Two phase thermosiphon loops: part 1 (an experimental study of a 3/8 in. diameter 2ft-4ft loop).

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TWO PHASE THERMOSIPHON LOOPS - PART 1

(AN EXPERIMENTAL STUDY OF A 3/8 IN. DIAMETER 2 FT * 4 FT LOOP)

A THESIS

Submitted to the Faculty of Graduate Studies through the Department of Mechanical Engineering in Partial Fulfilment of the Requirements for the Degree of Master of Applied Science at the University of Windsor

By

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1976
TO MY WIFE
ABSTRACT

An experimental investigation was carried out to study the performance of two-phase thermosiphon loop. The loop performance was characterized by defining a loop conduct-and "U", the heat transfer rate per unit of evaporator tube inside surface area, and per unit of temperature difference between the evaporator and condenser tube surfaces.

An instrumented loop, rectangular in shape, having 1.22 m (4') long evaporator and condenser tubes and 0.61 m (2') long interconnecting tubes of 3/8 inch (7.6 mm) diameter was designed, built and tested. Tests were carried out to determine the effect on the loop conductance of; loop orientation within a gravity field, the percentage charge of working fluid (90%-7%), the source and sink temperatures (75-15°C), and the working fluid (R-113 and R-11).

For a given working fluid the optimum loop performance is achieved when the % charge and orientation of the loop are such that none of the condenser is submerged and dryout occurs at the top of the evaporator.

A simplified analytical model of a thermosiphon loop was developed primarily to study the effect of evaporator dryout and condenser flooding on the loop conductance. The model illustrates these effects very well.
ACKNOWLEDGEMENTS

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# TABLE OF CONTENTS

| ABSTRACT                                      | iv  |
| ACKNOWLEDGEMENTS                              | v   |
| TABLE OF CONTENTS                             | vi  |
| LIST OF FIGURES                               | x   |
| NOMENCLATURE                                  | xii |
| CHAPTER                                       |     |
| I    INTRODUCTION                             | 1   |
| 1.1   GENERAL                                 | 1   |
| 1.2   INVESTIGATION AT UNIVERSITY OF WINDSOR   | 4   |
| 1.3   PRINCIPLE OF OPERATION OF THERMOSIPHON LOOP | 5   |
| 1.4   LITERATURE SURVEY                       | 11  |
| 1.5   CO-CURRENT STUDIES                      | 13  |
| II   DEVELOPMENT OF A SIMPLIFIED THERMOSIPHON LOOP MODEL | 14  |
| 2.1   MODEL ASSUMPTION                        | 14  |
| 2.2   CRITIQUE OF MODEL                       | 14  |
| 2.3   MODEL DEVELOPMENT                       | 16  |
| III  APPARATUS AND INSTRUMENTATION            | 21  |
| 3.1   LOOP ASSEMBLY                           | 21  |
| 3.1.1 Main flow loop                         | 21  |
| 3.1.2 Source and Sink Heat Exchanger Design   | 24  |
| 3.1.3 Dial pressure gage                      | 26  |
| 3.1.4 Auxiliary vacuum chamber                | 26  |
| 3.1.5 Insulation                             | 28  |
3.2 LOOP ORIENTATION MEASUREMENT DEVICES 28
3.3 LEAKAGE TEST 29
  3.3.1 Flow Loop 29
  3.3.2 Annulus 29
3.4 HEAT SOURCE AND SINK 29
3.5 AUXILIARY CHARGING APPARATUS 30
3.6 DATA ACQUISITION SYSTEM 33
  3.6.1 Indirect Analysis Method 33
  3.6.2 Direct Analysis Method 36
3.7 THERMOCOUPLE INSTRUMENTATION CALIBRATION 36

IV EXPERIMENTAL PROCEDURE 40
  4.1 TESTING PROCEDURE 40
  4.2 DETAILED DESCRIPTION OF PARAMETER VARIATION 44
    4.2.1 Data Acquisition Tests 44
    4.2.2 Test Sequence I 44
    4.2.3 Test Sequence II 45
    4.2.4 Test Sequence III 45
    4.2.5 Test Sequence IV 45
    4.2.6 Test Sequence V 46
    4.2.7 Test Sequence VI 46
    4.2.8 Test Sequence VII 46
  4.3 CALCULATION PROCEDURE 46
    4.3.1 Loop Conductance 46
      i) Calculation of energy transport 47
      ii) Calculation of temperature difference 50
    4.3.2 Percentage Charge 57
V RESULTS AND DISCUSSION

