FURTHER STUDIES IN FILMWISE
CONDENSATION OF STEAM ON
HORIZONTAL FINNED TUBES

by

Keith Andrew Swensen

March, 1992

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### Abstract
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The present work uses an instrumented smooth tube to obtain accurate inside heat-transfer correlations both with and without inserts and uses these to obtain accurate outside coefficients for a family of uninstrumented finned tubes with a view to finding an optimum fin spacing for steam condensation.

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- Filmwise condensation, integral finned tubes, vapor velocity

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Further Studies in Filmwise Condensation of Steam on Horizontal Finned Tubes

by

Keith Andrew Swensen
Lieutenant, United States Navy
B.S., Brigham Young University, 1985

Submitted in partial fulfillment of the requirements for the degree of

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ABSTRACT

Over the years, there has been significant variation in the filmwise steam condensation data at NPS on horizontal low-integral finned tubes. With a view to increasing the accuracy of the data, inserts were used inside the tubes to reduce inside thermal resistance; however, significant discrepancies then occurred in the calculated outside coefficient when compared to data taken without an insert. These discrepancies arose due to the data reduction technique which assumes a known inside heat-transfer resistance and subtracts this from a measured overall resistance. If the assumed value on the inside is inaccurate, then the outside value is equally inaccurate.

The present work uses an instrumented smooth tube to obtain accurate inside heat-transfer correlations both with and without inserts and uses these to obtain accurate outside coefficients for a family of uninstrumented finned tubes with a view to finding an optimum fin spacing for steam condensation.
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NOMENCLATURE

\begin{itemize}
  \item \(A_i\) effective inside surface area \((m^2)\)
  \item \(A_o\) effective outside surface area \((m^2)\)
  \item \(C_i\) Sieder-Tate leading coefficient
  \item \(C_f\) mass flow rate correction factor
  \item \(c_p\) specific heat at constant pressure \((J/kg \ K)\)
  \item \(D_i\) inside tube diameter \((m)\)
  \item \(D_o\) outside tube diameter \((m)\)
  \item \(D_r\) finned tube outside root diameter \((m)\)
  \item \(g\) gravitational constant \((9.81 \text{ m/s}^2)\)
  \item \(h_{fs}\) specific enthalpy of vaporization \((J/kg)\)
  \item \(h_i\) inside heat transfer coefficient \((W/m^2K)\)
  \item \(h_o\) outside heat transfer coefficient \((W/m^2K)\)
  \item \(k_c\) thermal conductivity of cooling water \((W/mK)\)
  \item \(k_f\) condensate film thermal conductivity \((W/mK)\)
  \item \(k_m\) thermal conductivity of metal tube \((W/mK)\)
  \item \(L\) length of exposed tube \((m)\)
  \item \(L_{MTD}\) log mean temperature difference \((K)\)
  \item \(L_1\) length of inlet portion of tube \((m)\)
  \item \(L_2\) length of outlet portion of tube \((m)\)
\end{itemize}
\( m_{\text{act}} \) Corrected mass flow rate
\( m_{\text{calc}} \) Computed mass flow rate
\( m \) mass flow rate (kg/s)
\( N_u \) Nusselt number
\( P_{\text{sat}} \) saturation pressure (Pa)
\( P_r \) Prandtl number
\( Q \) heat transfer rate (W)
\( q \) heat flux (W/m\(^2\))
\( R_e \) Reynolds number
\( R_{e_{2p}} \) two phase Reynolds number
\( R_i \) inside resistance (K/W)
\( R_o \) outside resistance (K/W)
\( R_w \) wall resistance (m\(^2\)K/W)
\( \Delta T_f \) temperature across condensate film (K)
\( T_b \) mean bulk fluid temperature (K)
\( T_m \) mean coolant film temperature (K)
\( T_w \) mean inner tube wall temperature (K)
\( T_{\text{sat}} \) vapor saturation temperature (K)
\( T_1 \) cooling water inlet temperature (K)
\( T_2 \) cooling water outlet temperature (K)
\( U_o \) overall heat transfer coefficient (W/m\(^2\)K)
\( U_w \) vapor velocity (m/s)
$V$  cooling water velocity (m/s)

$\alpha$  dimensionless coefficient

$\epsilon_{\Delta T}$  enhancement ratio based on the constant $\Delta T$

$\epsilon_q$  enhancement ratio based on constant $q$

$\mu_c$  dynamic viscosity of cooling water at bulk temperature (N·s/m$^2$)

$\mu_f$  dynamic viscosity of condensate film (N·s/m$^2$)

$\mu_w$  dynamic viscosity of cooling water at mean inner tube wall temperature (N·s/m$^2$)

$\rho_f$  condensate film density (kg/m$^3$)

$\rho_v$  vapor density (kg/m$^3$)

$\eta$  surface efficiency
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I. INTRODUCTION

A. BACKGROUND

A reduction in size and weight of all types of heat exchangers aboard Naval vessels will allow more efficient use of space. The benefits might include greater equipment accessibility for maintenance or greater heat exchanger capacity (without a corresponding increase in size and weight) with a corresponding increase in fuel efficiency.

For the past ten years, the Naval Postgraduate School in collaboration with the David Taylor Research Center and the National Science Foundation has conducted research that is directed at the development of smaller, more efficient steam condensers. Improved designs can result in significant space savings, always a primary concern on Naval vessels, especially submarines.

Uncertainties in past data using steam were apparently due to the lack of detailed information about the inside heat-transfer correlations used during the data reduction process. Previously, the standard Sieder-Tate correlation was assumed to be valid for the inside heat-transfer coefficient, but it may not be the best correlation to use with the particular test arrangement used in this research program.

A large amount of enhanced condensation data has been collected in previous studies at the Naval Postgraduate School on more than 90 different condenser tubes of varying fin height, fin spacing, and tube material, with a view to finding an
optimum fin geometry for both steam and refrigerant condensation. Satisfactory results have been obtained with refrigerant data using R-113. However, some troublesome questions of possible contamination and instrument inaccuracy still remain with the steam data. It is felt that in order to address these questions, a fundamental evaluation of the heat transfer apparatus on which this data was collected and of the data reduction process was considered appropriate. Current data, taken on a carefully cleaned and calibrated apparatus, could be compared to previously recorded data and a determination as to its validity and reproducibility could be made. In addition, a thorough evaluation of the best inside heat-transfer correlation would lead to more reliable steam condensation results.

B. CONDENSATION

Condensation occurs when a vapor is cooled below its saturation temperature, or when a vapor/gas mixture is cooled below its dew point. Surface condensation occurs in condensers when a cooled surface (kept at a temperature below the saturation temperature of the vapor) contacts the vapor. The vapor molecules that contact such a surface stick to that surface and condense into liquid molecules. Condensation may occur in one of the following modes: filmwise, dropwise, or mixed mode (a combination of filmwise and dropwise) condensation. In the filmwise mode, the liquid wets the cold surface to form a continuous film. If the liquid does not wet the surface but instead forms discrete drops on the cold surface, dropwise or mixed
mode condensation will occur and is often caused by some form of contamination. [Ref. 1]

The condensate forming on the tube surface offers a resistance to heat transfer between the vapor and the surface, which increases with the thickness of the liquid layer. Even though dropwise condensation results in much larger heat-transfer coefficients than filmwise condensation, it is difficult to maintain a stable dropwise condition over prolonged periods. Therefore, in most cases condenser design calculations are based on the assumption of filmwise condensation, resulting in lower heat-transfer coefficients and more conservative designs. [Ref. 1]

In a condenser, the coolant side, tube wall, and vapor side thermal resistances control the heat transfer rate from vapor to coolant. Also, for experimental work we always use clean tubes, but in real condensers tubeside fouling can play an important role in increasing the coolant side resistance. The magnitudes of these resistances depend on the fluid, tube geometry, and flow conditions on the vapor and coolant side. For steam condensation, it is the coolant side thermal resistance which tends to dominate. Methods to lower this inside resistance include the use of inserts or roped tubing to promote turbulence, thereby raising the convection heat-transfer coefficient. However, such modifications lead to increased pressure drop through the tubes, which must be compensated for by providing extra pumping capacity. Heat transfer through the tube wall is conductive and is fixed once tube thickness and material are selected. The vapor side resistance is due to the condensate film which forms on the outside of the tube. For filmwise condensation, the outside
resistance can be reduced by the addition of low integral fins. These have the effect of not only increasing the outside surface area of the tube, but also of thinning the condensate film around the fins due to surface tension forces. Too small a fin spacing may result in condensate flooding, whereas too large a fin spacing approaches the smooth tube case; there should be an optimum fin spacing somewhere in between these two extremes. Horizontal fin spacing is therefore of prime importance, and finding the optimum spacing is one of the objectives of this long-term research program.

C. CONDENSATION RESEARCH AT NAVAL POSTGRADUATE SCHOOL

The research effort at NPS has included the study of differing fin dimensions (i.e. fin height, fin width, fin spacing) on low-integral finned horizontal tubes. Experimentation has included the use of three different test fluids (steam, R-113, and ethylene glycol) under various operational conditions using a number of different tube diameters.

Van Petten [Ref. 2] provides a summary of research efforts at NPS through the end of 1988. Van Petten and subsequent researchers have analyzed small, medium, and large diameter finned tubes to find the optimum fin spacing for maximum heat-transfer enhancement of the fluids mentioned above. However, discrepancies found by Guttendorf [Ref. 3] in the data processing technique (modified Wilson plot), which resulted in different values of heat-transfer enhancement (for the same tube
under the same operating conditions depending on whether an insert was or was not used), have raised doubts about the accuracy of the inside heat-transfer correlation.

Rouk [Ref. 4] investigated the use of an optimization technique to predict the inside heat-transfer correlation. When the optimization effort proved unsuccessful, he next used the instrumented smooth tube data of Georgiadis [Ref. 5] to develop an inside heat transfer correlation, but could not find a correlation with sufficient accuracy based on previous data. He recommended that once an overhaul on the test apparatus was complete, an increase in data precision would allow the development of an accurate inside heat-transfer correlation. This work is a follow on effort to develop inside heat-transfer correlations which can predict the value of the inside heat-transfer coefficient with good accuracy under a variety of flow conditions. Once an inside correlation can be found, the object of this effort is to reprocess previous data and see if the discrepancies reported by former researchers for finned tubes on this apparatus can be rectified.

D. OBJECTIVES

The main objectives of this thesis were to:

1. Disassemble and meticulously clean the apparatus to eliminate any existing contamination with a view to eliminating dropwise condensation problems experienced in the past.

2. Carefully reassemble the apparatus using new gasket material, and make modifications to improve system performance.

3. Recalibrate all system instrumentation to ensure the greatest achievable accuracy.

5
4. Investigate the possibility of manufacturing large, medium, and small diameter instrumented smooth tubes.

5. Use the new instrumented tubes and the one existing medium diameter smooth instrumented tube (fabricated by Poole [Ref. 6]) to obtain accurate inside heat-transfer correlations for a number of insert types as well as the no insert condition.

6. Evaluate the accuracy of the currently used data processing technique (modified Wilson plot) using instrumented tube data.

7. Reprocess previous data using the new correlations with a view to comparing current and past smooth tube and finned tube data to provide continuity with previous studies.
II. LITERATURE SURVEY

A. INTRODUCTION

When a vapor condenses in the filmwise mode on a smooth horizontal tube it forms a thin continuous film of condensate on the surface of the tube. The condensate film thickens around the tube due to gravity. This condensate film provides a resistance to heat transfer which may be lowered through the use of fins. For quite some time, it was thought to be impractical to use finned tubes with high surface tension fluids such as water, due to condensate retention and flooding between the fins. However, a number of studies conducted on finned tubes using steam have shown that substantial heat-transfer enhancement may be achieved.

A significant amount of research at the Naval Postgraduate School and elsewhere has addressed the issue of optimum fin height, thickness, and spacing required for maximum heat transfer. Yau et al [Ref. 7] reported that "with an increase in fin density, up to a limit (this limit is not yet known in a generalized manner), the heat-transfer coefficient increases at a rate faster than the increase in the outside area due to the presence of fins. This additional enhancement is due to the thinning effect of the surface-tension forces on the condensate film. Unfortunately, surface-tension forces also adversely affect heat transfer by causing condensate to be retained between fins" [Ref. 8]. Katz et al [Ref. 9] also found that
on finned tubes the portion of the surface occupied by condensate is dependent upon the ratio of condensate surface-tension to density and the fin geometry.

Condensate retention and the behavior of the condensate film on the tube surface under various conditions are critical parameters in the heat transfer process on horizontal finned tubes. Several models have been developed to predict this behavior and the reader is referred to an extensive review of horizontal finned tube heat transfer by Marto [Ref. 10] for a more detailed coverage of the topic.

B. VAPOR SIDE CONSIDERATIONS

The filmwise condensation of vapor on a horizontal tube is a complex two-phase heat transfer process, for which a suitably complex model would be required to accurately predict heat transfer performance under all conditions.

In 1916, Nusselt [Ref. 11] set forth his theoretical work on the study of laminar filmwise condensation of a "stationary" vapor on a vertical or inclined plate and a horizontal tube. Nusselt's simplifying assumptions included the following [Ref. 12]:

1. Pure saturated vapor
2. Negligible vapor velocity \(U_\infty = 0\)
3. Heat transfer across the condensate film by conduction only
4. Laminar condensate flow governed only by gravitational and viscous forces
5. Condensate properties constant
6. Isothermal condensing surface
7. Negligible interface temperature drop
Nusselt's result for the mean heat-transfer coefficient for a horizontal tube was obtained:

\[ Nu = 0.728 \left[ \frac{k_f \rho_f (\rho_f - \rho_v) g h_{fg}}{\mu_f D_o \Delta T_f} \right]^{1/4} \]  \hspace{1cm} (2.1)

or

\[ Nu = 0.655 \left[ \frac{k_f \rho_f (\rho_f - \rho_v) g h_{fg}}{\mu_f D_o q} \right]^{1/3} \]  \hspace{1cm} (2.2)

where:

- \( Nu \) = mean Nusselt number
- \( k_f \) = thermal conductivity of condensate film (W/m \cdot k)
- \( \rho_f \) = condensate film density (kg/m\(^3\))
- \( \rho_v \) = vapor density (kg/m\(^3\))
- \( g \) = gravitational constant (9.81 m/s\(^2\))
- \( h_{fg} \) = specific enthalpy of vaporization (J/kg)
- \( \mu_f \) = dynamic viscosity of condensate film (N\cdot s/m\(^2\))
- \( D_o \) = outside tube diameter (m)
- \( \Delta T_f \) = average temperature difference across condensate film (K)
- \( q \) = heat flux based on outside area (Q/A\(_o\)) (W/m\(^2\))

Many workers have improved on Nusselt's theoretical analysis by accounting for some of the terms he neglected through his simplifying assumptions. However, equations (2.1) and (2.2) have been found to be remarkably accurate over a wide range of conditions for a stationary vapor. High vapor velocity can increase film condensation heat transfer substantially. This enhancement, which refers to the amount of heat transfer above or below the value predicted by the Nusselt analysis, is due to the effect of thinning the condensate film. However, vapor shear is the one
assumption which if applied can lead to significant increases in the heat-transfer coefficient.

The theoretical result of Shekriladze and Gomelauri (1966) [Ref. 13], who considered interfacial shear stress due to vapor velocity, is shown in equation (2.3).

\[
\frac{Nu}{Re_{2\phi}^{1/2}} = 0.64(1 + (1 + 1.69F)^{1/2})^{1/2}
\]  

(2.3)

where:

\begin{align*}
Nu &= \text{Nusselt number for the vapor side} \\
Re_{2\phi} &= \text{two phase Reynolds number, } (\rho_f U_\infty D/\mu_f)
\end{align*}

For steam condensation, the empirically derived correlation of Fujii et al [Ref. 14] is shown in equation (2.4).

\[
\frac{Nu}{Re_{2\phi}^{1/2}} = 0.96F^{1/5}
\]  

(2.4)

The Nusselt expression (equation (2.1) can be expressed in similar form:

\[
\frac{Nu}{Re_{2\phi}^{1/2}} = 0.728F^{1/4}
\]  

(2.5)

Whereas the vapor velocity, \( U_\infty \), cancels out in the Nusselt expression (stationary vapor assumption), the Fujii correlation includes the vapor velocity effect. Therefore we can expect equation (2.4) to more accurately predict steam-side heat
transfer coefficients for those cases where vapor velocity begins to have a significant impact.

For further review of basic theoretical studies on the subject of laminar film condensation on smooth tubes the reader is referred to Rose [Ref. 15].

C. COOLANT SIDE CONSIDERATIONS

For a turbulent flow regime inside a pipe (Re > 10,000), a number of coolant-side correlations have been used; many of these have taken the form:

\[ Nu = C_i Re^n Pr^m \]  

(2.6)

where:

\[ Nu \] = mean coolant Nusselt number for turbulent flow
\[ C_i \] = correlating coefficient
\[ Re \] = coolant Reynolds number
\[ Pr \] = coolant Prandtl number

The most common correlations with the same form as equation (2.6) are that of Dittus and Boelter (1930) [Ref. 16]:

\[ Nu = 0.023 Re^{0.8} Pr^{0.4} \]  

(2.7)

and Colburn (1933) [Ref. 17]:

\[ Nu = 0.023 Re^{0.8} Pr^{1/3} \]  

(2.8)
Sieder and Tate (1936) [Ref. 18] applied a correction factor to equation (2.8) to account for cases in which the bulk to inner wall temperature difference is large enough to cause substantial variations in coolant viscosity as follows:

\[ Nu = 0.027 Re^{0.8} Pr^{1/3} \left( \frac{\mu_c}{\mu_w} \right)^{0.14} \]  \hspace{1cm} (2.10)

where:

- \( \mu_c \) = coolant viscosity evaluated at mean bulk temperature \((N\cdot s/m^2)\)
- \( \mu_w \) = coolant viscosity evaluated at mean inner tube wall temperature \((N\cdot s/m^2)\)

The fluid properties in equations (2.7), (2.8), and (2.9) are evaluated at mean coolant bulk temperature \( T_{m,\text{bulk}} \):

\[ T_{m,\text{bulk}} = \frac{T_1 + T_2}{2} \]  \hspace{1cm} (2.11)

where:

- \( T_1 \) = tube coolant inlet temperature (K)
- \( T_2 \) = tube coolant outlet temperature (K)

Equations (2.7), (2.8) and (2.9) are valid for \( Re > 10^4 \) and \( 0.7 < Pr < 100 \), and were developed for long smooth pipes with no inserts [Ref. 12].

