scanner, and a multimeter. Communication between the PC, 6 1/2 digit multimeter, and a channel scanner takes place over an IEEE interface bus. Communications between the PC and a power supply controller for the electric boiler takes place using a RS-232 interface.

Flow Rate Measurements
Thermosyphon side flow rate is measured using a Coriolis mass flow meter (±0.2%+0.001kg/s) while the forced fluid flow (ethylene glycol mixture) is measured using a turbine flow meter (±0.25% reading). Both flow meters are calibrated before the testing protocol begins. Two calibrated venturi flow meters are used to calibrate the turbine and Coriolis flow meters. One venturi has been calibrated in water over a range of 0.2 to 1.0 GPM, while the second venturi has been calibrated from 1.2 to 4.0 GPM.

Pressure Drop Measurements
Pressure drop on the thermosyphon side of the heat exchanger is measured across two sections of the heat exchanger shell using a differential pressure transmitter (accuracy = ±0.075% FS) with a range of 0 to 125 Pa (0 - 0.5 "H2O). Pressure drop between the inlet and outlet and across a 50 cm length of the rod bundle section on the shell side are measured. Figure 4 shows the locations of the pressure taps and temperature measurements on the heat exchanger. The pressure drop across the inlets is of practical importance in developing a predictive model for thermosyphon heat exchangers. The pressure drop measurement across the tube bundle section will provide more fundamental information about the apparent friction coefficient for each bundle configuration.

Figure 4. Locations of temperature and pressure measuring points on each heat exchanger shell.
Measuring the shear pressure drop in vertical passages is complicated because the hydrostatic pressure in the heat exchanger and connecting pressure lines must be considered, Eq. (2). The actual pressure drop due to shear, $\Delta P_s$, could be different from the measured pressure drop by the pressure transmitter, $\Delta P_m$, if the temperature distributions in the vertical sections of heat exchanger and connecting lines are not considered. To aid the measurement of the shear pressure drop, a 3.2 mm (1/8") copper tube is soldered to the exterior of the copper heat exchanger shell to utilize a pressure transmitter with a small range and high accuracy. Figure 5 shows sketch of how the pressure drop measurements will be made. It is hoped the temperature difference between the fluid bulk temperature and the measured wall temperature is small (within thermocouple measurement accuracy) so the last two terms in Eq. (2) cancel. The temperature difference between the shell wall and the fluid bulk temperature was found to be within 0.5°C in previous experiments on a well insulated tube-in-shell heat exchanger. If this temperature difference is 0.5°C, the error in the measured shear pressure drop would be 1.8 Pa (0.007 inches H_2O @ 20°C) on a heat exchanger that is 1 meter in length.

$$\Delta P_s = \Delta P_m + \int_0^H \rho(z) g dz \bigg|_{\text{tube}} - \int_0^H \rho(z) g dz \bigg|_{\text{HX}}$$

(2)

Figure 5. Schematic showing the shear pressure drop measurement method.

**Temperature Measurements**

Temperature measurements are needed at several sections of the heat exchanger in order to determine the local and average heat transfer coefficients on the mixed convection side of the heat exchanger. The locations of the temperature measurements are shown in Figure 4. The local tube wall and bulk temperatures are required to evaluate the local heat transfer coefficients.

The local wall temperature of the brass tubes will be determined using 36 gauge type-T thermocouple probes inserted into grooves along the tube walls. The tubes are brass and have a wall thickness of 3.2 mm (1/8"). Four grooves are machined longitudinally into four of the brass tubes at 90° rotations around the tube. Thermocouples are equally spaced at ten axial locations using four instrumented tubes as shown in Figure 6. Each groove is 1.02 mm deep by 1.59 mm wide (0.040" deep by 0.0625" wide). The thermocouple wire is seated into each groove using a high temperature, high thermally conductive epoxy. Once cured, the epoxy is sanded and filed down to maintain the circular cross section of each rod. Perturbation errors are expected to be minimal because of the small dimension thermocouple wire, groove size, and the incorporation of highly conductive filler material around each probe. The thermocouple probes encompass approximately 5% of the tube exterior surface area. Figure 7 presents an enlarged, scaled drawing of the thermocouples placed in the brass tubes.
Figure 6. Schematic showing position and axial locations of thermocouples on brass tubes to measure tube wall temperature. (Lengths are in units of cm and are in reference to the tube end.)