5.1 EFFECT OF CHARGE AND LOOP ORIENTATION

5.1.1 Effect of % Charge and Rotation angle

5.1.2 Inclination Angle of the loop

5.2 COMPARISON OF R-113 AND R-11

5.3 EFFECT OF PRESSURE

5.4 EFFECT OF TEMPERATURE DIFFERENCE

5.5 EFFECT OF CHANGING THE SINK TEMPERATURE

5.6 SIMULATION OF THE EXPERIMENTAL LOOP

5.7 RILIABILITY OF THE EXPERIMENTAL RESULTS

5.7.1 Reproducibility

5.7.2 Probable error in calculated results

VI CONCLUSION

REFERENCES

APPENDICES

APPENDIX A: List of equipment and Instrumentation

APPENDIX B: Computer Program for "Indirect Analysis"

APPENDIX C: Computer Output for "Indirect" Analysis

APPENDIX D: Computer Program for "Direct" Analysis

APPENDIX E: Computer Output for "Direct" Analysis

APPENDIX F: Computer Output for "Indirect" thermocouple calibration

APPENDIX G: Computer Output for "Direct" thermocouple calibration
<table>
<thead>
<tr>
<th>APPENDIX</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>H</td>
<td>Calculations of the boiling and condensation lengths</td>
<td>96</td>
</tr>
<tr>
<td>I</td>
<td>Error Analysis</td>
<td>99</td>
</tr>
<tr>
<td>J</td>
<td>Model of experimental loop</td>
<td>106</td>
</tr>
<tr>
<td>K</td>
<td>Price comparison of two phase thermosiphon with a recuperator</td>
<td>113</td>
</tr>
<tr>
<td>L</td>
<td>Tabulated Test Results</td>
<td>118</td>
</tr>
<tr>
<td>M</td>
<td>Table of Properties for R-113 and R-11</td>
<td>133</td>
</tr>
<tr>
<td></td>
<td>VITA AUCTORIS</td>
<td>134</td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>-----------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>1</td>
<td>Thermosiphon loop heat exchanger</td>
<td>3</td>
</tr>
<tr>
<td>2</td>
<td>Thermosiphon tube heat exchanger (Larkin's)</td>
<td>12</td>
</tr>
<tr>
<td>3</td>
<td>A vertical cross section through an inclined condenser tube</td>
<td>20</td>
</tr>
<tr>
<td>4</td>
<td>Loop assembly</td>
<td>22</td>
</tr>
<tr>
<td>5</td>
<td>End cap assembly</td>
<td>23</td>
</tr>
<tr>
<td>6</td>
<td>Thermocouple tube installation</td>
<td>25</td>
</tr>
<tr>
<td>7</td>
<td>Thermocouple seal plug</td>
<td>27</td>
</tr>
<tr>
<td>8</td>
<td>Loop orientation measurement device</td>
<td>31</td>
</tr>
<tr>
<td>9</td>
<td>Heat sink bath</td>
<td>31</td>
</tr>
<tr>
<td>10</td>
<td>Schematic Diagram of Deareation Unit</td>
<td>32</td>
</tr>
<tr>
<td>11</td>
<td>Schematic Diagram of Distillation Unit</td>
<td>34</td>
</tr>
<tr>
<td>12</td>
<td>Data acquisition unit</td>
<td>35</td>
</tr>
<tr>
<td>13</td>
<td>Experimental Test Apparatus</td>
<td>39</td>
</tr>
<tr>
<td>14</td>
<td>Heat Losses vs Temperature Levels above ambient</td>
<td>49</td>
</tr>
<tr>
<td>15</td>
<td>Loop Conductance vs Rotation angle for R-113</td>
<td>53</td>
</tr>
<tr>
<td>16</td>
<td>Loop Conductance vs Rotation Angle for R-11</td>
<td>54</td>
</tr>
<tr>
<td>17</td>
<td>Loop Conductance vs Charge for R-113</td>
<td>55</td>
</tr>
<tr>
<td>18</td>
<td>Loop Conductance vs Charge for R-11</td>
<td>56</td>
</tr>
<tr>
<td>19</td>
<td>Effect of Inclination Angle for R-113</td>
<td>59</td>
</tr>
<tr>
<td>20</td>
<td>Comparison of R-11 and R-113</td>
<td>61</td>
</tr>
<tr>
<td>21</td>
<td>Effect of System pressure for R-113</td>
<td>64</td>
</tr>
<tr>
<td>22</td>
<td>Effect of System pressure for R-11</td>
<td>65</td>
</tr>
<tr>
<td>23</td>
<td>Effect of temperature difference for R-11</td>
<td>67</td>
</tr>
</tbody>
</table>
Effect of Changing sink temperature for R-113 70
Comparison of the Simulated and Experimental U 72
for R-113
Comparison of the Simulated and Experimental U 73
for R-11
Reproducibility of the experimental results 75
Reciprocal of the liquid fraction vs rotation angle 111
Effect of Rotation on Static Liquid Levels in a 112
Rectangular Loop
NOMENCLATURE