The use of inserts and the effect of bends close to the tube entrance region can affect the values of both the leading coefficient, \( C_i \), and Reynolds number exponent, \( m \), in equation (2.6). One of the major focuses of this study is to determine the values of \( C_i \) and \( m \) for the no insert, wire wrap insert, and Heatex insert cases.
Other well-known turbulent pipe flow heat-transfer correlations (i.e. Petukhov-Popov [Ref. 19], Sleicher-Rouse [Ref. 20], etc.) and the results of an ANL (Argonne National Laboratory) [Ref. 21] study which evaluated several such correlations for accuracy are reviewed in section VI C.
III. APPARATUS AND SYSTEM INSTRUMENTATION

A. SYSTEM OVERVIEW

The apparatus used for this research was basically the same as was used by Van Petten [Ref. 2] and Guttendorf [Ref. 3] with certain modifications. A system schematic is provided in Figure 1. Steam generated from distilled water in the .30 m diameter pyrex glass boiler using ten 4 kW, 440 V Watlow immersion heaters was the working medium for this set of experiments. From the boiler section the steam passed up through a reducing section and a 2.13 m straight length of pyrex glass piping, (ID of 0.15 m), it was then turned through 180 degrees using two 90 degree pyrex glass elbows, and then descended down a 1.52 m straight length of pyrex glass piping. The steam then entered the stainless steel test section containing the horizontally mounted condenser tube (see Figures 1 and 2); any steam not condensing there was condensed in the auxiliary condenser located just beneath the test section. The auxiliary condenser was constructed of a single copper coil mounted to a stainless steel baseplate enclosed within a pyrex glass condenser section. Coolant flow through the auxiliary condenser was used to control system pressure, and all condensed liquid was returned via the condenser baseplate drain to the boiler section by gravity.

Coolant for the auxiliary condenser was provided via a throttled water connection with associated flowmeter. Coolant flow through the single horizontal
Figure 1. Schematic of Single Tube Test Apparatus
Figure 2. Schematic of Test Section Insert
tube was provided by a separate system consisting of a sump tank with two centrifugal pumps connected in series. Coolant flow rate was measured by a carefully calibrated flowmeter. By varying coolant flow-rate through the single horizontal tube, the rate of steam condensation on the tube (and hence heat-transfer coefficient) could be varied.

Non-condensible gases were removed using the vacuum pump system shown in Figure 3. The condensing coil for this purge system, located in the sump tank, served to condense steam carried through the vacuum line during the purging process. The vacuum line took its suction from the base of the auxiliary condenser, the coolest spot in the apparatus and the place where non-condensible gases (i.e. air) were most likely to accumulate.

B. SYSTEM INSTRUMENTATION

The power to the 440 V heaters was controlled through a panel mounted potentiometer. A description of the power calculation for input into the data acquisition system can be found in Poole [Ref. 6].

System pressure was monitored in three ways:

1. A Setra model 204 pressure transducer
2. A Heise solid front pressure gauge (visual reading only)
3. System saturation pressure from vapor temperature measurement
Figure 3. Schematic of Purging System and Cooling Water Sump
System vapor temperature was monitored using both teflon, and metal sheathed type-T copper/constantan thermocouples located juxtaposed in the test section; this position was just upstream of the test condenser tube. Condensate temperature was also monitored using a teflon coated type-T copper/constantan thermocouple located on the condensate return line between the auxiliary condenser and boiler. Coolant temperature rise in the condenser tube was measured using four methods:

1. Two teflon coated type-T copper/constantan thermocouples
2. Two metal sheathed type-T copper/constantan thermocouples
3. Two Hewlett-Packard 2804A quartz crystal thermometers
4. A ten-junction teflon coated type-T copper/constantan thermopile

These were all placed at the inlet to and exit from the condenser tube; at the outlet, all thermocouples were placed just downstream of a coolant mixing chamber.

Two data reduction programs were used to collect and reduce data on this apparatus; "DRPINST", and "DRPKS". The instrumented tube constructed by Poole [Ref. 6] was used to determine an accurate inside heat-transfer correlation for inserts used; this instrumented tube contained six wall thermocouples. For the instrumented tube the appropriate calibration equations were accessed in the data acquisition program "DRPINST". For non-instrumented tubes, the data reduction program "DRPKS" was used. Fluid property equations used in the data reduction programs are given in Appendix A and calibrations were conducted for all system...
instrumentation (flowmeter, thermocouples, pressure transducer, etc.) and are included in Appendix B.

The data as monitored by the aforementioned system instrumentation was processed by an HP-3497A data acquisition system controlled by an HP-9826A computer provided with the correct data acquisition program. The raw data was processed and stored on computer disks. Program channel assignments are given in Table 1.

C. SYSTEM MODIFICATIONS

At the beginning of this investigation the apparatus was entirely disassembled to facilitate complete overhaul of the system. Modifications and details of assembly were as follows:

1. The apparatus was taken apart piece by piece, inspected, meticulously cleaned with a warm solution of Sparkleen biodegradable soap and subjected to a complete acetone rinse prior to reassembly.

2. A new pyrex glass riser section above the boiler was 0.31 m shorter than the previous section and allowed the addition of a new aluminum stand on which to place the heater baseplate. This new stand allowed much easier access to the 440 V heater wiring plus the adjustable legs allowed level adjustment of the entire apparatus, to ensure proper alignment of the single horizontal tube.

3. The two pyrex glass elbows were replaced.

4. Every gasket in the system was replaced (using Buna-N rubber) and, using a standard star torque pattern, all flanged joints were tightened to a final torqued of 30 inch-pounds (manufacturer recommended maximum torque...
<table>
<thead>
<tr>
<th>CHANNEL</th>
<th>QUANTITY MEASURED</th>
<th>APPLICABLE TYPE DESIGNATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>Vapor Temperature</td>
<td>T-57 (See Appendix B.1)</td>
</tr>
<tr>
<td>41</td>
<td>Vapor Temperature</td>
<td>T-56 (See Appendix B.1)</td>
</tr>
<tr>
<td>42</td>
<td>Room Temperature</td>
<td>T-58 (See Appendix B.1)</td>
</tr>
<tr>
<td>43</td>
<td>Tube Coolant in 1</td>
<td>T-58</td>
</tr>
<tr>
<td>44</td>
<td>Tube Coolant out 1</td>
<td>T-58</td>
</tr>
<tr>
<td>45</td>
<td>Tube Coolant in 2</td>
<td>T-55 (See Appendix B.1)</td>
</tr>
<tr>
<td>46</td>
<td>Tube Coolant out 2</td>
<td>T-55</td>
</tr>
<tr>
<td>47</td>
<td>Condensate Return</td>
<td>T-58</td>
</tr>
<tr>
<td>48</td>
<td>Instrumented Tube</td>
<td>T-57 or T-55 as applicable</td>
</tr>
<tr>
<td>49</td>
<td>Instrumented Tube</td>
<td>T-57 or T-55 as applicable</td>
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<tr>
<td>50</td>
<td>Instrumented Tube</td>
<td>T-57 or T-55 as applicable</td>
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<td>Instrumented Tube</td>
<td>T-57 or T-55 as applicable</td>
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<td>52</td>
<td>Instrumented Tube</td>
<td>T-57 or T-55 as applicable</td>
</tr>
<tr>
<td>53</td>
<td>Instrumented Tube</td>
<td>T-57 or T-55 as applicable</td>
</tr>
<tr>
<td>60</td>
<td>10 - Junction Thermopile</td>
<td>T-61 (See Appendix B.1)</td>
</tr>
<tr>
<td>61</td>
<td>Voltage</td>
<td>NA</td>
</tr>
<tr>
<td>62</td>
<td>Current</td>
<td>NA</td>
</tr>
<tr>
<td>64</td>
<td>Pressure Transducer</td>
<td>(See Appendix B.4)</td>
</tr>
</tbody>
</table>
specification was 60 inch-pounds). The bolts holding the flanges together should be checked for tightness on a regular basis, since the thermal cycling of the apparatus has been shown to cause loosening of nut and bolt assemblies.

5. The previous vacuum pump was replaced with a Gast model 2567-V108 vacuum pump which could draw vacuum to 130 mmHg with an installed check valve to prevent pump back spin when the pump was stopped. The new pump could draw vacuum much more rapidly, but could not operate at less than 130 mmHg. Remaining non-condensible gases were removed by flushing the system with steam. The steam flushing procedure for removal of non-condensible gases is given in the operating procedures section. Once the system is completely filled with steam, operating pressures well below 130 mmHg could be achieved utilizing the auxiliary condenser.

6. The double coil auxiliary condenser was replaced with a single coil used originally by Van Petten [Ref. 2]. The single coil was not coated with the special oxide coating used by Guttendorf [Ref. 3]. It was felt to be superior to the double coil in that the baseplate welds were of much higher quality and were preferred on the basis of vacuum tightness.

7. The aluminum side plates attached to the pyrex glass auxiliary condenser housing were replaced with new stainless steel side plates with penetrations for pressure bleed, vacuum line, and a pressure transducer. These three penetrations were fitted with screw threaded stainless steel connectors. The stainless steel connectors were heli-arc welded in place. Prior to this modification (completed 24 January 1992) a leak test conducted from 21 December 1991 to 2 January 1992 revealed a mean vacuum leak rate of \( -3.4 \) mmHg per day (see Figure C.1). A subsequent leak test conducted from 6 February 1992 to 19 February 1992 showed an improvement in the mean leak rate to \( -1.7 \) mmHg per day (see Figure C.2).

8. System instrument modifications included the addition of the Setra pressure transducer and the Heise pressure gauge, and the removal of the mercury manometer. All system instrumentation was recalibrated and the results incorporated into the data reduction programs.

9. Finally the apparatus was lagged with Halstead insulating foam to reduce heat loss as much as possible. The test section, which was left uncovered previously, was also lagged.
IV EQUIPMENT OPERATION AND EXPERIMENTAL PROCEDURE

A. SYSTEM STARTUP AND SHUTDOWN PROCEDURES

Startup of the system is accomplished in the following manner:

1. Ensure distilled water level in the boiler is 4 to 6 inches above the top of the heating elements. The boiler is filled by gravity drain via a hose connection from the distilled water tank to the boiler fill valve. Ensure the vent valve on the side of the auxiliary condenser is open when filling or draining the boiler. The boiler may be drained by removing the hose connection and opening the fill valve, which allows drainage into the trench directly beneath the boiler.

2. Once the boiler is filled to the appropriate level, shut the boiler fill valve and the distilled water tank valve.

3. Shut the system vent valve.

4. Turn on the data acquisition system, computer and printer. Load the appropriate program (DRPINST or DRPKS) and check for proper operation. Then check all thermocouple outputs, by stepping through the appropriate data acquisition system channels, to verify that all are registering ambient temperature.

5. Open the fill valve to the coolant water sump tank to a level such that the tank overflow drain box does not overflow (the valve is located between the boiler control panel and heat pipe apparatus).

6. Turn on the cooling water supply pumps and adjust the tube flow rate from 20% to 60% of the rotameter setting and check for leaks. Reset flow rate to desired level.

7. Open valves from tap water system to auxiliary condenser and adjust coolant flow rate to at least 30% and check for leaks. Reset flow rate to at least 10%.

8. Energize heaters and adjust voltage to approximately 50 volts (40 volts if the system is already at vacuum below 100 mmHg to limit the vibrational shock to the system from oversized vapor bubble formation). To energize the heaters there are three switches which must be placed in the on position. The first is located in power panel p5 located in the main hallway adjacent to the
lab and is labeled switch 3 / heater controller room 106. The second is the
heater load bank circuit breaker located on the side of the boiler control
panel. The third is the condensing rig boiler power switch on the front of the
boiler control panel. Increase the voltage gradually in 10 volt increments to
the desired level.

9. Turn on the vacuum pump and open the vacuum line valve. Allow the
vacuum pump to run until system pressure is below 3 psi, then shut the
vacuum line valve just prior to turning off the vacuum pump.

10. As system warmup continues and pressure increases to above 4 psi energize
the vacuum pump as necessary to flush the non-condensible gases out of the
system through the vacuum line by forcing the gases out with steam. To
ensure that the non-condensibles gather at the base of the auxiliary
condenser, where the vacuum line suction is located, ensure that the
horizontal tube is not supplied with coolant flow, and adjust coolant flow
through the auxiliary condenser as necessary to ensure steam is filling the
entire system. The auxiliary condenser may be touched lightly by hand, along
its entire length, to ensure the system is completely filled with steam; any cool
spots indicate the presence of non-condensible gases which means that the
flushing process is not complete. The flushing process takes 15 to 30 minutes
to accomplish, and should be repeated periodically for long periods of
operation.

11. At the conclusion of the flushing process, shut the vacuum line valve and
secure the vacuum pump.

12. In order to ensure that filmwise condensation occurs on the tube, coolant flow
through the tube must be initiated as follows:

   a. Allow the apparatus vapor temperature measurement (channel 40) to
      reach at least 3800 microvolts.

   b. Cut in the auxiliary coolant flow (50% or 60% level) to cool the vapor
      temperature to roughly 3200 microvolts.

   c. Secure coolant flow through the auxiliary condenser, and allow the
      vapor temperature level to climb to about 3700-3800 microvolts, which
      allows a steam blanket to cover the tube.

   d. Initiate coolant flow through the single horizontal tube at the 80% level.
e. Cut in coolant flow to the auxiliary condenser to control pressure, and observe the condensation process to ensure that a condensate film has formed on the tube.

13. Run software program DRPINST for an instrumented tube (DRPKS for uninstrumented tubes) by pressing "run" on the keyboard.

To take data for an instrumented tube, the questions for DRPINST can be answered as follows:

- Select fluid ... Enter 0 for steam
- Select option ... Enter 1 to take new data
- Enter month, date and time ... Press enter
- Enter input mode ... Enter 1 for new data
- Give a name for the raw data file ... Enter name
- Enter geometry code ... Enter 1 for finned, 0 for plain
- Select insert type ... Enter 0 for none, 1 for twisted tape, 2 for wire wrap, 3 for Heatex
- No. of thermocouples in wall? ... Enter 4, 5, or 6 depending on the tube
- Select tube diameter type ... Enter 2 for medium
- Enter pressure condition ... Enter 0 for vacuum, 1 for atmospheric
- Give a name for the wall temperature file ... Enter name
- Select input ... Enter 1 for short, 2 for long, or 3 for raw data
- Like to check NG (non-condensible gas) concentration ... Enter 1 for yes, 2 for no; you must answer yes for the first data point
- Enter flowmeter reading (%) ... Enter 2 digit number (i.e. 20 or 58 etc.)
- Connect voltage line ... Flip the voltage line toggle switch, located on the power control panel, to the on position and press enter
• Disconnect voltage line ... Flip the voltage line toggle switch off and press enter

• Enter pressure gauge reading (Pga) ... Enter reading off gauge in psi

• Select measurement ... Enter 0 for teflon, 1 for metal sheath, 2 for quartz, 3 for thermopile

• Change TCOOL rise? ... Enter 1 for yes, 2 for no

• OK to store this data set? ... Enter 1 for yes, 0 for no

• Will there be another run? ... Enter 1 for yes, 0 for no; starts at check NG concentration for following runs.

To take data for an uninstrumented tube the questions for DRPKS can be answered as follows:

• Select fluid ... Enter 0 for steam

• Select option ... Enter 0 to take new data

• Enter month, date and time ... Press enter

• Enter disk number ... Enter number

• Enter input mode ... Enter 0 for new data

• Select Ci ... Enter 0 to find a Ci value, 2 to use a Ci value stored in the program.

• Give a name for the raw data file ... Enter name

• Enter geometry code ... Enter 1 for finned, 0 for plain

• Enter insert type ... Enter 0 for none, 1 for twisted type, 2 for wire wrap, 3 for Heatex

• Select tube type ... Enter 0 for thick wall (only thick wall tubes were tested)

• Select material code ... Enter 0 for copper (only copper tubes were tested)
• Select tube diameter type ... Enter 1 for medium (no small or large diameter tubes were tested)

• Enter pressure condition ... Enter 0 for vacuum, 1 for atmospheric

• Want to create a file for NR vs F? ... Enter 1 for yes, 0 for no

• Give a name for plot data file ... Enter name; easiest to use the raw data file name preceded by a P

• Select output ... Enter 0 for short, 1 for long, 2 for raw data

• Like to check NG concentration ... Enter 1 for yes, 2 for no; you must answer yes for the first data point

• Enter flowmeter reading (%) ... Enter 2 digit number (i.e. 20 or 60 etc.)

• Connect voltage line ... Flip the voltage line toggle switch on and press enter

• Disconnect voltage line ... Flip the voltage line toggle switch off and press enter

• Enter pressure gauge reading (Pga) ... Enter reading off gauge in psi

• Select measurement ... Enter 0 for teflon, 1 for metal sheath, 2 for quartz, 3 for thermopile

• Change TCOOL rise ... Enter 1 for yes, 2 for no

• OK to store this data set ... Enter 1 for yes, 0 for no

• Will there be another run ... Enter 1 for yes, 0 for no; starts at check NG concentration for following runs.

14. Only answer the program questions up to "Enter flowmeter readings". Monitor system temperature using the vapor thermocouple voltage reading (the program automatically resets to channel 40) closely until system warmup is complete.

15. Monitor system temperature and pressure carefully to prevent a system overpressure during warmup (especially at atmospheric conditions).
16. If conducting a vacuum run, gradually adjust voltage to 90 volts (usually in 10 volt increments). Obtain the desired operating condition by manually controlling coolant flow through the auxiliary condenser until channel 40 reads 1970 ± 20 microvolts (≈48°C). Vapor velocity ~ 2 m/s.

17. If conducting an atmospheric run, gradually adjust voltage to 175 volts from the 90 volt level in 10-20 volt increments. Again the desired operating condition is obtained by manually controlling coolant flow through the auxiliary condenser until channel 40 reads 4280 ± 20 microvolts (≈100°C). Vapor velocity ~ 1 m/s.

18. Monitor the condensation process using the glass viewing window periodically to ensure that filmwise condensation is maintained. To clear the viewing window of fog and moisture increase coolant flow through the auxiliary condenser briefly to 50% or 60%, then reset to desired flow rate.

19. When taking readings be sure to check the flowmeter setting prior to entering it into the computer (it has a tendency to fluctuate slightly).

20. If conducting vacuum and atmospheric runs on the same day always conduct the vacuum run first. If the atmospheric run is done first it takes too long for the system to cool down to vacuum operating temperatures.