Figure 7. Enlarged view of the thermocouple placements on the brass tubes.
By having a total of sixteen temperature sensors on the brass tubes, duplicate temperature measurements can be taken at six different axial locations. Possible uses of the duplicate axial temperature measurements include the investigation of symmetry inside the heat exchanger shell, that is, variations in the temperature of each tube inside the shell can be observed. The temperature of a tube facing another tube can also be compared to a tube wall facing the shell of the heat exchanger.

The temperature of the fluid inside the shell will be measured at five axial locations using small diameter (0.51 mm or 0.020") thermocouple probes inserted through the shell. The probes can be traversed across the diameter of the heat exchanger shell to determine the average fluid temperature. Temperatures on the heat exchanger shell will also be measured using thermocouples attached to the outside surface with high conductive epoxy. The average temperature indicated by the thermocouple probe will be compared to the shell wall temperature at each axial location. It is hoped that the temperatures indicated on the shell can be directly related to the temperatures indicated by each of the probes. Previous experiments on a particular heat exchanger design showed that a difference of only 0.1°C to 0.4°C was observed at each axial location. The difference was within the accuracy of the thermocouple probes, ±0.5°C. The heat exchanger shell was well insulated in the previous testing and will be again in the present study.

Five junction thermopiles (±(0.5% of reading + 0.05°C)) will be used to measure temperature differences across the heat exchanger inlets and outlets and thermocouples with an accuracy of (±0.5°C) will be used to measure various system temperatures.

Power Measurement

The electrical power will be measured using a Watt transducer and a current transformer, which has a range of 0 to 4500 Watts. The accuracy of the instrument is ±1.25% of the reading.

Experimental Procedure

The influence of three parameters on the performance of the thermosyphon heat exchangers will be investigated in this study. They include the Reynolds number and Grashof number on the thermosyphon flow side of the heat exchanger and the forced fluid flow rate. The third factor, collector fluid flow rate is not relevant in the tests involving the electrically heated tubes.

A full factorial design is planned to investigate the effect of each parameter and any interaction effects that may play a role in the performance of the heat thermosyphon heat exchanger designs. Figure 8 demonstrates the range of parameter values that will be covered in this study. Their ranges are representative of the operating conditions expected in solar water heating systems and are limited by the operational limits of the experimental facility.

Table 2 shows the coded experiments to be conducted. A minimum of eight experiments is needed to investigate the effects of the three parameters Gr, Re, and m_c when the collector fluid is circulated through the brass tubes. Each condition will be repeated to test for repeatability.

Only two parameters are important in the experiments where electric heating rods are used, Re and Gr. The Grashof number will be based on the heat flux instead of a temperature difference in the tests involving electric heating. The effect of the inlet water temperature will be also be investigated in this set of experiments. Table 3 shows the coded experiments required for the experiments involving the electrically heated tubes.

The range of operating conditions that the present study will encompass will yield results in the mixed and forced convection flow regimes. Results will be obtained near the purely natural convection regime when the Reynolds number is very low but it will not be pure laminar natural convection.
Figure 8. Range of parameters to be tested in the present study.

Table 2 Two-level factorial design coding for experiments using collector fluid flow.

<table>
<thead>
<tr>
<th>Experiment #</th>
<th>$\Delta T$ (Gn)</th>
<th>Ret</th>
<th>$m_c$ (Rec)</th>
<th>Experimental Coding</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>25</td>
<td>0.5</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>25</td>
<td>0.5</td>
<td>+</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>1300</td>
<td>0.5</td>
<td>-</td>
<td>+</td>
</tr>
<tr>
<td>4</td>
<td>1300</td>
<td>0.5</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>5</td>
<td>25</td>
<td>2</td>
<td>-</td>
<td>+</td>
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<td>-</td>
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<td>+</td>
</tr>
<tr>
<td>9</td>
<td>60</td>
<td>650</td>
<td>1.25</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>60</td>
<td>650</td>
<td>1.25</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>60</td>
<td>650</td>
<td>1.25</td>
<td></td>
</tr>
</tbody>
</table>

Experiments will to be conducted in a random order. Perform 2 or tests at each experimental condition to check for repeatability. Analyze for significant effects and any interaction effects Conduct extensive tests over the Gr and Re operating range.
Once the two-level factorial tests are completed, additional data will be obtained at several intermediate conditions to fully investigate the shape of the operating surface based on the prescribed influential parameters. That is, several data points will be taken such that correlations can be developed which adequately characterize the performance of the heat exchanger designs.