A - Surface area (m²)
C₁ - Coefficient in Eq. (J-1)
C₂ - Coefficient in Eq. (J-2)
Cₖ - Coefficient in Eq. (1-8)
Cₜ₃ - Coefficient in Eq. (1-7)
Cₕ₄ - Specific heat of saturated liquid (J/Kg°K)
D - Tube diameter (m)
g - Acceleration due to gravity (m/s²)
gₙ - Conversion factor (4.17 * 10⁻⁸ lbm ft/lb₆ hr²)
G - Mass flow flux (Kg/s·m²)
f - Function
h - Heat transfer coefficient (w/m²°K)
hₜₚₔ - Latent heat (J/Kg)
K - Thermal conductivity (w/m²°K)
Kₚ - Proportionality constant defined by Eq. 1-9d
L - Length (m)
ṁ - Mass flow rate (Kg/s)
P - Pressure (N/m²)
ΔP - Pressure drop (N/m²)
Pr - Prandtl number
Q - Heat flow rate (w)
q - Heat flux (w/m²)
Re - Reynold's number
Rₜ - Transport resistance (m²°K/w)
r - Exponent of Eq. (1-8)
\( s \) - Exponent of Eq. (1-8)

\( S_{fg} \) - Entropy of vaporization (J/Kg\(^{\circ}\)K)

\( T \) - Temperature (\(^{\circ}\)C)

\( U \) - Loop conductance (w/m\(^{2}\)K)

\( V_{fg} \) - Latent specific volume (m\(^3\)/Kg)

\( x \) - Quality

\( z \) - Length (m)

\( \rho \) : Density (Kg/m\(^3\))

\( \rho_m \) : Mean density (Kg/m\(^3\))

\( \rho_v \) : Density of saturation vapour (Kg/m\(^3\))

\( \mu \) : Viscosity (Kg/m-s)

\( \theta_1 \) : Angle of inclination from the vertical (degrees)

\( \theta_R \) : Angle of rotation from the vertical (degrees)

\( \zeta \) : Fin efficiency

\( e \) : Effectiveness

\( \sigma \) : Surface tension (N/m)

\( \delta \) : Condensation film thickness

**Subscripts**

\( a \) - Air

\( b \) - Boiling region in the evaporator

\( c \) - Film condensation or condenser

\( c_\ell \) - Subcooled liquid area in the condenser or heat loss from the condenser

\( cw \) - Cold water
CHAPTER I
INTRODUCTION

1.1 GENERAL

In recent years there has been increasing interest in developing new energy sources and conserving our present energy due to decreasing energy resources throughout the world. One method of conserving energy is through the recovery of waste heat from hot exhaust gases such as from power plant boilers, gas turbines, and ventilation air in large buildings.

Such energy transfers are achieved through the use of direct-heat transfer heat exchangers (recuperators) or rotary regenerative heat exchangers or liquid-coupled indirect-transfer units.