The system is secured in the following manner:

1. Secure power to the heating elements.

2. Secure coolant flow through the tube, through the auxiliary condenser, and to the sump tank.

3. If desired to maintain the system at vacuum conditions until the next run the shutdown is complete. Continued cooling water circulation may be used to assist in cooling down the system.

4. To bring the system back to atmospheric conditions slowly open the vent valve.

5. The data acquisition system may be turned off whenever it is not necessary to monitor system parameters.

6. Periodically change distilled water in the boiler.
7. If an emergency should arise such as abrupt overpressurization or breakage, immediately secure power to the heaters and open the vent valve, then let the system cool down before checking the apparatus for damage.

B. EXPERIMENTAL PROCEDURES AND OBSERVATIONS

Water is a poor wetting medium and therefore great care was taken to ensure that uniform filmwise condensation was the only condensation mode occurring during a data run. Even though the apparatus was meticulously cleaned (as mentioned previously), a continuing problem with dropwise condensation manifested itself. Subsequent to steam cleaning the system with a Sparkleen soap solution, by operating the system with a soapy solution in the boiler (the solution bubbled through the entire apparatus), dropwise condensation was observed on the installed instrumented tube. After taking some data when in the dropwise condition, the tube was removed and rigorously cleaned using a warm Alconox soap solution with a scrub brush. However, after observing the filmwise mode initially, the condensation mechanism soon transitioned to mixed mode and then back to the dropwise mode.

Since only a filmwise condition over several hours would suffice, the tube chemical treatment procedure used by Guttendorf [Ref. 31] and several other researchers was used to produce filmwise condensation. The tube was chemically treated prior to installation as follows:

1. Clean the internal and especially the external surfaces of the tube using a soft brush and mild soap (using the Alconox detergent in warm water), rinse with acetone then rinse thoroughly with distilled water. Repeat the cleaning
procedure until the distilled water rinse perfectly wets the tube surface; any breaks in the wetting film at this point are likely to result in dropwise condensation spots once the tube is installed in the apparatus.

2. Place the tube in a steam bath.

3. Mix equal amounts of ethyl alcohol and a 50% by weight solution of sodium hydroxide. Keep the solution warm so that a watery consistency is maintained.

4. Apply the solution to the tube with a small paint brush, retaining the tube in the steam bath. If the tube has not been treated previously, apply a coating of the solution every 10 minutes for an hour. If the tube has been previously treated, apply a coating every 5 minutes for a period of 20 minutes.

5. Remove the tube from the steam bath and thoroughly rinse the tube with distilled water to remove any excess solution. Install the tube in the test section immediately, being careful not to touch the tube surface. Oil or dirt from any source may contaminate the tube surface and result in mixed mode or dropwise condensation.

The oxide layer which forms on the tube is very thin, and has negligible thermal resistance and high wetting characteristics.

Once the tube was installed in the apparatus (with the desired insert in place), the system startup procedure outlined in section IV A was followed to take data at desired conditions.

At vacuum conditions, when single tube coolant flow was initiated with vapor velocity at ~ 2 m/s, the condensation on the tube did not develop as a perfect film but instead left patches where the film was broken. These patches, or streaks seemed to occur at regular intervals, and it was postulated that they were due to vortex shedding of vapor around the tube. Therefore, the procedure in the startup section IV A, step 12, was used to promote the development of uniform filmwise
condensation by inducing a stationary vapor condition around the tube. This allowed a steam blanket to form around the tube prior to coolant flow initiation. After flow initiation the appearance of the condensate film on the tube surface was continuous with no breaks. Momentary film instabilities were observed at vacuum conditions at pressures below ~ 20 kPa after a continuous film was established at higher pressures. These instabilities seemed to interrupt the film sheet only for an instant and then disappear, and may have been caused by vortex shedding of the vapor around the tube as already mentioned. A possible mechanism to explain these instabilities is that the higher vapor velocity at vacuum conditions momentarily thins the condensate film via vortex shedding, yet this thinning effect is overcome by surface tension forces in the film sheet which tend to restore the continuous film. These instabilities could only be seen for an instant and then would vanish, being very transitory in nature. There also did not appear to be any pattern whatsoever to the instability formation. The instabilities were not observed at pressures above ~ 20 kPa.

The data taking regimen for each data set involved starting, then verifying the existence of a filmwise condensation condition, then taking data at flow rates (in %) of 80, 70, 60, 50, 40, 30, and 20 then back to 80 and 50 to check for repeatability within the data set. Two data points were taken at each of the first seven data points and one each for the last two, which gave a total of 16 data points. It was usually quite clear from the two comparison points, and from data taken previously under similar conditions, whether the data set should be rejected or accepted. After tube installation, the appearance of one or more small patches (breaks in the film) after
~ 7-10 hours of operation signaled the beginning of tube contamination which got worse with time. The tube would then be removed and cleaned.

C. TUBES TESTED

The data taken during this study involved extensive use of the instrumented smooth tube (S01) fabricated by Poole [Ref. 6], (six wall thermocouples spaced 60 degrees apart placed at midwall and midlength). Due to excessive thermocouple wear, only 5 thermocouples in this instrumented tube functioned properly. The tube was positioned in the apparatus to make optimum use of functioning thermocouples. The preferred arrangement placed 4 thermocouples at 10°, 190°, 250°, and 310° from the top dead center position of the tube. This arrangement provided readings from the top and bottom of the tube and two intermediate points, giving the most accurate mean tube wall temperature and best temperature profile readings. At the conclusion of this study, the manufacture of the new instrumented tubes was not yet complete; progress to date (March 1992) is recorded in Appendix F.

Data was also taken on a uninstrumented smooth tube (S02) and four finned tubes (S03, S04, S05, S06) with fin spacings of 0.5 mm, 1.0 mm, and 1.5 mm, and 2.0 mm. All tubes tested were classified as medium tubes, with an outside diameter of 19.05 mm, and an inside diameter of 12.70 mm. The four finned tubes all had the same fin height of 1.0 mm, and fin width of 1.0 mm. Data runs were taken either with no insert, with the wire wrap or the Heatex insert installed. By more efficiently mixing the coolant an insert significantly increases the inside heat-transfer coefficient.
The Heatex insert consists of a central wire core onto which are wound a series of wire loops, each inclined at a common angle to the core. The loops come into direct contact with the tube wall, and each loop provides a significant amount of coolant mixing as the coolant flows through the loop mesh. [Ref. 12].

The wire wrap insert was a copper wire spirally wrapped around a central stainless steel rod with a uniform pitch. This insert induced a swirling coolant motion, which enhanced turbulent mixing within the tube. This particular wire wrap insert was the same insert used by Guttendorf [Ref. 3], Coumbes [Ref. 22], and Van Petten [Ref. 2] to collect data on the medium family of tubes.

A summary of data runs is given in Table 2.

Table 2. DATA RUN SUMMARY

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<th>Filename</th>
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Tube descriptions:

S01 instrumented smooth tube fabricated by Poole

S02 uninstrumented smooth tube

S03 0.5 mm finned tube

S04 1.0 mm finned tube

S05 1.5 mm finned tube (this tube designated as F096 by Guttendorf; also a similar 1.5 mm finned tube designated as F006 by Guttendorf and Van Petten was not tested); the 1.5 mm finned tube results were not included in subsequent finned tube analysis since the reason for differences in experimental results for these two tubes has not yet been resolved.

S06 2.0 mm finned tube
V. THEORETICAL BACKGROUND AND DATA REDUCTION PROCEDURES

A. THEORETICAL BACKGROUND

The overall or total resistance to heat transfer from vapor to coolant consists of the sum of the vapor side resistance ($R_o$), the tube wall resistance ($R_w$), and the coolant side resistance ($R_i$); this neglects any fouling resistance since clean tubes are always used.

$$R_{total} = R_o + R_w + R_i$$  \hspace{1cm} (5.1)

The vapor and coolant side resistances are convective in nature and may be expressed by the reciprocal of their respective heat-transfer coefficient and surface area product.

$$R_o = \frac{1}{h_o A_o}$$  \hspace{1cm} (5.2)

$$R_i = \frac{1}{h_i A_i}$$  \hspace{1cm} (5.3)

where:

- $R_o$ = outside vapor side resistance to heat transfer (K/W)
- $h_o$ = outside heat-transfer coefficient (W/m$^2$.K)
- $A_o$ = effective outside surface area (m$^2$)
- $R_i$ = inside coolant side resistance to heat transfer (K/W)
- $h_i$ = inside heat-transfer coefficient (W/m$^2$.K)
- $A_i$ = effective inside surface area (m$^2$)

The tube wall resistance is conductive in nature and is represented by the radial conduction equation.
\[ R_w = \frac{\ln \frac{D_r}{D_i}}{2\pi L k_m} \]  

where:

- \( R_w \) = tube wall resistance (K/W)
- \( D_r \) = outside or root (if finned) diameter (m)
- \( D_i \) = inside tube diameter (m)
- \( L \) = active condensing length (133 mm)
- \( k_m \) = thermal conductivity of tube wall (W/m-K)

The effective outside area of the tube is calculated using the following expression:

\[ A_o = \pi D_r L \]  

where:

- \( D_r \) = outside or root (if finned) diameter (m)
- \( L \) = active condensing length (133 mm)

The effective inside area includes the inside surface area involving the active condensing length and the inside surface area of the insulated inlet and outlet portions of the tube. These portions of the tube act as fins and remove heat via axial conduction. The extended fin assumption with associated fin efficiencies was used to account for these end losses.

\[ A_i = \pi D_i (L + \eta_1 L_1 + \eta_2 L_2) \]  

where:

- \( D_i \) = inside diameter of tube (m)
- \( L \) = active tube condensing length (m)
- \( \eta_1 \) = fin efficiency of inlet portion of tube
- \( L_1 \) = length of inlet portion of tube (m)
- \( \eta_2 \) = fin efficiency of outlet portion of tube
- \( L_2 \) = length of outlet portion of tube (m)

The overall thermal resistance to heat transfer may be expressed by:
\[
\frac{1}{U_oA_o} = R_o + R_w + R_i \quad (5.7)
\]

Substituting equations (5.2) and (5.3) into equation (5.7) yields:

\[
\frac{1}{U_oA_o} = \frac{1}{h_oA_o} + R_w + \frac{1}{h_iA_i} \quad (5.8)
\]

where:

- \( U_o \) = overall heat-transfer coefficient (W/m\(^2\)K)
- \( A_o \) = effective outside surface area (m\(^2\))
- \( h_o \) = outside heat-transfer coefficient (W/m\(^2\)K)
- \( R_w \) = Tube wall resistance (K/W)
- \( h_i \) = inside heat-transfer coefficient (W/m\(^2\)K)

The single tube condenser apparatus uses the log mean temperature difference (LMTD) analysis for calculation of the heat transfer between the hot vapor and cold coolant.

\[
Q = U_oA_o(LMTD) \quad (5.9)
\]

where:

- \( Q \) = heat transfer rate to the cooling water (W)
- \( U_o \) = overall heat-transfer coefficient (W/m\(^2\)K)
- \( A_o \) = effective outside surface area (m\(^2\))
- LMTD = log mean temperature difference between vapor and coolant (K)

The log mean temperature difference (LMTD) is given by:

\[
LMTD = \frac{(T_2 - T_1)}{\ln \left[ \frac{T_{sat} - T_1}{T_{sat} - T_2} \right]} \quad (5.10)
\]
where:
\[ T_1 = \text{coolant inlet temperature (K)} \]
\[ T_2 = \text{coolant outlet temperature (K)} \]
\[ T_{sa} = \text{vapor saturation temperature (K)} \]

In this and previous studies at NPS, the quartz thermometer output for \( T_1 \) and \( T_2 \) were used in the calculations for the coolant temperature rise, and the saturation temperature, \( T_{sa} \), was measured using the vapor thermocouple (channel 40).

The total heat transfer across the tube is experimentally determined by measuring the mass flow rate of fluid through the tube and its accompanying temperature rise.

\[ Q = m c_p (T_2 - T_1) \]  
(5.11)

where:
\[ Q \] = heat transfer rate (W)
\[ m \] = mass flow rate of coolant (kg/s)
\[ c_p \] = specific heat of coolant at constant pressure (J/kg·K)
\[ T_1 \] = coolant inlet temperature (K)
\[ T_2 \] = coolant outlet temperature (K)

Equation (5.11) may be used directly from the experimental data. The resultant heat transfer rate, \( Q \), is then substituted into equation (5.9) to find the overall heat-transfer coefficient, \( U_o \).

\[ U_o = \frac{Q}{A_o (LMTD)} \]  
(5.12)

where:
\[ Q \] = heat transfer rate from eq. (5.11) (W)
\[ A_o \] = effective outside surface area (m²)
\[ LMTD \] = log mean temperature difference; eq. (5.10) (K)
Since $R$, $U$, $A$, and $A_i$ are known quantities, this leaves only two unknowns in equation (5.8), $h_o$ and $h_i$, the outside and inside heat transfer coefficients.

Often the coolant side thermal resistance is dominant, and inserts such as those mentioned in section IV C are used to lower the inside resistance. This allows a more accurate computation of the outside heat-transfer coefficient, $h_o$, when using the modified Wilson plot technique mentioned in section V B. Vapor side heat transfer may also be enhanced through the use of fins, drainage strips, or dropwise condensation promoters.

B. MODIFIED WILSON PLOT TECHNIQUE

The ideal way to solve for $h_o$ and $h_i$ in equation (5.8) is through the use of instrumented, which accurately determine a mean tube wall temperature. The inside and outside mean tube wall temperatures may then be obtained directly by assuming a linear temperature profile across the wall. Since the vapor temperature and mean coolant temperature are known, the inside and outside heat-transfer coefficients may then be calculated directly using equation (5.13).

$$q = h \Delta T$$

(5.13)

where:

- $q$ = heat flux (W/m²)
- $h$ = heat-transfer coefficient ($h_i$ for inside, $h_o$ for outside) (W/m²·K)
- $\Delta T$ = temperature difference across resistive medium ($\Delta T = T_{sat} - T_{wall, outside}$ for the vapor side, and $\Delta T = T_{wall, inside} - T_{coolant}$ for the coolant side)

For data collection on a large number of tubes, the use of instrumented tubes is impractical due to the high cost and difficulty involved in manufacturing so many tubes. Therefore, the modified Wilson plot technique was developed, which solves for the inside and outside heat-transfer coefficients simultaneously without using wall
thermocouples. To obtain the most accurate results with this method, it is necessary that the inside and outside coefficients be relatively equal in magnitude.

The modified Wilson plot technique requires that the "form" of the equation for both the inside and outside heat-transfer coefficients be known. The Nusselt theory and Sieder-Tate correlation are used to represent the "form" of the outside and inside heat-transfer coefficients respectively. The Nusselt theory when based on q can be represented by:

\[ h_o = \alpha \left[ \frac{k_f \rho_f g h_{fg}}{\mu_f D_o q} \right]^{1/3} = \alpha Z \]  

or

where:
  - \( h_o \) = outside heat-transfer coefficient (W/m\(^2\)·K), based on q
  - \( \alpha \) = dimensionless coefficient
  - \( k_f \) = thermal conductivity of condensate film (W/m·K)
  - \( \rho_f \) = condensate film density (kg/m\(^3\))
  - \( g \) = gravitational constant (9.81 m/s\(^2\))
  - \( h_{fg} \) = specific enthalpy of vaporization (J/kg)
  - \( \mu_f \) = dynamic viscosity of condensate film (N·s/m\(^2\))
  - \( D_o \) = outside tube diameter (m)
  - \( q \) = heat flux based on outside area (Q/A\(_o\)) (W/m\(^2\))

The Sieder-Tate correlation may be represented by:

\[ h_i = C_i \frac{k_c}{D_i} Re^{0.8} Pr^{1/3} \left( \frac{\mu_c}{\mu_w} \right)^{0.14} = C_i \Omega \]  

where:
  - \( h_i \) = inside heat transfer coefficient (W/m\(^2\)·K)
  - \( C_i \) = Sieder-Tate leading coefficient
  - \( k_c \) = thermal conductivity of cooling water (W/m·K)
  - \( D_i \) = inside tube diameter (m)
  - \( Re \) = Reynolds number
  - \( Pr \) = Prandtl number
  - \( \mu_c \) = dynamic viscosity of cooling water at bulk temperature (N·s/m\(^2\))
\[ \mu_w = \text{dynamic viscosity of cooling water at mean inner wall temperature (N\cdot s/m}^2) \]

Substituting equations (5.14) and (5.15) into equation (5.8) gives:

\[ \left[ \frac{1}{U_o} - R_w A_o \right] Z = \frac{A_o Z}{C_i \Omega A_i} + \frac{1}{\alpha} \quad (5.16) \]

By letting:

\[ Y = \left[ \frac{1}{U_o} - R_w A_o \right] Z \quad (5.17) \]

\[ X = \frac{A_o Z}{A_i \Omega} \quad (5.18) \]

\[ m = \frac{1}{C_i} \quad (5.19) \]

\[ b = \frac{1}{\alpha} \quad (5.20) \]

then equation (5.16) reduces to:

\[ Y = mX + b \quad (5.21) \]

The parameters \( \Omega \) and \( Z \) are both temperature dependent, and must be determined iteratively. A least-squares fit of equation (5.21) is used to determine \( C_i \) and \( \alpha \). Once \( C_i \) is known, \( h_i \) can be calculated using equation (5.15). With \( h_i \) and
Uo known, the value of ho can then be easily determined by rearranging equation (5.7) as in equation (5.22), or by using equation (5.14) with the value of α known.

\[
\frac{1}{h_o} = \frac{1}{U_o} \left[ \frac{A_o}{h_i A_i} + R_w A_o \right]
\]  

(5.22)

C. INSTRUMENTED TUBE IMPROVEMENTS FOR DATA REDUCTION

In previous work at NPS, the standard form of the Sieder-Tate equation was used with a Reynolds number exponent of 0.8, as in equation (2.10). One of the aims of this thesis was to use an instrumented tube to directly determine the inside and outside coefficients, hi and ho, and then use the data to determine a more "exact" form of the Sieder-Tate-type equation to be used for each insert. The coolant side correlations mentioned in section II C were based on a long, straight entrance length. The sharp 90° bend just prior to the test section tube entrance undoubtedly creates entrance effects which lead to discrepancies between our experimental data and heat transfer behavior predicted by the Sieder-Tate correlation.