**Calculation of Local and Average Heat Transfer Coefficients**

The local heat transfer coefficients on the thermostyphon side of the heat exchanger can be determined using the measured wall temperatures, bulk temperatures, and thermostyphon flow rate. The local heat transfer coefficient is determined using

\[ Q = hA(T_w - T_b) \]  \hspace{1cm} (3)

where the heat transferred, \( Q \), can be determined using

\[ Q = mc_p(T_{b2} - T_{b1}). \]  \hspace{1cm} (4)

\( T_{b2} \) and \( T_{b1} \) are the bulk temperatures at axial locations on either side of the point of interest. The mass flow rate will be measured and \( c_p \) is constant for water.

The average heat transfer coefficient on the thermostyphon side of the heat exchanger will be calculated using three methods. The first method involves integrating the local heat transfer coefficient over the length of the heat exchanger, Eq. (5). The second method utilizes the average wall and bulk temperatures, Eq. (6). Both of these methods should yield approximately the same value.

\[ \bar{h} = \frac{1}{L} \int_0^L \! h \, dz \]  \hspace{1cm} (5)

\[ Q = \bar{h}A(T_{\text{avg.wall}} - T_{\text{avg.bulk}}) \]  \hspace{1cm} (6)

The Nusselt number can then be determined.
The third method involves the calculation of the UA product for the heat exchanger, which will be calculated at each condition using

\[ Q = UA \Delta T_{LMTD} \]

where

\[ \Delta T_{LMTD} = \frac{(T_{h,o} - T_{c,i}) - (T_{h,i} - T_{c,o})}{\ln\left(\frac{T_{h,o} - T_{c,i}}{T_{h,i} - T_{c,o}}\right)} \]

The UA value will be used along with published forced flow heat transfer correlations for \( h_f \) on the forced side of the heat exchanger to determine the average heat transfer coefficient, \( h_a \), on the thermosyphon side of the heat exchanger. This method is expected to be the least reliable since published correlations will be assumed to apply to the forced flow side of the heat exchanger. A comparison of the third method with the previous two methods will be an interesting exercise.

The apparent friction coefficient, \( fRe \), will be calculated using the measured pressure drop across the heat exchanger and Eq. (11).

\[ \Delta P_s = \frac{2(fRe)\mu VL}{D_h^2} \]

The heat transfer coefficients on the thermosyphon side of the heat exchanger obtained using the three methods will be compared.

The apparent friction coefficient, \( fRe \), will be calculated using the measured pressure drop across the heat exchanger and Eq. (11).

The experimental uncertainties associated with the temperature sensors, pressure transmitters, flow meters, and the Watt transducer will be taken into account using the approach established by Kline and McClintock (1953).

**Expected Results**

It is expected that the Nusselt number (and/or UA) and the friction factor, \( fRe \), (and/or \( \Delta P_s \)) can be correlated as a function of \( Re \), Gr, and Pr for each forced flow rate. The statistical package SYSTAT will be used to develop the predictive correlations. Perhaps the forced fluid flow rate can also be incorporated into the correlations to describe the performance of each heat exchanger design. Heat transfer correlations in the form of Eq. (1) will be attempted. The experimentally obtained mixed convection correlations will be compared to forced flow correlations to demonstrate the influence of natural convection.

Some insight will be gained about the axial temperature profile in the heat exchanger under a variety of conditions. The temperature profile is required to calculate the driving force for
thermosyphon flow. The possibility of using surface mounted temperature sensors on the heat exchanger shell to determine the fluid temperature inside the shell will be evaluated. Making surface measurements to estimate the temperature profile in each heat exchanger will greatly simplify the testing of new heat exchanger designs and eliminate the need to assume or guess a temperature profile.

The thermal and hydraulic correlations will be incorporated into a model to predict the performance of the thermosyphon heat exchangers. The model will be developed for incorporation into TRNSYS, a transient thermal system simulation computer program. Additional experiments will be conducted to validate the established predictive model. Once the model has been validated, a testing protocol will be established which provides the required information for the predictive model with a minimal number of data points.

References