The recuperator and rotary regenerator permit high heat transfer rates for given weight and space limitations. However, both of these systems require that hot and cold stream ducts should be adjacent to each other and hence require special ducting considerations. In addition the rotary regenerator permits some mixing of the two fluid streams. These problems do not arise in the liquid-coupled indirect-transfer heat exchangers, but they do require circulation of the coupling liquid and generally occupy a greater space and are of heavier construction.
A new alternative concept for a coupled indirect-transfer type exchanger is to use a two-phase thermosiphon loop, as shown in figure 1. Such loops have been undergoing study for the past two years at the University of Windsor.

In contrast to the earlier liquid-coupled type, the two-phase thermosiphon loop type employs a naturally circulating liquid. The loop is filled with a fixed mass of working fluid, part liquid and part vapour, which is used to transport the thermal energy picked up in the hot stream to the cold stream.

The potential advantages of the two-phase thermosiphon loop type relative to the earlier liquid-coupled type are as follows:

a) No power requirement to circulate working fluid.

b) Considerably lighter since the unit is only partially filled with liquid.

c) Noiseless operation.

d) Can be made either bidirectional or unidirectional.

To be reversible (bidirectional) both evaporator and condenser must be at essentially the same elevation.
Fig. 1  THERMOSIPHON LOOP HEAT EXCHANGER
1.2 INVESTIGATION AT UNIVERSITY OF WINDSOR

The ultimate goal of the study being carried out at the University of Windsor is to be able to predict the performance of any two phase thermosiphon loop through the use of a computer simulation program which would be verified using experimental data.

The independent parameters that affect the loop performance are: the size and shape of the loop; its orientation in a gravity field; the operating pressure, properties and percent charge of the working fluid; and the heat flux.

In this experimental study, hot and cold water were used as the energy source and sink respectively. This permitted the evaporator and condenser tube surface temperatures to vary along their length in response to internal flow conditions as it would in a real gas-to-gas heat exchanger installation. The use of water permitted easy control of the source and sink temperature between 16°C (60°F) and 81°C (180°F).

Refrigerants, R-113 and R-11, were chosen as the working fluids to be tested, because they are commonly used refrigerants, both have saturation pressures which are within the limit of safe operation of the loop between 16°C and 81°C, and both are nontoxic.

A loop fixed size and shape was constructed which permitted a study of the effect of

(a) Loop orientation
(b) Percent charge of working fluid
(c) System pressure (operating temperature level).

(d) Heat flux on the performance of the loop.

1.3 PRINCIPLE OF OPERATION OF A THERMOSIPHON LOOP

A thermosiphon loop is a closed system charged with a known mass of working fluid, (see figure 2). The ratio of this mass to the mass charge when full defines the % charge.

When a portion of the loop containing liquid is heated its temperature and hence local vapour pressure start to rise. The resulting difference in vapour pressure causes the vapour to flow from the high temperature (higher pressure) region, where it is replenished by evaporation, to the low temperature (lower pressure) region, where it condenses. If the condenser region is at the same or a higher elevation than the evaporator region then the liquid will return by gravity to the evaporator.

The transport of thermal energy is thus controlled by convection and boiling mechanisms in the evaporator, by condensation and convection mechanisms in the condenser and by the flow resistance in the loop. These three factors control the saturation pressures, the temperature drop between the evaporator and condenser and the mass flow rate around the loop.
The performance of such thermosiphon loops may be characterized by defining a loop conductance "U" and loop resistance "R" as follows:

\[ UA_e = \frac{Q}{(T_e - T_c)} = \frac{1}{R} \text{, w/}^\circ \text{K} \quad 1-1 \]

where \( Q \) is the heat transferred from the evaporator to the condenser, \( (T_e - T_c) \) is the average temperature difference between the evaporator and condenser tube surfaces and \( A_e \) is the interfacial surface area between the evaporator tube and the working fluid.