Assuming the final form of the inside heat-transfer correlation to be that of equation (2.10) gives the following:

\[
Nu = C_i Re^m Pr^{1/3} \left( \frac{\mu_c}{\mu_w} \right)^{0.14}
\]  

(5.23)

where:

\[
m = \text{Reynolds number exponent to be determined}
\]
Rearranging equation (5.23) gives:

\[
\frac{Nu}{Pr^{1/3}\left(\frac{\mu_c}{\mu_w}\right)^{0.14}} = C_i Re^m
\]  

(5.24)

Taking the natural log of equation (5.24) yields:

\[
\ln \left(\frac{Nu}{Pr^{1/3}\left(\frac{\mu_c}{\mu_w}\right)^{0.14}}\right) = \ln C_i + m \ln Re
\]  

(5.25)

Equation (5.25) is in the form of a linear equation, and by plotting \(\ln (Nu/(Pr^{1/3}(\mu_c/\mu_w)^{0.14}))\) versus \(\ln Re\), the slope and intercept, namely the Reynolds number exponent and Sieder-Tate coefficient may be determined from the instrumented data.

With the unknown parameters of equation (5.23) determined, the new inside heat-transfer correlations (one for each insert) could be used in the data reduction program to give the value of the inside heat-transfer coefficient, \(h_i\), directly and to provide a more accurate calculation of the outside heat-transfer coefficient \(h_o\).

D. ENHANCEMENT RATIO

Following the development of Van Petten [Ref.2], and Nusselt theory, experimental data can be curve fitted, using a least-squares analysis, to an equation of the following form:
\[ q = a \Delta T^n \]  
\( (5.26) \)

where:

- \( q \) = heat flux based on outside area (\( Q/A_o \)) (\( \text{W/m}^2 \))
- \( \Delta T \) = temperature drop across the condensate film (K)

Substituting equation (5.12) into his expression yields:

\[ h_o = a \Delta T^{n-1} \]  
\( (5.27) \)

From Nusselt theory \( n = 0.75 \), therefore the enhancement ratio, based on constant temperature drop across the condensate film, can be expressed as:

\[ \epsilon_{\Delta T} = \frac{h_o}{h_{os}} = \frac{a_f}{a_s} \]  
\( (5.28) \)

where:

- \( \epsilon_{\Delta T} \) = enhancement ratio based on constant temperature drop across the condensate film
- \( f \) = subscript denoting finned tube
- \( s \) = subscript denoting smooth tube
- \( h_o \) = outside heat-transfer coefficient (\( \text{W/m}^2\cdot\text{K} \))
- \( a \) = constant of proportionality introduced in equation (5.26)

Also for constant \( \Delta T \); using equation (5.14):

\[ \epsilon_{\Delta T} = \frac{a_f}{a_s} = \frac{\alpha_f}{\alpha_s} \left[ \frac{q_s}{q_f} \right]^{1/3} = \frac{\alpha_f}{\alpha_s} \left[ \frac{a_s \Delta T}{a_f \Delta T} \right]^{1/3} = \frac{\alpha_f}{\alpha_s} \left[ \frac{a_s}{a_f} \right]^{1/3} \]  
\( (5.29) \)

If the heat flux is kept constant the values of \( Z_f \) and \( Z_s \) remain equal, which results in equation (5.30).
\[ \varepsilon_q = \frac{h_{ef}}{h_{os}} \bigg|_q = \frac{\alpha_f Z_f}{\alpha_s Z_s} = \frac{\alpha_f}{\alpha_s} \]  

(5.30)

Combining equations (5.29 and (5.30) gives the relationship between \( \varepsilon_{\Delta T} \) and \( \varepsilon_q \) in equation (5.33).

\[ \varepsilon_{\Delta T} = \frac{\alpha_f}{\alpha_s} \left[ \frac{a_s}{a_f} \right]^{1/3} = \frac{a_f}{a_s} \]  

(5.31)

\[ \varepsilon_q = \frac{\alpha_f}{\alpha_s} = \left[ \frac{a_f}{a_s} \right]^{4/3} \]  

(5.32)

\[ \varepsilon_q = (\varepsilon_{\Delta T})^{4/3} \]  

(5.33)

Note that \( \varepsilon_{\Delta T} \) and \( \varepsilon_q \) are independent of \( q \) and \( \Delta T \).
VI. RESULTS AND DISCUSSION

A. DROPWISE CONDENSATION

As mentioned in section IV B, a dropwise condensation condition was obtained initially. Data were taken during dropwise conditions at vacuum and atmospheric pressure with the instrumented tube, and this data is compared to filmwise data in Figure 4. The figure shows a marked contrast between filmwise and dropwise condensation data. The dropwise heat-transfer enhancement, compared to the filmwise data, varied from ~2 to 7 for vacuum conditions and ~9 to 10 for atmospheric conditions. Marto et al [Ref. 23] studied the use of organic coatings for the promotion of dropwise condensation of steam, and obtained an outside heat-transfer coefficient of ~55 kW/m²·K for a Fluoroacrylic coating compared to ~30 to 85 kW/m²·K found under vacuum conditions in this study (both taken at P~ 11 kPa and vapor velocity ~2 m/s). The difficulty in accurately measuring the temperature drop across the condensate film for the dropwise condition is illustrated by the large amount of scatter in the dropwise data; this problem is caused by the lack of a stable film and the intermittent presence of drops near the instrumented tube wall thermocouples. The exact cause of the dropwise conditions was never determined. Clearly, some organic contamination either from the boiler feed water or from the gasket material was depositing on the test tube. Since this thesis was devoted to filmwise condensation, great efforts were made to clean the test tube and prepare it chemically so that the condensate would wet the surface.
Figure 4. Comparison of Dropwise and Filmwise Condensation Data (Smooth Instrumented Tube, Heatex Insert)
B. INSTRUMENTED TUBE RESULTS

After resorting to the oxide coating procedure mentioned previously to mitigate the dropwise contamination problem, a series of runs were made with the instrumented tube fabricated by Poole [Ref. 6] with the Heatex insert installed. Using a mean wall temperature, the inside and outside heat-transfer coefficients could be evaluated directly.

Figure 5 shows filmwise condensation data taken at various pressures and vapor velocities. The four data sets for increasing vapor velocity (from 1 to 3.5 m/s) and decreasing vapor pressure (from 101 to 28 kPa) show the effect of vapor shear thinning the condensate film, giving an increase in the outside heat-transfer coefficient. This series of data runs was taken by maintaining the heater voltage at 175 volts and adjusting coolant flow through the auxiliary condenser to control pressure.

The four data sets in Figure 5 with constant vapor velocity (−1 m/s) and decreasing vapor pressure (from 101 to 28 kPa) show reduction of the outside heat-transfer coefficient with decreasing saturation temperature (due to decreasing saturation pressure). This effect is thought to be due to an increase in condensate viscosity at lower temperatures which tends to prevent the condensate from flowing around and draining from the tube as easily as at higher temperatures. The resulting condensate film thickening provides an additional resistance to heat transfer thereby lowering the outside heat-transfer coefficient. This series of data runs was taken by both adjusting heater power and auxiliary condenser coolant flow to obtain the same pressure conditions as above with a constant vapor velocity of −1 m/s.
Figure 5. Effect of Pressure and Vapor Velocity on the Steam Heat-Transfer Coefficient (Smooth Instrumented Tube, Heatex Insert)
Figure 6 shows how increased pressure, with vapor velocity held constant, gives a larger increase in the outside heat-transfer coefficient over the predicted Nusselt value. This discrepancy is also thought to be caused by the viscosity effect since the Nusselt treatment seemingly takes into account the other possible causes. Also depicted more clearly is the dramatic enhancement due to vapor shear effects for increased vapor velocity at constant pressure.

As mentioned previously, the use of inserts allows more effective mixing of coolant and facilitates greater accuracy in the calculation of the outside heat-transfer coefficient. Figure 7 shows the outside heat-transfer coefficient determined from the use of the two inserts together with no insert. The results for the wire wrap and Heatex inserts are closely grouped, whereas the no insert data shows much greater scatter. This seems to indicate that the use of an effective insert does indeed enhance accuracy for the instrumented tube data.

The mean temperature difference across the condensate film for the no insert case has a significantly lower value than either of the insert cases; with the outside and tube wall conditions unchanged, this indicates a higher inside resistance for the no insert case. To illustrate this point a mid-range instrumented tube data point at the same conditions for each insert case is shown in Table 3 (P_{sat}~101 kPa, V_{vapor}~1.1 m/s, V_{coolant}~2.75 m/s). It shows that both inserts have a comparable effect (slightly better with Heatex) and roughly provide a factor of two enhancement over the no insert case in the inside heat transfer coefficient. Due to the increased inside resistance for the no insert case, the heat flux shows a decrease of about 15%.
Figure 6. Comparison of Experimental Results with Nusselt Theory for Varying Pressures and Vapor Velocities (Smooth Instrumented tube, Heatex Insert)
Figure 7. Comparison of Steam Heat-Transfer Coefficients at Atmospheric Conditions ($P_{sat} \approx 101$ kPa, $V_{vap} \approx 1.1$ m/s) for Three Insert Conditions (Smooth Instrumented Tube)
Table 3. PERFORMANCE COMPARISON OF TUBE INSERTS AT ATMOSPHERIC CONDITIONS

<table>
<thead>
<tr>
<th></th>
<th>Heatex Insert</th>
<th>Wire Wrap Insert</th>
<th>No Insert</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_i$ (kW/m²·K)</td>
<td>30.85</td>
<td>27.63</td>
<td>15.31</td>
</tr>
<tr>
<td>$q$ (kW/m²)</td>
<td>481.4</td>
<td>477.7</td>
<td>409.6</td>
</tr>
<tr>
<td>$h_o$ (kW/m²·K)</td>
<td>9.86</td>
<td>9.82</td>
<td>10.75</td>
</tr>
<tr>
<td>$\Delta T$(K)</td>
<td>48.82</td>
<td>48.65</td>
<td>38.10</td>
</tr>
</tbody>
</table>

Figure 8 shows the wall temperature profiles for the data points listed in Table 3 along with a vacuum data set ($P_{sat} = 28$ kPa, $V_{vapor} = 1$ m/s, $V_{coolant} = 2.75$ m/s) for comparison, where 0 degrees is at the top dead center position of the tube. The shape of the temperature profiles shows the effect of condensate film thickening toward the bottom of the tube. The higher resistance through a thicker condensate film results in a lower tube wall temperature toward the bottom of the tube as shown.

As expected, the wire wrap insert and Heatex insert temperature profiles at atmospheric pressure are very similar. The no insert profile has a shape similar to the wire wrap and Heatex profiles, but the mean wall temperature is about 11 K higher and shows the effect of the increased inside resistance to heat transfer. The vacuum run temperature profile shows the effect of a lower temperature gradient between the steam and coolant; the lower heat-transfer potential results in lower heat fluxes and a flatter temperature profile.

Figure 9 shows the comparison of the instrumented data with the predictions of Nusselt, Fujii, and Shekriladze and Gomelauri covered in section II B. The data depicted ranges from $P_{sat} = 28$ kPa and $V_{vapor} = 3.5$ m/s to $P_{sat} = 101$ kPa and $V_{vapor} = 1.1$ m/s. The data seems to follow the Shekriladze and Gomelauri prediction the closest, but is also very near the Fujii prediction.
Figure 8. Horizontal Tube Wall Temperature Profiles (Smooth Instrumented Tube, $V_{\text{vapor}} \sim 1 \text{ m/s}$)
Figure 9. Comparison of Experimental Results with the Predictions of Nusselt, Fujii, and Shekriladze-Gomelauri.
C. INSIDE HEAT-TRANSFER CORRELATIONS FROM INSTRUMENTED TUBE RESULTS

The instrumented tube provides a method of determining an inside heat-transfer coefficient for each insert from direct wall temperature measurements. Wanniarachchi et al [Ref. 8], in a similar effort to resolve the differences between the Sieder-Tate correlation and his experimental results, developed a correlation based on the Sieder-Tate expression. Using an intercept form, with data taken on the same apparatus as used in this study, and a least-squares fitted leading coefficient, the correlation took the following form:

\[ Nu = 0.064 Re^{0.8} Pr^{1/3} \left( \frac{\mu_c}{\mu_w} \right)^{0.14} + 26.4 \]  

(6.1)

However, Rouk [Ref. 4], using an optimization technique, showed that the value of the intercept had little effect on the results, and it was the accurate determination of the Reynolds exponent that was more critical.

With a view to finding the appropriate Reynolds exponent and leading coefficient for equation (5.23), the data for no insert, wire wrap, and Heatex insert were plotted as \( \ln \) \( Re \) versus \( \ln (Nu/Pr^{1/3} (\mu_c/\mu_w)^{0.14}) \) as explained in section V C. The plotted data are shown in Figures 10, 11 and 12 for no insert, wire wrap insert, and Heatex insert respectively. A line of best fit (typically with a regression coefficient of 0.99) was used to obtain the value of the intercept. Figure 13 shows the plot of all three cases on the same graph for comparison. Note that the insert data is closely grouped on the upper regions of the graph when compared to the no insert data. The increase in the Nusselt number again indicates more efficient inside heat transfer for the insert vice the no insert case. The Reynolds exponent, or slope,
Figure 10. Log-Log Plot of Re versus $\frac{Nu}{Pr^{1/3}(\mu_d/\mu_w)^{0.14}}$ for No Insert (Smooth Instrumented Tube)
Figure 11. Log-Log Plot of Re versus Nu/Pr^{1/3}(\mu_d/\mu_w)^{0.14} for Wire Wrap Insert (Smooth Instrumented Tube)
Figure 12. Log-Log Plot of Re versus Nu/Pr^{1/3}(\mu_e/\mu_w)^{0.14} for Heatex Insert (Smooth Instrumented Tube)
Figure 13. Combined Log-Log Plot of Re versus Nu/Pr^{1/3}((\mu/\mu_0)^{0.14} for Three Insert Conditions (Smooth Instrumented Tube)
differs from the Sieder-Tate-type equation of 0.8 in all three cases. The following derived correlations apply specifically only to the medium tube, but should be applicable to any tube with the same inside diameter.

The no insert inside heat-transfer correlation had the form:

\[ Nu = 0.013 \ Re^{0.89} Pr^{1/3} \left( \frac{\mu_c}{\mu_w} \right)^{0.14} \]

Equation (6.2) was used to reprocess some current smooth tube runs, with no insert, as well as those of Guttendorf [Ref. 3] and Van Petten [Ref. 2] to check their values for the smooth tube Nusselt coefficient \( \alpha \).

The wire wrap insert inside heat-transfer correlation had the form:

\[ Nu = 0.052 \ Re^{0.82} Pr^{1/3} \left( \frac{\mu_c}{\mu_w} \right)^{0.14} \]

Equation (6.3) was used to reprocess previous data taken on the same apparatus by Guttendorf and Van Petten, who used the wire wrap insert for their finned and smooth tube experiments, with a view to checking their results with this new correlation. Table 4 shows a comparison between the new wire wrap insert correlation (eq. 6.3), and Wanniarachchi's correlation (eq. 6.1) both with and without the intercept value included. This comparison shows that equation 6.1 is actually more accurate without the intercept value included when compared to the results of the new wire wrap insert correlation.
Table 4. COMPARISON OF EQUATIONS (6.1) AND (6.3) FOR \( Pr^{1/3} (\frac{\mu_C}{\mu_w})^{0.14} = 1.4 \) (held constant for comparison)

<table>
<thead>
<tr>
<th>Reynolds Number</th>
<th>20,000</th>
<th>30,000</th>
<th>40,000</th>
<th>50,000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nusselt Number</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( \text{Nu} = 0.064 \ Re^{0.8} Pr^{1/3} (\frac{\mu_C}{\mu_w})^{0.14} + 26.4 )</td>
<td>273.7</td>
<td>368.4</td>
<td>456.9</td>
<td>541.0</td>
</tr>
<tr>
<td>Nusselt Number</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( \text{Nu} = 0.064 \ Re^{0.8} Pr^{1/3} (\frac{\mu_C}{\mu_w})^{0.14} )</td>
<td>247.3</td>
<td>342.0</td>
<td>430.5</td>
<td>514.6</td>
</tr>
<tr>
<td>Nusselt Number</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( \text{Nu} = 0.052 \ Re^{0.82} Pr^{1/3} (\frac{\mu_C}{\mu_w})^{0.14} )</td>
<td>244.9</td>
<td>341.5</td>
<td>432.3</td>
<td>519.1</td>
</tr>
</tbody>
</table>

The Heatex insert inside heat-transfer correlation had the form:

\[
\text{Nu} = 0.22 \ Re^{0.69} Pr^{1/3} (\frac{\mu_C}{\mu_w})^{0.14}
\]  

Equation (6.4) was used to reprocess all Heatex data.

Memory [Ref. 12] conducted condensation experiments on a different apparatus with a smooth instrumented tube. He also determined the Reynolds number exponent for no insert, wire wrap insert (made locally and somewhat different from the one used in this study), and Heatex insert. The exponents he obtained are reported in Table 5. The Heatex insert exponent of 0.68 compares very favorably with the value of 0.69 found in this study. The wire wrap exponent of 0.73 was well below the value of 0.82, most likely due to differences in the insert. The no insert case gave a value of 0.85 and compared well to the value of 0.89 found in this study. The difference in the no insert exponent is most likely due to the difference in the tube entrance region; Memory used a long run of straight pipe for the tube entrance, whereas the present work had a sharp bend just prior to the condenser tube.

Rouk [Ref. 4] used the instrumented smooth tube data of Georgiadis [Ref. 5] to find the appropriate value of the Reynolds number exponent for the no insert
and wire wrap cases. His results compare quite well with this study and are also
given in Table 5.

ANL (Argonne National Laboratory) [Ref. 21] conducted an assessment of
heat-transfer correlations for turbulent pipe flow with water to determine the best
correlation(s) on which to base their design of Ocean Thermal Energy Conversion
(OTEC) heat exchangers. ANL used two shell-and-tube heat exchangers, with no
inserts, for analysis and reported the following:

1. The Dittus-Boelter (eq 2.7) and Sieder-Tate (eq 2.9) correlations under-
predicted the data by 5% to 15% and were considered too conservative for
design.