In order to better understand the effect of various factors on the loop performance, it is instructive to model the real system by a loop where the working fluid in the evaporator is isothermal at its saturation temperature \( T_{ef} \) and where the working fluid in the condenser is also isothermal at its saturation temperature \( T_{cf} \). For such a model the thermal resistance of the loop, \( R \), is equal to the sum of the evaporator, condenser and transport resistance.

\[ R = \frac{1}{(UA_e)} = R_{\text{evaporator}} + R_{\text{transport}} + R_{\text{condenser}} \quad 1-2a \]

or

\[ R = \frac{1}{(\overline{h}_e A_e)} + \frac{(T_{ef} - T_{cf})}{Q} + \frac{1}{(\overline{h}_c A_c)} \text{, } ^\circ \text{K/w} \quad 1-2b \]

where

\[ \overline{h}_e A_e = \int_0^{A_e} h_e \, dA_e = (h A)_{el} + (h A)_b + (h A)_d \quad 1-3 \]
and
\[ \overline{h} \frac{A_c}{A_c} = \int_0^1 h_c \, dA_c = (h_A)_{C} + (h_A)_{CL} \]

represent the average conductance in the evaporator and condenser respectively and are made up of the sum of the \( hA \) products for each region along the tube.

In the evaporator there may be a subcooled liquid convection region, \((h_A)_{eL}\), a boiling two-phase flow region, \((h_A)_{b}\) and a dryout vapour superheat region, \((h_A)_{d}\). In the condenser two regions are significant, the vapour condensation region \((h_A)_{C}\) and the liquid subcooled convection region \((h_A)_{CL}\).

The proportions of \( A_{eL}, A_b \) and \( A_d \) in the evaporator and \( A_C \) and \( A_{eL} \) in the condenser are functions of the orientation, the % charge in the loop, and the void fraction in the evaporator, which in turn is affected by the heat flux \( Q \).

The forced convection coefficients \( h_{eL}, h_d \) and \( h_{CL} \) are dependent upon the flow conditions and the fluid properties.

In the laminar range a commonly used equation that was proposed by Sieder and Tate [15] is
\[ h_v = 1.86 \left( \frac{k_x}{D} \right) \left( \frac{Re}{Pr} \right)^{\frac{1}{3}} \left( \frac{\mu_L}{\mu_w} \right) 0.14 \]

for turbulent flow the Dittus and Beelter [15] equation can be used:
\[ h_v = 0.023 \left( \frac{k_x}{D} \right) Re^{0.8} Pr^n \]
where \( n = 0.4 \) for heating

\( n = 0.3 \) for cooling.

In both these equations the fluid properties are evaluated at the bulk free stream temperature.

For the boiling heat transfer region, Rohsenow and Choi [12] correlation for pool boiling may be used to show the parameters which effect \( h_b \).

\[
\frac{C_A(T_w - T_{sat})}{h_{fg}} = C_{sf} \left\{ \frac{q_A}{\mu_k h_{fg}} \left( \frac{g_\alpha \sigma}{g(\rho_\alpha - \rho_g)} \right)^\frac{1}{2} \right\}^r \left( Pr \right)^s
\]

where \( C_{sf} \) and \( r \) are empirical constants which depend upon the nature of the heating surface and working fluid and the exponent, "s" accounts for surface cleanliness. Tang [10] and Ali [16] recommended the following values for \( C_{sf} \), \( r \) and \( s \):

For Copper - R-113

\( C_{sf} = 0.0037 \) \( r = 0.38 \) \( s = 1.7 \) Tang

For Copper - R-11

\( C_{sf} = 0.01 \) \( r = 0.77 \) \( s = 1.7 \) Ali

Rohsenow and Choi [12] equation for the average heat transfer coefficient for vapour condensing is:

\[
\bar{h}_c = C_c \left[ \frac{\rho_\alpha (\rho_\alpha - \rho_g) g h_{fg} k_\alpha^3}{k_\alpha \left( T_{sat} - T_w \right)} \right]^{\frac{1}{4}}
\]

where \( h_{fg} = h_{fg} + 3/8 C_\alpha (T_{sat} - T_w) \).
For a horizontal tube

\[ C_c = 0.725 \text{ (Rohsenow and Choi [12])} \]
\[ C_c = 0.555 \text{ for R-113 (Chato [12])} \]

For:

\[ C_c = 0.943 \text{ (Nusselt [14] used the height and width L was used instead of D)} \]