2. Overall, the "best" correlations were found to be Petukhov-Popov (eq 6.5) and
Sleicher-Rouse (eq 6.6), both of which showed excellent agreement (± 5%)
with the experimental data (at Pr=6.0 and Pr=11.6).

\[
\begin{align*}
\text{Nu} &= \frac{(\epsilon/8) Re Pr}{K_1 + K_2 (\epsilon/8)^{1/2} (Pr^{2/3} - 1)} \quad (6.5) \\
\text{(valid for } 0.5 < Pr < 2000 \text{ and } 10^4 < Re < 5\times10^6) \\
\end{align*}
\]

where:

\[
\begin{align*}
\epsilon &= (1.82 \log_{10} Re - 1.64)^2 \\
K_1 &= 1 + 3.4\epsilon \\
K_2 &= 11.7 + 1.8 Pr^{1/3} \\
\end{align*}
\]

\[
\begin{align*}
\text{Nu} &= 5 + 0.015 Re^a Pr_w^b \\
\text{(valid for } 0.1 < Pr < 10^5 \text{ and } 10^4 < Re < 10^6) \\
\end{align*}
\]

where:

\[
\begin{align*}
a &= 0.88 - 0.24/(4+Pr_w) \\
b &= 1/3 + 0.5e^{0.6Pr_w} \\
\end{align*}
\]
3. The most accurate correlations (i.e. Petukhov-Popov and Sleicher-Rouse) seem to yield effective Reynolds exponents in the neighborhood of 0.85 (uncertainty range: m = 0.82 to 0.88).

4. The potential sources of uncertainty in the Wilson procedure included waterside flow maldistribution, entrance effects, experimental error in $U_p$, and the uncertainty in the Reynolds number exponent. Of these, they concluded that the uncertainty in the Reynolds number exponent was, by far, the most significant. In fact, the results of the Wilson procedure were found to be highly sensitive to the value of the Reynolds number exponent.

Table 5. COMPARISON OF REYNOLDS NUMBER EXPONENTS FOR SIEDER-TATE-TYPE CORRELATIONS

<table>
<thead>
<tr>
<th></th>
<th>Experimental Data</th>
<th>Rouk</th>
<th>Memory</th>
<th>ANL</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Insert</td>
<td>0.89</td>
<td>0.90</td>
<td>0.85</td>
<td>0.85</td>
</tr>
<tr>
<td>Wire Wrap</td>
<td>0.82</td>
<td>0.78</td>
<td>0.73</td>
<td>---</td>
</tr>
<tr>
<td>Heatex</td>
<td>0.69</td>
<td>---</td>
<td>0.68</td>
<td>---</td>
</tr>
</tbody>
</table>

D. ANALYSIS OF SMOOTH TUBE RESULTS

When using the modified Wilson plot technique to reprocess data files, the solution option can be specified to use either the stored value of the Sieder-Tate coefficient (for direct computation of $h_o$) or let the coefficient value "float", which allows the program to calculate its own value of the coefficient. In order to determine which method was most accurate, the instrumented data files were reprocessed using each method and then compared with the values of the heat-transfer coefficient which were obtained by direct measurement of the tube wall temperature. A high, medium, and low coolant flow rate was chosen from each run to facilitate the comparison. The results were tabulated and are shown in Appendix D; it can be seen that the fixed coefficient method yielded the more accurate results at least 75% of the time. The mean error of the fixed method was ±2.0%, and that of the floating method was ±5.4%. The error for the lowest coolant flow rate (Re
<20,000) was noticeably higher than for higher coolant flow rates for both methods. The choice of using the fixed coefficient method represents a departure from the practice of previous researchers on this apparatus who exclusively used the floating coefficient method.

Prior to the instrumented tube runs reported in Figure 5 (from which the new correlations were empirically derived) a series of data runs were made using a plain smooth uninstrumented tube (S02) of the same dimensions using the Heatex insert. The plain smooth tube data was then reprocessed using the new correlation for the Heatex insert and the results are plotted in Figure 14. These data sets were taken at the same conditions as the instrumented data of Figure 5 except for the set at the highest vapor velocity of 6.2 m/s vice 3.5 m/s for the instrumented tube. Similar effects of vapor shear and vapor pressure, as mentioned previously for Figure 5, are clearly seen, and again illustrate the vapor shear effect on the outside heat-transfer coefficient.

With the exception of the two data runs at high vapor velocity, the data from Figures 5 and 14 are shown together in Figure 15. The close agreement of the reprocessed plain smooth tube data with the instrumented smooth tube data allows a high degree of confidence in the accuracy of the new correlations and the choice of the fixed coefficient method.

To provide a baseline from which to evaluate finned tube performance it was necessary to obtain the smooth tube Nusselt coefficient, \( \alpha \), for the specific conditions under which the comparison was to be made. The condition chosen was atmospheric pressure (101 kPa) and a vapor velocity of \( \sim 1 \) m/s.

For 8 complete sets of data the average value of \( \alpha \), using the fixed method, was found to be 0.876. The average value for each data set was found by taking the
Figure 14. Effect of Pressure and Vapor Velocity on the Steam Heat-Transfer Coefficient (Non-instrumented Smooth Tube, Heatex Insert)
Figure 15. Comparison of Instrumented Smooth Tube Results with Non-Instrumented Smooth Tube Data After Reprocessing with the New Heatex Insert Inside Heat-Transfer Correlation.
measured value of \( h_0 \) for each data point, dividing it by the Nusselt theory prediction of \( h_0 \), and then multiplying by 0.728 (the Nusselt coefficient); the average of all the data points in the set was then taken. Interestingly the Wilson plot floating coefficient method gave a value of 0.835, somewhat lower than the average value. Originally it was thought that this discrepancy might be due to "outlier" data points (high or low coolant velocity) in each set. However, removing the highest or lowest coolant flow rates within a set had little or no effect on the Wilson plot result. The reason for the discrepancy is still not known and merits future study.

The value of \( \alpha \) was calculated for several other flow conditions; these are shown in Table 6. The trend of the readings, like that of Figure 6, shows that vapor velocity has a much greater effect than pressure on the value of \( \alpha \), as expected.

<table>
<thead>
<tr>
<th>File Name</th>
<th>P(kPa)</th>
<th>Vapor Velocity (m/s)</th>
<th>( \alpha )</th>
</tr>
</thead>
<tbody>
<tr>
<td>FIMAVSH1</td>
<td>28</td>
<td>3.5</td>
<td>1.015</td>
</tr>
<tr>
<td>FIMAVSH2</td>
<td>41</td>
<td>2.5</td>
<td>.985</td>
</tr>
<tr>
<td>FIMAVSH3</td>
<td>68</td>
<td>1.5</td>
<td>.930</td>
</tr>
<tr>
<td>FIMAVSH4</td>
<td>101</td>
<td>1.1</td>
<td>.866</td>
</tr>
<tr>
<td>FIMAVSH7</td>
<td>69</td>
<td>1.0</td>
<td>.836</td>
</tr>
<tr>
<td>FIMAVSH6</td>
<td>41</td>
<td>1.0</td>
<td>.818</td>
</tr>
<tr>
<td>FIMAVSH5</td>
<td>28</td>
<td>1.0</td>
<td>.786</td>
</tr>
</tbody>
</table>

E. ANALYSIS OF FINNED TUBE RESULTS

With an accurate value of \( \alpha \), and the newly determined inside heat-transfer correlations, the medium family finned tube data of Van Petten [Ref. 2] was evaluated. Figure 16 shows the data Van Petten reported in his thesis; it also shows his data after being reprocessed using the new wire wrap insert correlation with
Figure 16. Comparison of the Steam Heat-Transfer Enhancement Data of Van Petten, for the Medium Finned Tube Family, Using the Modified Wilson Plot and New Wire Wrap Insert Inside Correlation.
both the fixed coefficient and Wilson plot floating coefficient methods. Since Van Petten used the Wilson plot method, it is not surprising that the original thesis data and new Wilson plot floating coefficient data are comparable since the Reynolds exponent only varied from 0.8 to 0.82. The fixed coefficient method enhancement is substantially higher than the Wilson plot results.

Since the assertion is that the fixed coefficient method is more accurate than the Wilson plot method, then the conclusion must be that the enhancement for this set of finned tubes is actually higher than previously reported.

During this study, limited medium finned tube experiments were conducted for purposes of comparison. Figure 17 shows the comparison between this data and the newly reprocessed data of Van Petten (using the fixed coefficient method) and shows reasonable agreement. To more clearly illustrate this point, the data taken on the 2.0 mm fin spacing tube has been given in more detail in Figure 18. Excellent agreement is seen between the experimental results of this study and that of Van Petten using the known inside heat-transfer correlation with the fixed coefficient method. Again, as shown in Figure 16, the Wilson plot prediction is significantly below the fixed coefficient results.
Figure 17. Comparison of the Steam Heat-Transfer Enhancement Data of Van Petten and Swensen for the Medium Finned Tube Family.
Figure 18. Comparison of the Steam Heat Transfer Data of Van Petten and Swensen for the 2.0 mm Fin Spacing Medium Tube.
VII. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

1. The inside heat-transfer correlation is highly sensitive to the Reynolds number exponent.
2. Each insert condition must be analyzed separately to determine the appropriate "form" of the inside heat-transfer correlation.
3. Calculations based on a known inside heat-transfer correlation are more accurate than modified Wilson plot results.
4. Armed with accurate inside heat-transfer correlations, previous data may be reprocessed to give more accurate results.
5. The source of contamination in the test apparatus, which has caused a dropwise condensation problem for a number of years, is most probably due to a contaminated distilled water source.

B. RECOMMENDATIONS

1. Reprocess all previous medium and large diameter finned tube data using the fixed coefficient method to obtain more accurate results.
2. Continue with construction of smooth instrumented tubes of different diameters (i.e. small, medium, and large) to confirm the medium tube results and develop correlations specifically for the small and large diameter tubes.
3. Construct one representative instrumented finned tube to test the validity of applying instrumented smooth tube results to finned tube data.
4. Test representative water samples that have been collected from both the distiller and boiler to confirm the presence of impurities and validate their origin.
5. Replace current distiller with a deionized pure water source (either commercial purchase of water or new distilling apparatus).
APPENDIX A. PHYSICAL AND THERMODYNAMIC PROPERTIES OF WATER

The physical and thermodynamic properties of water were based on the following equations:

Heat Capacity (J/kg·K):

\[ C_p = 4.211 - T \times [2.268 \times 10^{-3} - T \times (4.424 \times 10^{-5} + 2.714 \times 10^{-7} \times T)] \]  \hspace{1cm} (A.1)

where: \( T \) = temperature (celsius)

Dynamic viscosity, (kg/m·s):

\[ \mu = (2.4 \times 10^{-5}) \times 10^{247.8/(T+133.15)} \]  \hspace{1cm} (A.2)

Thermal conductivity (W/m·K):

\[ k = -0.9225 + x \times (2.8395 - x \times (1.8007 - x \times (0.5258 - 0.0734 \times x))) \]  \hspace{1cm} (A.3)

where: \( x = (T + 273.15) / 273.15 \)

Density (kg/m\(^3\)):

\[ \rho = 999.5295 + T \times (0.0127 - T \times (5.4825 \times 10^{-3} - T \times 1.2341 \times 10^{-5})) \]  \hspace{1cm} (A.4)
Latent heat of vaporization (J/kg):

\[ h_{fg} = 2477200 - 2450 \times (T - 10) \]  \hspace{1cm} (A.5)

Fluid Enthalpy (J/kg):

\[ h_f = T \times (4.2038 - T \times (5.8813 \times 10^{-4} - T \times 4.5516 \times 10^{-6})) \]  \hspace{1cm} (A.6)

Saturation pressure (Pa):

\[ P_{sat} = 22120000 \times Pr \]  \hspace{1cm} (A.7)

where:

\[ Pr = e^{Br} \]

\[ Br = \frac{\text{SUM} / \left[ T_r (1 + 4.1671 (1 - T_r) + 20.9751 (1 - T_r)^2) - (1 - T_r) / ((1 \times 10^9) (1 - T_r)^2 + 6) \right]} {\text{SUM}} \]

\[ \text{SUM} = (-7.6912)(1-T_r) - 26.0802 (1-T_r)^2 - 168.1707 (1-T_r)^3 + 64.2329 (1-T_r)^4 - 118.9646 (1-T_r)^5 \]

\[ T_r = (T + 273.15) / 647.3 \]
APPENDIX B. SYSTEM CALIBRATIONS AND CORRECTIONS

B.1 Thermocouple and Quartz Thermometer Calibration

Several different thermocouples and the quartz crystal thermometer (HP 2804A, Ser. No. 2244AD1192) were calibrated against a platinum resistance probe in a mixed isothermal ethylene glycol bath with a Rosemont Galvanometer model 920A commutating bridge (Ser. no. 013494) from 23 to 26 September, 1991.

NOMENCLATURE

\[ T_1 \quad \text{Quartz Thermometer Measuring Probe } T_1 \text{ (Ser. No. 2120A-00707, with dial setting 481)} \]

\[ T_2 \quad \text{Quartz Thermometer Measuring Probe } T_2 \text{ (Ser. No. 2120A-60459 with dial setting 510)} \]

\[ T_1-T_2 \quad \text{The } T_1-T_2 \text{ reading on the quartz thermometer} \]

\[ T-55 \quad \text{Large diameter metal sheath thermocouple (diameter } = 0.040''\text{)} \]

\[ T-56 \quad \text{Small diameter metal sheath thermocouple (diameter } = 0.020''\text{)} \]

\[ T-57 \quad \text{Old Thermocouple (taken from rig during disassembly set to HP 3497A internal zero); (Type: Omega, TT-T-30, Lot# HCP093HC0306)} \]

\[ T-58 \quad \text{New 1 Thermocouple (set to HP 3497A internal zero); (Type: Omega, TT-T-30 SLE, Lot# OCP1453PTCC01473P)} \]

\[ T-59 \quad \text{New 2 Thermocouple (referenced to ice bath zero)} \]

\[ T-60 \quad \text{10 Junction Thermopile #1 (referenced to ice bath zero)} \]

\[ T-61 \quad \text{10 Junction Thermopile #2 (referenced to ice bath zero)} \]

\[ T-62 \quad \text{10 Junction Thermopile #3 (referenced to ice bath zero)} \]

The calibration was performed by taking the bath temperature up from 290 K to 393 K by 5 K increments then back down to check for hysteresis; no hysteresis was
observed. The data was fitted to a fifth-order polynomial (regression coefficient = 1.000 for each polynomial fit) in each case with the following results:

\[ T56 = 273.15 + (2.5878e-2)V - (5.9853e-7)V^2 - (3.1242e-11)V^3 + (1.3275e-14)V^4 - (1.0188e-18)V^5 \]  
\[ T57 = 273.15 + (2.5923e-2)V - (7.3933e-7)V^2 + (2.8625e-11)V^3 + (1.9717e-15)V^4 - (2.2486e-19)V^5 \]  
\[ T59 = 273.15 + (2.5471e-2)V - (3.7621e-7)V^2 - (1.0105e-10)V^3 + (2.3928e-14)V^4 - (1.6440e-10)V^5 \]  
\[ T60 = 273.17 + (2.5571e-2)V - (1.9980e-7)V^2 - (1.6385e-10)V^3 + (2.6164e-14)V^4 - (1.0295e-18)V^5 \]  
\[ T61 = 273.15 + (2.6119e-2)V - (9.0449e-7)V^2 + (1.1214e-10)V^3 - (1.5623e-14)V^4 + (1.0646e-18)V^5 \]
\[ T62 = 273.15 + (2.5996 \times 10^{-2})V - (7.4405 \times 10^{-7})V^2 \]
\[ + (2.4733 \times 10^{-1})V^3 + (3.3236 \times 10^{-15})V^5 \]
\[ - (3.7460 \times 10^{-19})V^5 \]

For \( T55, T56, T57, T59; \) \( V=\) Voltage in microvolts (eg. 0.010 volts = 10,000 microvolts).

For \( T60, T61, T62; \) \( V=\) Voltage in microvolts/10 (eg. 100,000 microvolts ÷ 10 = 10,000).

The HP 2804A Quartz Crystal Thermometer was also calibrated on 26 September, 1991 with the results summarized in Table B.1.

<table>
<thead>
<tr>
<th>Reference Temperature (deg C)</th>
<th>( T1 ) (deg C)</th>
<th>Error (deg C)</th>
<th>( T2 ) (deg C)</th>
<th>Error (deg C)</th>
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<td>0.013</td>
<td>30.164</td>
<td>0.024</td>
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<td>0.006</td>
<td>34.998</td>
<td>0.021</td>
</tr>
</tbody>
</table>

The mean error for the Quartz Thermometer as shown in the Table B.1 is \(-+0.02\) deg C. However, it is well noted that \( T1-T2, \) the critical measurement, indicates an apparent error of less than \(+0.005\) deg C when the temperature reading is near or below 25 deg C; \( T1 \) and \( T2 \) tracking well together lowers the error estimate.
B.2 Flow Meter Calibration

The flow meter calibration for coolant flow through the single horizontal tube was completed on Oct 28, 1991 using a stopwatch, portable tank, and a Toledo model 31-0851 IV, Se. No. 1326 scale with 1/10 pound graduations. The following relation was obtained via linear regression:

\[ m = 6.7409F + 13.027 \]  

where:

- \( m \) = mass flow rate (grams per second)
- \( F \) = flow meter reading (eg. 10% => 10)

The applicable range of the calibration was 10% to 95%. The water temperature on the day of the calibration was 17.5°C (290.6 K) and water density was 998.5 kg/m³. The data is shown in Figure B.1

B.3 Mass Flow Rate Correction

The inlet water temperature from the cooling water sump varies anywhere from 15°C to 25°C depending on environmental conditions. To account for these temperature variations the following function was used to calculate a correction factor for viscosity variation with temperature [Ref. 5].

\[ Cf = 1.0365 - (1.9644E - 3) Tin + (5.2500e - 6) Tin^2 \]  

where:

- \( Cf \) = mass flow rate correction factor
- \( Tin \) = inlet temperature (celsius)
Figure B.1  Horizontal Tube Coolant Flowmeter Calibration Chart

\[ y = 6.7409x + 13.027 \]
The value of \( Cf \) for the flow meter calibration \( (T_{in}=17.5^\circ C) \) was 1.0037. Therefore, the actual mass flow rate was calculated using the following equation:

\[
m_{act} = m_{calc} \frac{Cf}{1.0037} \tag{B.11}
\]

where:

- \( m_{act} = \) corrected mass flow rate (grams per second)
- \( m_{calc} = \) computed mass flow rate (eq. B.9) (grams per second)

### B.4 Pressure Transducer Calibration

Three methods of pressure measurement were available on the apparatus:

1. Direct pressure reading off the Heise solid front - CM-104119 pressure gauge, (range 0-15 psia).
2. Converted voltage readings from the Setra, model 204, Ser. no. 63982 pressure transducer (range 0-14.7 psia; 0-5 volts; 5V—0 psia).
3. Steam saturation temperature measurement with the apparatus producing steam at steady state. The steam saturation temperature/pressure relation was utilized via standard steam tables.

The pressure transducer was calibrated versus the vapor temperature probe reading on 12 December 1991. Equation B.12 gives the desired relationship. The data is shown in Figure B.2.

\[
P = -2.9360V + 14.7827 \tag{B.12}
\]

where:

- \( P = \) pressure (psia)
- \( V = \) pressure transducer voltage reading (volts)

### B.5 Friction Temperature Correction

As coolant flows through the tube there is a bulk temperature rise due to frictional heating, which is highly dependent on fluid velocity. Although small, this
Figure B.2 Pressure Transducer Calibration Chart
can have a significant effect on the calculated overall heat-transfer coefficient. Measurements were made for no insert, Heatex, wire wrap and twisted tape inserts as shown in Figure B.3, on 5 December 1991. Each data set was curve fitted to a third order polynomial which is depicted in Table B.2. Each respective polynomial corrects the temperature rise measurement for the heating due to the particular type of insert used.

Table B.2  FRICITION TEMPERATURE RISE POLYNOMIALS

<table>
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<tr>
<th>Insert Type</th>
<th>Polynomial</th>
</tr>
</thead>
<tbody>
<tr>
<td>None</td>
<td>Trise = -1.960x10^{-5}V^3+9.349x10^{-4}V^2+1.749x10^{-4}V-2.728x10^{-4}</td>
</tr>
<tr>
<td>Wire Wrap</td>
<td>Trise = 8.160x10^{-5}V^3+1.451x10^{-3}V^2+2.745x10^{-3}V-3.991x10^{-4}</td>
</tr>
<tr>
<td>Heatex</td>
<td>Trise = 8.160x10^{-5}V^3+1.080x10^{-3}V^2+1.232x10^{-3}V+8.570x10^{-5}</td>
</tr>
<tr>
<td>Twisted Tape</td>
<td>Trise = 4.070x10^{-5}V^3+4.451x10^{-4}V^2+1.711x10^{-3}V-6.440x10^{-5}</td>
</tr>
</tbody>
</table>

where: Trise = temperature rise (K)

V = fluid velocity (m/s)
Figure B.3 Friction Temperature Rise Curves for Heatex Insert, Wire Wrap Insert, Twisted Tape Insert, and No Insert.
APPENDIX C. SYSTEM INTEGRITY / LEAK TESTING

As mentioned in section III C, the material for the auxiliary condenser penetration plates was changed from aluminum to stainless steel on 24 January 1992. The stainless steel screw thread connectors in the aluminum side plate had loosened to the extent that leakage could be detected. The cause of this loosening was due to thermal cycling of the apparatus and the differential contraction/expansion of the aluminum/stainless steel combination.

An initial leak test was conducted from 20 December 1991 through 2 January 1992; the results are shown in Figure C.1. The initial mean leak rate was 3.4 mmHg per day.

Subsequent to the structural modification noted above, another leak test was conducted 6-19 February 1992; the results are shown in Figure C.2. The mean leak rate was found to be 1.7 mmHg per day, an noticeable improvement. In general, a leak rate of 2 mmHg per day is considered acceptable.
Figure C.1  Apparatus Leak Test I

\[ y = 3.404x + 16.425 \]
Figure C.2 Apparatus Leak Test II

\[ y = 1.657x + 21.332 \]
APPENDIX D. COMPARISON OF FIXED C_i vs FLOATING C_i SOLUTION METHODS FOR MODIFIED WILSON PLOT DATA REPROCESSING.

As related in section VI D the "fixed" C_i method and "floating" C_i method were evaluated against the original instrumented tube results in order to choose the most accurate method of reprocessing non-instrumented data. The results of the fixed coefficient versus modified Wilson plot floating coefficient method comparison are shown in Table D. The fixed coefficient method was determined to be more accurate than the floating coefficient method 75% of the time. For the three coolant flow rates considered for each run, the overall fixed coefficient method mean error was ±2.0%, and the floating coefficient method mean error was ±5.4%.

For Re <20,000 the error was noticeably higher, which tends to support the assertion of a number of researchers that data with Reynolds numbers below 20,000 should not be used, since the flow may not be fully turbulent.
Table D. COMPARISON OF FIXED $C_i$ vs FLOATING $C_i$ REPROCESSING METHODS

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<th>Inst Tube Data</th>
<th>Fixed Method</th>
<th>% error</th>
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<th>% error</th>
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APPENDIX E. UNCERTAINTY ANALYSIS

In the measurement of a physical quantity, there will always be a difference between the measured value of the quantity and the actual value. The magnitude of this difference depends on the accuracy of the measuring device calibration, operator experience, environmental effects, etc. Even though the error associated with a single measurement may be rather small, the error may grow to substantial proportions when combined with other measured quantities in a given calculation scheme. The best estimate of the difference between a calculated or measured quantity and the actual value of the quantity is known as the uncertainty.

The uncertainty may be estimated by the method of Kline and McClintock [Ref. 24]. This states that for a quantity $R$, which is a function of several measured quantities ($R = R(x_1, x_2, x_3, ..., x_n)$), the uncertainty in $R$ is given by the following relation:

$$W_R = \left[ \left( \frac{\partial R}{\partial x_1} W_1 \right)^2 + \left( \frac{\partial R}{\partial x_2} W_2 \right)^2 + \cdots + \left( \frac{\partial R}{\partial x_n} W_n \right)^2 \right]^{1/2} \quad (E.1)$$

where:

$W_R$ = the uncertainty of the desired dependent variable

$x_1, x_2, x_3, ..., x_n$ = the measured independent variables

$W_1, W_2, W_3, ..., W_n$ = the uncertainties in the measured variables.

Georgiadis [Ref. 5] gives a complete description of the uncertainty analysis for this experiment. The uncertainty analysis program written by Mitrou [Ref. 25] was used to calculate uncertainties. The uncertainty for runs using an insert ranged from
±2.2% to ±4.6%. The uncertainty for runs without an insert ranged from ±10.9% to ±17.1%. Sample outputs of the uncertainty evaluations are included in this appendix.
DATA FOR THE UNCERTAINTY ANALYSIS:

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DATA FOR THE UNCERTAINTY ANALYSIS:

File Name: FNMAVSH1
Pressure Condition: Vacuum
Vapor Temperature = 53.611 (Deg C)
Water Flow Rate (%) = 30.00
Water Velocity = 1.70 (m/s)
Heat Flux = 3.233E+05 (W/m²)
Tube-metal thermal conduc. = 395.0 (W/m.K)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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File Name: FNMAVSH4
Pressure Condition: Atmospheric (101 kPa)
Vapor Temperature = 100.086 (Deg C)
Water Flow Rate (%) = 90.00
Water Velocity = 4.34 (m/s)
Heat Flux = 7.258E+05 (W/m²)
Tube-metal thermal conduc. = 395.0 (W/m.K)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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<td>Heat Flux, $q$</td>
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<td>Overall H.T.C., $U_o$</td>
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<td>Sieder-Tate constant</td>
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UNCERTAINTY ANALYSIS:

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UNCERTAINTY ANALYSIS:

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DATA FOR THE UNCERTAINTY ANALYSIS:

File Name: FNMAVSH6
Pressure Condition: Vacuum
Vapor Temperature = 76.541 (Deg C)
Water Flow Rate (%) = 20.00
Water Velocity = 1.16 (m/s)
Heat Flux = 3.965E+05 (W/m²)
Tube-metal thermal conduc. = 305.0 (W/m.K)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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DATA FOR THE UNCERTAINTY ANALYSIS:

File Name: FNMAFOSI
Pressure Condition: Atmospheric (101 kPa)
Vapor Temperature = 100.138 (Deg C)
Water Flow Rate (%) = 80.00
Water Velocity = 4.32 (m/s)
Heat Flux = 1.291E+06 (W/m²)
Tube-metal thermal conduc. = 305.0 (W/m.K)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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File Name: FNMAFO51
Pressure Condition: Atmospheric (101 kPa)
Vapor Temperature
Water Flow Rate (%) = 100.116 (Deg C)
Water Velocity = 50.00
Heat Flux = 2.73 (m/s)
Tube-metal thermal condu. = 1.139E+06 (W/m^2)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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<td>Heat Flux, q</td>
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<td>Log-Mean-Tem Diff, LMTD</td>
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<td>Wall Resistance, Rw</td>
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<td>Overall H.T.C., Uo</td>
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DATA FOR THE UNCERTAINTY ANALYSIS:

File Name: FNMAFO51
Pressure Condition: Atmospheric (101 kPa)
Vapor Temperature
Water Flow Rate (%) = 100.271 (Deg C)
Water Velocity = 20.00
Heat Flux = 1.16 (m/s)
Tube-metal thermal condu. = 0.513E+05 (W/m^2)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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DATA FOR THE UNCERTAINTY ANALYSIS:

File Name: FNMAF201
Pressure Condition: Atmospheric (101 kPa)
Vapor Temperature = 99.787 (Deg C)
Water Flow Rate (%) = 80.00
Water Velocity = 4.33 (m/s)
Heat Flux = 1.431E+06 (W/m^2)
Tube-metal thermal conduc. = 385.0 (W/m.K)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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DATA FOR THE UNCERTAINTY ANALYSIS:

File Name: FNMAF201
Pressure Condition: Atmospheric (101 kPa)
Vapor Temperature = 100.075 (Deg C)
Water Flow Rate (%) = 50.00
Water Velocity = 2.74 (m/s)
Heat Flux = 8.941E+05 (W/m^2)
Tube-metal thermal conduc. = 385.0 (W/m.K)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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DATA FOR THE UNCERTAINTY ANALYSIS:

File Name: FNMAF201
Pressure Condition: Atmospheric (101 kPa)
Vapor Temperature = 100.142 (Deg C)
Water Flow Rate (%) = 20.00
Water Velocity = 1.15 (m/s)
Heat Flux = 6.405E+05 (W/m²)
Tube-metal thermal conduc. = 395.0 (W/m.K)
Sieder-Tate constant = 0.2200

UNCERTAINTY ANALYSIS:

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<td>Vapor-Side H.T.C., Ho</td>
<td>4.59</td>
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DATA FOR THE UNCERTAINTY ANALYSIS:

File Name: FSONMASN1
Pressure Condition: Atmospheric (101 kPa)
Vapor Temperature = 100.1SE (Deg C)
Water Flow Rate (%) = 90.00
Water Velocity = 4.35 (m/s)
Heat Flux = 4.540E+05 (W/m²)
Tube-metal thermal conduc. = 395.0 (W/m.K)
Sieder-Tate constant = 0.0130

UNCERTAINTY ANALYSIS:

<table>
<thead>
<tr>
<th>VARIABLE</th>
<th>PERCENT UNCERTAINTY</th>
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<tr>
<td>Mass Flow Rate, Md</td>
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<tr>
<td>Reynolds Number, Re</td>
<td>1.11</td>
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<tr>
<td>Heat Flux, q</td>
<td>1.02</td>
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<tr>
<td>Log-Mean-Tem Diff, LMTD</td>
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<tr>
<td>Wall Resistance, Rw</td>
<td>2.67</td>
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<tr>
<td>Overall H.T.C., Uo</td>
<td>1.11</td>
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<tr>
<td>Water-Side H.T.C., H1</td>
<td>15.41</td>
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<td>10.81</td>
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DATA FOR THE UNCERTAINTY ANALYSIS:

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<th>Percent Uncertainty</th>
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<td>Heat Flux, q</td>
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<td>Log-Mean-Tem Diff, LMTD</td>
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<td>Water-Side H.T.C., Hi</td>
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<td>Vapor-Side H.T.C., Ho</td>
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UNCERTAINTY ANALYSIS:

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<td>Wall Resistance, Rw</td>
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<td>Overall H.T.C., Uo</td>
<td>1.17</td>
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<tr>
<td>Water-Side H.T.C., Hi</td>
<td>15.42</td>
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<tr>
<td>Vapor-Side H.T.C., Ho</td>
<td>12.42</td>
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APPENDIX F. INSTRUMENTED TUBE CONSTRUCTION

The medium diameter instrumented tube of Poole [Ref. 6], with six wall thermocouples, was fabricated by welding together a copper tube in three pieces after imbedding capillary tubes at mid depth in the tube wall. Teflon thermocouples were inserted into the capillary tubes for tube wall temperature measurement.

An attempt was made during this study to construct instrumented tubes of small, medium, and large diameter in the following steps:

1. Take thick-walled copper tube bar-stock and machine to specified inside diameter, outside diameter, and length.
2. Cut four evenly spaced slots, of given depth and width to accommodate metal sheathed thermocouples, from halfway along the tube to the end.
3. Solder metal sheathed thermocouples into groves.
4. Cut copper strips from another tube to fit into the top of the slot.
5. Clamp copper strips to the slots (using jubilee clips) and solder in place.
6. Turn off excess copper from strips to original outside diameter.
7. Send tubes to plating shop and plate the whole tube to a given thickness of plate.
8. Return tube to NPS machine shop and machine back down to desired outside diameter.

The process was completed through step 2 of the above procedure and the tubes are ready for step 3. Several of the latter steps were attempted using a practice tube with monel wire placed in the slots on the tube vice thermocouples; this was to ensure the process was safe, and to prevent destruction of assets (small, medium, and large diameter tubes machined to specifications, and the associated
metal sheathed thermocouples). The following summary documents the steps taken and lessons learned.

1. Completed tube fabrication through step 2 of the fabrication procedure (above). A practice tube was used from this point to continue the procedure.

2. Cutting the copper strips proved difficult, so the monel (similar to the stainless steel sheath on the thermocouples) wires were silver soldered after being pinned in place. No lead was permitted in the solder since copper will not plate to lead, but will plate to silver.

3. The practice tube was sent to a local contractor for electroplating. (Bay Custom Chrome, Marina, Ca.)

4. Two different plating procedures were used on the tube:
   a. First, a cyanide bath treatment was used to electroplate the tube. However, this procedure severely scorched the surface of the tube, resulting in the return of the tube to the NPS machine shop for repair.
   b. After repair, the tube was again sent off and subsequently treated with an acid bath procedure, which resulted in an acceptable tube surface finish.

5. The original thickness of electroplate was not thick enough for our application, so another acid bath treatment was performed to bring the surface thickness to the desired level.

6. The final product was suitable for our intended application with the exception of the following items which need to be corrected:
   a. Existing voids (from non-uniform soldering) or flaws in the tube surface would not fill in to give a uniform thickness around the tube.
   b. The practice wires, tube end pieces, and tube interior were not protected properly during the electroplating process, resulting in unacceptable copper deposition on critical areas.

It is recommended that the instrumented tube efforts be continued with particular attention to the following items:

1. Silver soldering is an acceptable method of fixing the thermocouples in place, since the thermocouple melting temperature is ~1700°C (silver solder is applied at ~1100°C).
2. If the voids left in previous soldering efforts persist, then cut copper strips to place over the thermocouples to hold them in place.

3. Ensure that critical areas on the tube are well protected during the electroplating process.
APPENDIX G. DRPINST PROGRAM LISTING

The program DRPINST, which was used to collect all instrumented data, is listed in this appendix.
1000: DRPINST
1001: WRITTEN FOR INSTRUMENTED TUBES
1002: BY S. MEMORY (10TH DECEMBER 1991)
1003: 
1004: USED TO COLLECT DATA ON THE INSTRUMENTED
1005: TUBE FABRICATED BY POOLE (1983) FROM WHICH
1006: NEW INSIDE HEAT-TRANSFER CORRELATIONS WERE
1007: EMPIRICALLY DETERMINED FOR NO INSERT, WIRE
1008: WRAP INSERT, AND HEATEX INSERT CASES BY
1010: 
1011: ALL INSTRUMENT CALIBRATIONS FOR FLOW METER
1012: THERMOCOUPLES, THERMOPILE, PRESSURE TRANSDUCER,
1013: FRICTION TEMP RISE FOR NO INSERT, WIRE WRAP INSERT,
1014: HEATEX INSERT, AND TWISTED TAPE INSERT WERE
1016: VARIOUS PROGRAMS WERE USED TO REPROCESS DATA
1017: COLLECTED; THE APPLICABLE FILES WITH THE APPROPRIATE
1018: REPROCESSING PROGRAMS ARE AS FOLLOWS:
1019: 
1020: FILES REPROCESSING PROGRAM
1021: SWENSEN FIMAUSH1-7 DRPINST2
1022: SWENSEN FIMAUS4,5 DRPINST1
1023: SWENSEN FIMAUS3-5 DRPINST1
1024: SWENSEN NON-INSTRUMENTED DATA DRPHS
1025: "ALL FN..... FILES"
1026: BUTTENDORF DATA DRPHS
1027: VAN PELTEN DATA DRPHS
1028: 
1029: FILE NAME DEFINITIONS FOR SWENSEN:
1030: 
1031: LETTER/NR NUMBER DEFINITION
1032: POSITION
1033: 1ST CONDENSATION CONDITION
1034: D=DROPWISE, F-FILMWISE
1035: 2ND TUBE TYPE I-INSTRUMENTED
1036: N=NON-INSTRUMENTED
1037: 3RD TUBE TYPE M=MEDIUM
1038: 4TH PRESSURE CONDITION
1039: A=ATMOSPHERIC (101 KPA)
1040: V-VACUUM
1041: WHEN A & V APPEAR TOGETHER AS IN
1042: FIMAUSH1 IT REFERS TO A SERIES
1043: OF RUNS AT VARIOUS Pressures AND
1044: VAPOR VELOCITIES.
1045: S=SMOOTH
1046: 6TH TUBE TYPE S-SMOOTH

108
F-FINNED

6TH TYPE OF INSERT INSTALLED

N-NONE

W-WIRE WRAP

H-HEATEX

THE HEATEX INSERT WAS USED EXCLUSIVELY FOR FINNED TUBE DATA, SO POSITIONS 6 & 7 REFER TO THE TUBE FIN SPACING

05=0.5mm

10=1.0mm

15=1.5mm

20=2.0mm

FINAL POSITION RUN NUMBER

MEANING OF ALL FLAGS IN PROGRAM

IFT: FLUID TYPE

ISO: OPTION WITHIN PROGRAM

IM: INPUT MODE

IFG: FINNED OR SMOOTH

INN: INSERT TYPE

IWT: LOOP NO. WITHIN PROGRAM

ITDS: TUBE DIAMETER

IPC: PRESSURE CONDITION

IOU: OUTPUT REQUIRED

COM /Cc/ C(7)

COM /CcS5/ T55(S)

COM /CcE/ T56(S)

COM /CcS7/ T57(S)

COM /CcS9/ T58(S)

DIM Enf(15),Twl(6),Twe(6),Twi(6)

DATA 0.10995091,25727.94369,-767345.9295,78025595.81

DATA -9247486599,6.97598E+11,-2.65192E+13,3.94078E+14

READ C(*)


READ T55(*)

DATA 273.15,2.5978E-2,-5.9863E-7,-3.1242E-11,1.3275E-14,-1.0188E-18

READ T56(*)

DATA 273.15,2.5923E-2,-7.3933E-7,2.8625E-11,1.9717E-15,-2.2486E-19

READ T57(*)


READ T58(*)

Diss=.1524 ' Inside diameter of stainless steel test section

Ax=PI*Diss^2/4

L=-.3335 ' Condensing length

L1=.060325 ' Inlet end "fin length"

L2=.034925 ' Outlet end "fin length"

PRINTER 15 1

BEEP
1107 INPUT "SELECT FLUID (0=STEAM, 1=R-113, 2=EG)",If
1108 PRINT USING "4X," "Select option:"
1109 PRINT USING "6X," "1 Take data or re-process previous data"
1110 PRINT USING "6X," "2 Print raw data (NOT COMPLETE)"
1111 PRINT USING "6X," "3 Purge"
1112 INPUT Isa
1113 IF Isa>1 THEN 3082
1114 PRINTER IS 701
1115 Ijob=0
1116 CLEAR 709
1117 BEEP
1118 INPUT "ENTER MONTH, DATE AND TIME (MM:DD:HH:MM:SS)",Date$
1119 OUTPUT 709:"TD";Date$
1120 OUTPUT 709:"TD"
1121 IF Ijob=1 THEN
1122 BEEP
1123 PRINT USING "SKP PAGE AND HIT ENTER",ok
1124 END IF
1125 ENTER 709:Date$
1126 PRINT "Month, date and time:";Date$
1127 PRINT
1128 PRINT USING "6X," "NOTE: Program name : DRPIST"
1129 IF Ijob=1 THEN 1189
1130 BEEP
1131 INPUT "ENTER INPUT MODE (1=NEW DATA, 2=EXISTING FILE)",Im
1132 BEEP
1133 IF Im=1 THEN
1134 BEEP
1135 INPUT "GIVE A NAME FOR THE NEW DATA FILE",D_file$
1136 PRINT USING "6X," "File name : ",D_file$
1137 CREATE EDAT D_file$15
1138 ASSIGN 0=file TO D_file$
1139 BEEP
1140 INPUT "ENTER GEOMETRY CODE : (1-FINNED, 0-PLAIN)",Ifg
1141 PRINTER IS 1
1142 Inn=0
1143 BEEP
1144 PRINT "SELECT INSERT TYPE:
1145 PRINT " 0NONE (DEFAULT)"
1146 PRINT " 1TWISTED TAPE"
1147 PRINT " 2WIRE WRAP"
1148 PRINT " 3HEATEX"
1149 INPUT Inn
1150 OUTPUT 0=file:Ifg,Inn
1151 INPUT "NO. OF THERMOCOUPLES IN WALL",Inw
1152 IF Ifg=1 THEN
1153 INPUT "ENTER FIN HEIGHT, FIN PITCH AND FIN WIDTH (Fh,Fp,Fw) IN MM",Fh,Fp,Fw
1154 ELSE
1155 Fh=0

110
Fp=0
Fw=0
END IF
OUTPUT @File;Inwt,Fp,Fw,Fh
ELSE
IF Ijob=1 THEN 1255
BEEP
INPUT "GIVE THE NAME OF THE EXISTING DATA FILE",D_file$;
PRINT USING "1EX,""This analysis was performed for data in file ",D_file$;
IF Ijob=1 THEN 1270
Nrun=18
BEEP
INPUT "ENTER NUMBER OF DATA SETS (DEF=18)",Nrun
ASSIGN @File TO D_file$
ENTER @File;Ifg,Inn
ENTER @File;Inwt,Fp,Fw,Fh
END IF
IF Ijob=1 THEN 1543
PRINTER IS 1
BEEP
PRINT "SELECT TUBE DIA TYPE:"
Itds=2
PRINT "1 SMALL"
PRINT "2 MEDIUM (DEFAULT)"
PRINT "3 LARGE"
INPUT Itds
PRINTER IS 701
D1=.0127 ID of medium and large tubes
Do=.01905 OD of medium tube
D1=.01905 OD of unheated inlet length
D2=.015875 OD of unheated outlet length
Dr=.015875 Thermocouple burial depth
IF Itds=1 THEN
D1=.009525
Do=.0127
Dr=.01 TO BE MODIFIED WHEN KNOWN
END IF
IF Itds=3 THEN
Do=.025
Dr=.02 TO BE MODIFIED WHEN KNOWN
END IF
Kcu=385
Rm=Do*LOG(Do/D1)/(2*Kcu) Wall resistance based on outside area
BEEP
INPUT "ENTER PRESSURE CONDITION (0-V,1-A)",Ipc
BEEP
PRINT USING "1EX,""This analysis includes end-fin effect"
PRINT USING "1EX,""Thermal conductivity = ",Kcu
PRINT USING "1EX,""Inside diameter, Di = ",D1*1000
1552 PRINT USING "16X,""Outside diameter, Do = "",DD,DD,"" (mm)"";Do=1000
1560 IF Ijob=1 THEN 1548
1569 BEEP
1576 INPUT "GIVE A NAME FOR WALL TEMPERATURE FILE",Wtf$
1591 CREATE BOAT Wtf$.
1604 ASSIGN @File1 TO Wtf$
1612 IF Ijob=1 THEN
1618 Iov=2
1624 GOTO 1702
1631 END IF
1636 BEEP
1643 INPUT "SELECT OUTPUT (1=SHORT,2=LONG,3-RAW DATA)",Iov
1650 IF Iov=0 THEN
1657 PRINT USING "16X,""Tube type : INSTRUMENTED SMOOTH"
1660 ELSE
1667 PRINT USING "16X,""Tube type : INSTRUMENTED FINNED"
1674 PRINT USING "16X,""Fin pitch, width, and height (mm): "",DD,DD,2x,Z,DD,2x,
7.2D"";fp,Fw,Fh
1681 END IF
1688 J=0
1695 Go_on=1
1702 Repeat:
1709 J=1+1
1716 IF Im=1 THEN
1723 BEEP
1730 OUTPUT 709:""AR AF40 AL40 VPS"
1737 OUTPUT 709:""AS SA"
1744 INPUT "LI+E TO CHEMICAL CONCENTRATION (1=Y,0=N)?:",Ng
1751 IF Ie=1 THEN
1758 OUTPUT 709:""AR AF40 AL40 VPS"
1765 OUTPUT 709:""AS SA"
1772 END IF
1779 BEEP
1786 INPUT "ENTER FLOWMETER READING",Fm
1793 OUTPUT 709:""AR AF40 AL40 VPS"
1800 OUTPUT 709:""AS SA"
1807 ENTER 709;Et
1814 OUTPUT 709:""AS SA"
1821 BEEP
1828 INPUT "CONNECT VOLTAGE LINE",Ok
1835 ENTER 709;Ok
1839 BEEP
1846 INPUT "DISCONNECT VOLTAGE LINE",Ok
1853 IF Bv=1 THEN
1860 BEEP
1867 INPUT "INVALID VOLTAGE - TRY AGAIN!",Ok
1874 GOTO 1919
1881 END IF
1888 OUTPUT 709:""AS SA"
1895 ENTER 709;Utran
1858 OUTPUT 709; "AS SA"
1861 ENTER 709iBamp
1862 Etp=Etp*1.5E+6
1867 OUTPUT 709; "AR AF40 ALS3 VRS"
1873 Nn=7Inwt
1876 FOR I=0 TO Nn
1879 OUTPUT 709; "AS SA"
1882 IF I>7 THEN
1885 Se=0
1889 FOR K=1 TO 20
1891 ENTER 709iE
1894 Se=Se+E
1897 NEXT K
1900 Emf(I)=ABS(Se/20)
1903 ELSE
1906 ENTER 709iE
1912 Emf(I)=ABS(E)
1915 END IF
1916 Emf(I)=Emf(I)*1.5E+6
1919 NEXT I
1921 OUTPUT 709; "AS SA"
1924 OUTPUT 713; "T1R2E"
1927 WAIT 2
1930 ENTER 713; T11
1933 OUTPUT 713; "T1R2E"
1936 WAIT 2
1939 ENTER 713; T2
1942 OUTPUT 713; "T1R2E"
1945 WAIT 2
1948 ENTER 713; T12
1951 T1=(T11+T12)*.5
1954 OUTPUT 713; "T3R2E"
1958 BEEP
1961 ENTER "ENTER PRESSURE GAUGE READING (Pga)" ,Pga
1962 Pvap1= 1Psi TO Pa
1965 ELSE
1968 ENTER 0F:1c;Bvol,Bamp,Ptran,Etp,Emf(*),Fm,T1,T2,Pvap1
1977 IF J=1 OR J=20 OR J=Nrun THEN
1980 Ng=1
1983 ELSE
1986 Ng=0
1989 END IF
2002 END IF
2008 Tsteam1=FNTvs(Emf(0))
2009 Tsteam1=Tsteam1-273.15
2011 Tsteam2=FNTvs(Emf(1))
2012 Tsteam2=Tsteam2-273.15
2014 Tsteam=(Tsteam1+Tsteam2)/2.
2015 Tsteam=Tsteam1
2017 Troom=FNTvs(Emf(2))
Tcon=NTnv58(Emf(7))
Troom=Troom-273.15
Tcon=Tcon-273.15
Tum=0.
FOR I=0 TO Inw-1
Tw(I)=FTnv57(Emf(I+8))
Tw(I)=Tw(I)-273.15
Tum=Tum+Tw(I)
NEXT I
Tw=Tw/Inw
Pst=FNPvst(Tsteam)
OUTPUT 709:"AR AFE4 ALG4 VRS"
OUTPUT 709:"AS SA"  ! PRESSURE TRANSDUCER
Sa=0
FOR K=1 TO 20
ENTER 709;Etran
Sa=Sa+Etran
NEXT K
Ptan=ABS(Sa/20)
BEEP
PRESSURE IN Pa FROM TRANSDUCER
Pvap2=(-2.93504*Ptan+14.7827)*6894.7
Ptest1=Pvap1  ! GAUGE
Ptest2=Pvap2  ! TRANSDUCER
CORRECTION FOR AIR CONTAMINATION
Pv=Pvap2*1.1*10^-3  ! TRANSDUCER IN kPa
Pv=Pvap2*1.1*10^-3  ! SAT. PRESSURE IN kPa
Pv=Pvap2*1.1*10^-3  ! GAUGE IN kPa
Tsat=FTnvsp(Pvap1)
Tvap1=FTnvsp(Pvap1)
Tvap2=FTnvsp(Pvap2)
Vst=FNVvst(Tsteam)
Mu=18.015
IF If1=1 THEN Mu=197.38
IF If1=2 THEN Mu=62.05
Mfng1=100*(Pvap1-Pst)/(Pvap1-(1-(Mu/28.96))*Pst)
Mfng2=100*(Pvap2-Pst)/(Pvap2-(1-(Mu/28.96))*Pst)
IF Iov=2 THEN
PRINT
PRINT USING "10X,""Data set number = "",DD";J
OUTPUT 709;"AR AF40 AL40 VRS"
OUTPUT 709;"AS SA"
END IF
PRINT
IF Iov=2 AND Ng=1 THEN
PRINT USING "1X,"" Pst Pga Ptan Ttrans Tsat(P) N61% N62%"
PRINT USING "1X,"" (kPa) (kPa) (kPa) (C) (C) Molal Molal"
PRINT USING "1X,3(3D.DD,2X),2(3D.DD,2X),2(M3D.D,2X)";PKs,Pkg,Pkp,Tveap2,Tsat,Mfng1,Mfng2
2164 PRINT
2165 IF Mfng1>.5 THEN
2166 BEEP
2167 PRINT
2168 PRINT USING "10X,";"Energize the vacuum system"
2169 BEEP
2170 INPUT "OK TO ACCEPT THIS RUN (1-Y,0-N)?",Ok
2171 IF Ok=0 THEN
2172 PRINT
2173 BEEP
2174 PRINT
2175 DISP "NOTE: THIS DATA SET WILL BE DISCARDED!!"
2176 WAIT S
2177 GOTO 1780
2178 END IF
2179 END IF
2180 END IF
2181 IF FmQ01 OR Fm>100
2182 IFm=0
2183 BEEP
2184 INPUT "INCORRECT FM (1-ACCEPT,9-DELETE(DEFAULT))",Ifm
2185 IF Ifm=9 THEN
2186 ANALYSIS BEGINS
2187 T1=FNTvsvSB(Emf(3))
2188 T2=FNTvsvSB(Emf(4))
2189 T1=T1-273.15
2190 T1=T2-273.15
2191 T1=T2-273.15
2192 T1=T2-273.15
2193 Tdel1=T1-T1
2194 Tdel1=T1-T2
2195 Tdel1=T2-T2
2196 Tdel1=T1-T2
2197 Tdel1=T2-T2
2198 Tdel1=T1-T2
2199 Tdel1=T2-T2
2200 Tdel1=T1-T2
2201 ETp1=Emf(3)+Etp/20.
2202 Dtde=2.5931E-2-1.504E-6*Etp1+1.21701E-10*Etp1^2-5.1164E-15*Etp1^3+3.2201
2203 E-19*Etp1^4
2204 Tri=Tdte+Etp/10.
2205 T3=T1+Tri
2206 PRINT USING "1X,";"TIN1 TOUT1 TIN2 TOUT2 TIN3 TOUT3 DELT1 DELT2 DELT3 TPILE"
2207 PRINT USING "1X,";"(Teflon) (Metal) (Quartz)"
2208 PRINT USING "2X,10(2D.DD,2X)";T11,T12,T2,T1,T2,Tdel1,Tdel2,Tdel3,Tri
2209 ET1=ABS(T1-T1)
2267  Er3=ABS(T2-T1)
2268  PRINTER IS 1
2270  IF Er1>.5 OR Er3>.5 AND Im=1 THEN
2271  BEEP
2272  PRINT "QCT AND TC DIFFER BY MORE THAN 0.5 C"
2275  O=1
2278  END IF
2277  IF 0=0 AND Er1>.5 AND Im=1 THEN 1790
2279  IF 0=0 AND Er3>.5 AND Im=1 THEN 1780
2280  Er2=ABS((T2-T1)-(Tr1))/T2-T1)
2283  IF Er2>.05 AND Im=1 THEN
2286  BEEP
2289  PRINT "QCT AND T-PILE DIFFER BY MORE THAN 5%"
2292  O=2
2295  IF O=2 AND Er2>.05 AND Im=1 THEN 1780
2298  END IF
2311  PRINTER IS 781
2312  INPUT "COOLANT TEMP. RISE MEAS. (0=TEFLON, 1=METAL S., 2=QUARTZ, 3=TPILE)"
2313  ,Im
2314  IF Im=0 THEN
2315  PRINT USING "1X","USING SINGLE TEFLOM THERMOCOUPLE FOR COOLANT TEMP. RISE"
2317  Ti:=T1
2320  Ti2:=T2
2323  END IF
2324  IF Im=1 THEN
2327  PRINT USING "1X","USING SINGLE METAL SHEATHED THERMOCOUPLE FOR COOLANT TEM
2329  RISE"
2332  Ti:=T1
2335  Ti2:=T2
2338  END IF
2339  IF Im=2 THEN
2342  PRINT USING "1X","USING THERMOPILE FOR COOLANT TEMP. RISE"
2344  Ti:=T1
2347  Ti2:=T2
2350  END IF
2351  IF Im=3 THEN
2354  PRINT USING "1X","USING THERMOMETER FOR COOLANT TEMP. RISE"
2357  Ti:=T1
2360  Ti2:=T2
2363  END IF
2366  Tavg=(Ti+T2)*.5
2369  If=0
2372  Cp=FCps(Tavg)
2375  Rho=FNRh(Tavg)
2378  Md=(5.7409*Fn+13.027)/1000.
2381  Md=Md*(1.0365+1.95544E-3*Ti+5.252E-6*Ti^2)/1.0037
2384  Mf=Md/Rho
2387  Vf=Mf/(Pi*0^1.2/4)
2390  IF Im=0 AND Vf>.5 THEN T2a=T2a-(-2.73E-4+1.75E-4*Vf+9.35E-4*Vf^2-1.96E-5*
Vw^3)
2389 IF Inn=1 THEN T2o=T2o-(-6.44E-6+1.71E-3*Vw+4.45E-4*Vw^2+4.07E-5*Vw^3)
2407 IF Inn=2 THEN T2o=T2o-(-3.99E-4+2.75E-3*Vw+1.45E-3*Vw^2+8.16E-5*Vw^3)
2408 IF Inn=3 THEN T2o=T2o-(-8.57E-5+1.23E-3*Vw+1.08E-3*Vw^2+8.16E-5*Vw^3)
2413 Q=Md*Cpw*(T2o-T11)
2416 Q=Q/(PI*Do*L)
2420 T2m=T2o-(3.99E-4+2.75E-3*Vw+1.45E-3*Vw^2+8.16E-5*Vw^3)
2421 T2m=T2o-(8.57E-5+1.23E-3*Vw+1.08E-3*Vw^2+8.16E-5*Vw^3)
2422 T2m=T2o-(3.99E-4+2.75E-3*Vw+1.45E-3*Vw^2+8.16E-5*Vw^3)
2423 T2m=T2o-(8.57E-5+1.23E-3*Vw+1.08E-3*Vw^2+8.16E-5*Vw^3)
2424 ITERATE TO FIND INNER WALL TEMPERATURES
2425 T2m3=T2m
2426 T2m4=T2m
2427 T2m3=(T2m3+T2m4)/2.
2428 Ct=Q*LOG(Do/Dr)/(2.1*PI*Kcu*L)
2429 T2m3=T2m-Ct
2430 IF ABS(T2m3-T2m1)>0.001 THEN GOTO 2420
2431 T2m=0.
2432 T2m=0.
2433 FOR I=0 TO Innt-1
2434 Two(I)=Tw(I)+Ct
2435 Two(I)=Tw(I)+Ct
2436 Twoo=Twoo+Two(I)
2437 Twoo=Twoo+Two(I)
2438 NEXT I
2439 Twoo=Twoo/Innt
2440 Twoo=Twoo/Innt
2441 PRINT
2443 PRINT USING "1X,""Position number : 1 2 3 4 5 6"
2444 PRINT USING "1X,""Local outer wall temps. (DEG C) : ",6(DD, DD, 1X)";Twoo(*
2445 PRINT
2447 PRINT USING "1X,""Average outer and inner wall temps = ",2(DD, DD, 1X)";Twoo,Twoo
2448 PRINT
2450 If t=0
2451 Ku=FNKw(Tavg)
2452 Muw=FNMuw(Tavg)
2453 Muwi=FNMuw(Twi)
2454 Rei=Rhcw*Uw*Di/Muw
2457 Prw=FNPrw(Tavg)
2458 Dtw=Q*LOG(Do/Dr)/(2*PI*Kcu*L)*.5
2459 Lmt=(T2o-T11)/LOG((Twi-T11)/(Twi-T2o))
2460 Perim=PI*Do
2461 Surfai=PI*Di*L
2462 PRINT Rei,Prw,Dtw,Lmtd,Perim,Surfai,Q

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Anarea1 = \( \pi \times (D_2 - D_1)^2 / 4 \),
Anarea2 = \( \pi \times (D_2 - D_1)^2 / 4 \),
\( H_1 = Q / (\text{Surfa} \times \text{Lmd}) \),
\( Ff_1 = (Kcu \times \text{Perim} \times \text{Anarea}_1)^.5 \),
\( Ff_2 = (Kcu \times \text{Perim} \times \text{Anarea}_2)^.5 \),
PRINT Anarea1, Anarea2, Hi, Ff1, Ff2
Hi = Hi
Hi2 = Hi
Alam1 = \( \left( \frac{(Hi2 \times \text{Perim})}{(Kcu \times \text{Anarea}_1)} \right)^.5 \),
Alam2 = \( \left( \frac{(Hi2 \times \text{Perim})}{(Kcu \times \text{Anarea}_2)} \right)^.5 \),
\( Ff_3 = Ff_1 \times Hi2^2 \),
\( Ff_4 = Ff_2 \times Hi2^2 \),
Func6 = \( \text{FNTanh}(\text{Alam}_1 \times \text{L1}) \),
Func7 = \( \text{FNTanh}(\text{Alam}_2 \times \text{L2}) \),
Func8 = \( \text{Surfa} \times (\text{Tumi} - \text{Tavg}) \),
Funcx = \( \text{Func6} + \text{Func7} + \text{Func8} + Q \),
Dfunc6 = \( (\text{Ffunc6} / \text{Hi2}) + 2, * Ff3 \times \text{L1} / (1. + \text{FNCosh}(2, * \text{Alam1} \times \text{L1})) \),
Dfunc7 = \( (\text{Ffunc7} / \text{Hi2}) + 2, * Ff4 \times \text{L2} / (1. + \text{FNCosh}(2, * \text{Alam2} \times \text{L2})) \),
Dfunc8 = \( \text{Surfa} \times (\text{Tumi} - \text{Tavg}) \),
Dfuncx = \( \text{Dfunc6} + \text{Dfunc7} + \text{Dfunc8} \),
H1 = H2 = \( (\text{Funcx} / \text{Dfuncx}) \),
IF \( \text{ABS}(H1 - H2) > .05 \) THEN GOTO 2475
H1 = H1
PRINT H1
Hfg = \( \text{FNHFg}(\text{Tsteam}) \),
Tfilm = \( \text{Tsteam} / 3 + \text{Twinc} \times 2 / 3 \),
Kf = \( \text{FNNu}(\text{Tfilm}) \),
Rhof = \( \text{FNPrho}(\text{Tfilm}) \),
Cpf = \( \text{FNCpw}(\text{Tfilm}) \),
Muf = \( \text{FNPmu}(\text{Tfilm}) \),
Q1 = 500
Qloss = \( \text{Q}/((100 - 2S) \times (\text{Tsteam} - \text{Troom})) \) TO BE MODIFIED
Cpsc = \( \text{FNCpw}(\text{Tco} + \text{Tsteam}) \times .5 \),
Mdvc = 0
Bp = \( (\text{Bvcl} + 100)^2 / 5.76 \),
Hsc = \( \text{Cpsc} \times (\text{Tsteam} - \text{Tcon}) \),
Mdvc = \( (Bp - \text{Qloss}) - \text{Mdvc} \times \text{Hsc} / \text{Hfg} \),
IF \( \text{ABS}(\text{Mdvc} - \text{Mdvc}) > .01 \) THEN
Mdvc = \( \text{Mdvc} \times \text{Mdvc} \times .5 \),
GOTO 2693
END IF
Mdvc = \( (\text{Mdvc} + \text{Mdvc}) \times .5 \),
Vg = \( \text{FNUvs}(\text{Tsteam}) \),
Vv = \( \text{Mdvc} \times \text{Vg} / \text{Ax} \),
Tdco = \( \text{Tsteam} - \text{Twinc} \),
Ho = \( \text{Qp} / \text{Tdco} \),
Pr = \( \text{Muf} \times \text{Cpf} / \text{Kf} \),
Retp = \( \text{Vv} \times \text{Do} \times \text{Rhof} / \text{Muf} \),
Nuss = \( \text{Ho} \times \text{Do} / \text{Kf} \),
Nure1 = \( \text{Nuss} / \text{Retp} \times .5 \).
2639  \( Ff = 9.81 \times Do \times Muf \times Hfg / (Vv \times 2 \times Kf \times TDCF) \)
2640  \( Hnus = 0.728 \times (Kf \times 3 \times 9.81 \times Hfg \times Rhof / 2 \times (Muf \times Do \times TDCF))^{0.25} \)
2642  PRINT
2643  PRINT USING "1X," " Uinf TDCF Qf1 Ho""
2644  PRINT USING "1X,2(DD,DD,2X),2(MZ.3DE,2X)";Vv, TDCF, Qp, Ho
2645  PRINT
2646  PRINT USING "1X," " Nu Retp NuRe Pr F Hnus e""
2648  PRINT USING "1X,2(MZ.3DE,2X),2(DD,DD,2X),2(MZ.3DE,2X)";NuSS, Retp, NuRe1, Pr, Ff, Hnus
2654  Ret = Muw / Muw1
2654  Anus = H1 * D1 / Kw
2655  Aprs = Muw * Cp / Kw
2676  Stco = Anus / (Aprs \times 0.8 \times Aprs \times 3333 \times Rat \times 1.4)
2677  Nupr = Anus / (Aprs \times 3333 \times Rat \times 1.4)
2679  PRINT
2680  PRINT USING "1X," " Ucool Rec Hi Nu Nuc Stcoef f 
2681  PRINT USING "1X,6(MZ.3DE,2X)";Uw, Pei, H1, Anus, Stco, Nupr
2682  INPUT "CHANGE TCOOL RISE? 1=Y 2=N", Itr
2683  IF Itr = 1 THEN GOTO 2312
2684  OUTPUT @File1:1;Two(*)
2704  IF Im = 1 THEN
2707  BEEP
2710  INPUT "OK TO STORE THIS DATA SET (1=Y,0=N)?", Oks
2713  IF Oks = 1 THEN
2715  OUTPUT @File1:Bvol, Bamp, Piran, Etp, Emf(*), Fm, T1, T2, Pva
2717  END IF
2719  BEEP
2721  INPUT "WILL THERE BE ANOTHER RUN (1=Y,0=N)?", Igo
2724  Nrun = 1
2727  IF Igo = 1 THEN 1777
2729  ELSE
2733  END IF
2836  INPUT "ENTER PLOT FILE NAME", Fplot$
2839  ASSIGN @File4 TO Fplot$
2941  FOR I = 1 TO Nrun
2945  ENTER @File4:Qp, Ho
2946  NEXT I
3001  PRINT USING "10X," "NOTES: ", ZZ," data runs were stored in file ",10A:J, D_file$
3007  BEEP
3010  IF Ip = 1 THEN
3013  PRINT
3015  PRINT USING "10X," "NOTE: ", ZZ," X-Y pairs were stored in plot data file ",10A:J,F_file$
3019  END IF
119
3073 ASSIGN @File TO * 3076 ASSIGN @File1 TO * 3079 ASSIGN @File4 TO * 3092 IF Iso=2 THEN CALL Raw 3092 IF Iso=3 THEN CALL Purge 3115 END
3118 DEF FNPvst(Tc)
3121 COM /Fld/ Ift
3124 DIM K(8)
3127 IF Itf=0 THEN
3130 DATA -7.691234564,-26.08023696,-168.1706546,64.23295504,-118.9646225
3133 DATA 4.16711732,20.9750676,1.996.
3136 READ K(*)
3139 T=(Tc+273.15)/647.3
3142 Sum=0
3145 FOR N=0 TO 4
3148 Sum=Sum+K(N)*(1-T)^N+(N+1)
3151 NEXT N
3157 Pr=EXP(Br)
3160 P=22.20000*Pr
3163 END IF
3166 IF Itf=1 THEN
3169 Tf=Tc*1.8+32+459.6
3172 P=10^(-32.0555-4330.99/Tf-9.2635*LST(Tf)+2.0539E-3*Tf)
3175 P=P*10^1.325/14.696
3178 END IF
3181 IF Itf=2 THEN
3184 A=9.3946E-3066.4/(Tc+273.15)
3187 P=12.32*10^A
3190 END IF
3193 RETURN P
3196 FEND
3199 DEF FNHfg(T)
3202 COM /Fld/ Ift
3205 IF Itf=0 THEN
3208 Hfg=2477200-2450*(T-10)
3211 END IF
3214 IF Itf=1 THEN
3217 Tf=T+1.8+32
3220 Hfg=7.05578E+1-Tf*(4.838052E-2+1.251904E-4*Tf)
3223 Hfg=Hfg+2326
3225 END IF
3229 IF Itf=2 THEN
3232 Tk=T+273.15
3235 Hfg=1.35264E+6-Tk*(6.38293E+2+Tk*.747462)
3239 END IF
3241 RETURN Hfg
3244 FEND
3247 DEF FNMuw(T)
3250 COM /F1d/ Ift
3253 IF Ift=0 THEN
3256 A=247.8/(T+133.15)
3259 Mu=2.4E-5*10'A
3262 END IF
3265 IF Ift=1 THEN
3268 Mu=8.9629819E-4*(1.1094609E-5*T+5.566829E-8)
3271 END IF
3274 IF Ift=2 THEN
3277 Tk=T+273.15
3280 Mu=EXP(-11.0179+Tk*(1.744E+3-Tk*(2.80335E+5-Tk*1.12661E+8)))
3283 END IF
3286 RETURN Mu
3289 FNEND
3292 DEF FNPvat(Tt)
3295 COM /F1d/ If:
3298 IF Ift=0 THEN
3301 P=FNPvat(Tt)
3304 T=Tt+273.15
3307 X=1500/T
3310 F1=1/(1+T*1.E-4)
3313 F2=1-EXP(-X)*2.5*EXP(X)/X.S
3316 B=.0015*F1-.000942*F2-.0004882*X
3319 k=2*/(4E1.52*T)
3322 U=(1+(1+2*B*K)^.5)/K
3325 END IF
3328 IF Ift=1 THEN
3331 Tt=T+1.932
3334 U=13.95397-Tt*(161279E2-Tt*1726190E-4)
3337 V=U/16.019
3340 END IF
3343 IF Ift=2 THEN
3346 Tk=T+273.15
3349 P=FNPvat(Tt)
3352 V=133.95+Tk/P
3355 END IF
3358 RETURN V
3361 FNEND
3364 DEF FNCpw(T)
3367 COM /F1d/ If:
3370 IF Ift=0 THEN
3373 Cp=4.211208E-9*T*(2.2626E-3-4.4236E-5+2.71429E-7*T)
3376 END IF
3379 IF Ift=1 THEN
3382 Cp=9.250727E-1+T*(9.3400433E-4+1.7207792E-6*T)
3385 END IF
3388 IF Ift=2 THEN
3391 Tk=T+273.15
3394 Cp=4.1968*(1.6984E-2+Tk*(3.35083E-3-Tk*(7.224E-6-Tk*7.61748E-9)))
3397 END IF
3400 RETURN Cpu*1000
3403 FNEND
3406 DEF FN Rhow(T)
3409 COM /F1d/ If T
3412 IF Ift=0 THEN
3415 Ro=999.52946+T*(.01269-T*(5.402513E-3-T*1.234147E-5))
3418 END IF
3421 IF Ift=1 THEN
3424 Ro=1.6224749E+3-T*(2.2166346+T*2.3578291E-3)
3427 END IF
3430 IF Ift=2 THEN
3433 Tk=1+273.15-338.15
3436 uf=9.24648E-4+Tk*(6.2795E-7+Tk*(9.2444E-10+Tk*3.057E-12))
3439 Ro=1/uf
3442 END IF
3445 RETURN Ro
3448 FNEND
3451 DEF FN Prw(T)
3454 Prw=FN Rpw(T)*FNMw(T)/FN K(T)
3457 RETURN Prw
3460 FNEND
3463 DEF FN K(T)
3466 COM /F1d/ Ift T
3469 IF Ift=0 THEN
3472 x=(T+273.15)/273.15
3475 K=0.82377+3-x*(1.8007-x/.52577-.07344*x)
3478 END IF
3481 IF Ift=1 THEN
3484 K=0.099238E-2-T*(1.224536E-4-T*2.3809524E-8)
3487 END IF
3490 IF Ift=2 THEN
3493 Tk=1+273.15
3496 Kw=1.1883E-4*(519.442+.320920*T)
3499 END IF
3502 RETURN Kw
3505 FNEND
3508 DEF FN Tanh(X)
3511 P=EXP(X)
3514 Q=EXP(-X)
3517 Tanh=(P-Q)/(P+Q)
3520 RETURN Tanh
3523 FNEND
3525 DEF FN V (U)
3528 COM /Ct/ C (7)
3532 T=C(0)
3535 FOR I=1 TO 7
3538 T=T+C(I)*U^I
3541 NEXT I
3544 T=T+4.73386E-3-T*(7.692834E-3-T*9.077927E-5)
3547 RETURN T

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3550  FNEND
3553  DEF FNHF(T)
3556  COM /FID/ IfT
3559  IF IfT=0 THEN
3562  Hf=T*(4.203849-T*(5.88132E-4-T*4.55160317E-6))
3565  END IF
3568  IF IfT=1 THEN
3571  Tf=T*1.8+32
3574  Hf=0.2079671+Tf*(.19467857+Tf*1.321426E-4)
3577  Hf=Hf+2.326
3580  END IF
3583  IF IfT=2 THEN
3586  Hf=250    ! TO BE VERIFIED
3589  END IF
3592  RETURN Hf*1000
3595  FNEND
3598  DEF FNTVSP(P)
3601  Tu=190
3604  Ti=10
3607  Ta=Tu+Ti+.5
3610  PC=FNVST(Ta)
3613  IF ABS((P-PC)/P)> .000! THEN
3616  IF PzP THEN Ti=Ta
3619  IF PzP THEN Tu=Ta
3622  GOTO 3619
3625  END IF
3628  RETURN Ta
3631  END IF
3634  FNEND
3637  DEF FNVSSE(V)
3640  COM /CSE/ TSS(S)
3643  T=TSS(0)
3646  FOR I=1 TO S
3649  T=T+TSS(I)*V*I
3652  NEXT I
3655  RETURN T
3658  FNEND
3661  DEF FNVSSE(V)
3664  COM /CSE/ TSS(S)
3667  T=TSS(0)
3670  FOR I=1 TO S
3673  T=T+TSS(I)*V*I
3676  NEXT I
3679  RETURN T
3682  FNEND
3685  DEF FNVSST(V)
3688  COM /CST/ TSST(S)
3691  T=TSST(0)
3694  FOR I=1 TO S
3697  T=T+TSST(I)*V*I
3699  NEXT I
3702  RETURN T
3705  FNEND
3708  DEF FNVSST(V)
3711  COM /CST/ TSST(S)
3714  T=TSST(0)
3717  FOR I=1 TO S
3720  T=T+TSST(I)*V*I
3723  NEXT I
3726  RETURN T
3729  FNEND
3732  DEF FNVSST(V)
3735  COM /CST/ TSST(S)
3738  T=TSST(0)
3741  FOR I=1 TO S
3744  T=T+TSST(I)*V*I
3747  NEXT I
3750  RETURN T
3753  FNEND
3756  FOR I=1 TO S
3759  T=T+TSST(I)*V*I
3762  NEXT I
3765  RETURN T
DEF FNT\text{vs}vS8(V)
\text{COM} /CcS8/ TS8(S)
T=TS8(0)
FOR I=1 TO 5
T=T+TS8(I)*V"I
NEXT I
RETURN T
FNEND

DEF FNCosh(X)
P=\text{EXP}(X)
Q=\text{EXP}(-X)
Cosh=.5*(P+Q)
RETURN Cosh
FNEND

SUB Raw
\text{COM} /F1d/ Ift
DIM Emf(13)
BEEP
INPUT "ENTER FILE NAME",File$
ASSIGN @File TO File$
BEEP
INPUT "ENTER PRESSURE CONDITION (0=V, 1=A)" ,Ipc
INPUT "ENTER NUMBER OF RUNS",Nrun
PRINTER IS 701
PRINT
PRINT USING "10X,""File Name:    ",14A";File$
IF Ipc=0 THEN
PRINT USING "10X,""Pressure Condition: Vacuum""
ELSE
PRINT USING "10X,""Pressure Condition: Atmospheric""
ENDIF
ENTER @File;Ilg,Inn
ENTER @File;Inwt,Fp,Fw,Fh
PRINT
PRINT USING "10X,""Data Vw Tin Tout Ts""
FOR I=1 TO Nrun
ENTER @File;Bvol,Bamp,Ptran,Etp,Emf(*),Fn,T1,T2,Pvap1
PRINT Bvol,Bamp,Ptran,Etp,Emf(*),Fn,T1,T2,Pvap1
NEXT I
ASSIGN @File TO *
SUBEND
SUB Purg
BEEP
INPUT "ENTER FILE NAME TO BE DELETED",File$
PURGE File$
GOTO 3908
SUBEND
LIST OF REFERENCES


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