The factors which have an effect on the transport resistance \( R_t \) may be determined by utilizing the Clausius Clapeyron equation

\[
\left( \frac{\partial P}{\partial T} \right)_v = \frac{\Delta P_{\text{sat}}}{\Delta T_{\text{sat}}} = \frac{S_{fg}}{V_{fg}} \quad 1-9a
\]

or

\[
\Delta P_{\text{sat}} = \left( \frac{S_{fg}}{V_{fg}} \right) \Delta T_{\text{sat}} \quad 1-9b
\]

For the case where the working fluid leaves the evaporator dry and saturated and enters as a saturated liquid then the transport resistance, \( R_t \) can be written as

\[
R_t = \frac{\Delta T_{\text{sat}}}{Q} = \frac{\Delta T_{\text{sat}}}{\dot{m} h_{fg}} \quad 1-9c.
\]

where \( \dot{m} \): mass flow rate of the working fluid.

The mass flow rate may be expressed as

\[
\dot{m} = \frac{\rho g}{k} (\Delta P_{\text{sat}})^n \quad 1-9d
\]
where \( 0.5 \leq n \leq 1.0 \)

\( K \): a tube "resistance" term, defined by this equation.

Eliminating \( \dot{m} \) and then \( \Delta P_{sat} \) in equation 1-9c using eqns. 1-9a & 1-9b yields

\[
R_t = \frac{K \Delta T_{sat}}{\rho_g (\Delta P_{sat})^n h_{fg}} = \frac{K (\Delta T_{sat})^{1-n}}{T_{sat}} \left( \frac{v_{fg}}{s_{fg}} \right)^{n+1}
\]

1-9e

where \( \rho_g = \frac{1}{v_{fg}} \).

In an operating loop, the ratio of \( \frac{v_{fg}}{s_{fg}} \) is dependent upon the working fluid used and on the operating temperature of the loop. This ratio decreases as the operating temperature increases. The temperature drop, \( \Delta T_{sat} \), is directly proportional to the heat flux. The tube resistance \( K \) is dependent on the fluid properties and the loop geometry.
1.4 LITERATURE SURVEY

A review of past literature reveals no reported work on closed two-phase thermosiphon loops with which the present data could be compared. Some investigations [1,2,3,4] have been carried out using a closed two-phase thermosiphon tube as shown in figure (2). The thermosiphon tube differs from the loop studied in this thesis in that in the tube, the condensate must return by gravity counter to the flow of the vapour to the heated region through the same tube whereas in the loop no net counter flow occurs. Cohen, H. and Bayley, F.J. [1] earlier suggested that the thermosiphon tube could be utilized to cool gas-turbine blades.

Larkin, B.S. [3,4,5] has studied this type of closed two-phase thermosiphon tube. His tests were carried out using a uniform heat flux power input. He found that the optimum performance of the tube was obtained for the least percentage charge of working fluid at which dryout did not occur in the heated length. In tests using different working fluids, a higher performance was achieved for working fluids having higher thermal conductivity and lower viscosity for the liquid state as well as higher latent heat of vaporization and higher saturation pressure, at a given temperature. For instance his tube conductance using R-21 was approximately 40% higher than when using R-11. The tube conductance was found to increase slightly with an increase in operating temperature (pressure) level.
FIG. 2  THERMOSIPHON TUBE HEAT EXCHANGER (LARKIN'S)
1.5 CO-CURRENT STUDIES

Mr. Ali [17] is currently developing a computer simulation program for thermosiphon loops. The data presented in the present thesis is being utilized to check this program.

Mr. DiCiccio [16] has carried out an experimental study on a similar but smaller instrumented rectangular loop 0.61 m (2') high with 0.31 m (1') separation between annular flow evaporator and condenser sections. This loop was designed to provide further data for comparison with the simulation program and also to provide for visualization of the flow in the evaporator and condenser sections.
CHAPTER II

2. DEVELOPMENT OF A SIMPLIFIED THERMOSIPHON LOOP MODEL

A simple mathematical model is postulated for the prediction of the conductance of a two-phase, rectangular thermosiphon loop as a function of the percent charge and rotation angle of the loop.

2.1 MODEL ASSUMPTIONS

The assumptions incorporated into the model are:

i) That the operation of the loop can be characterized by three temperatures; the evaporator tube surface temperature, \( T_e \), the working fluid temperature, \( T_f \), and the condenser tube surface temperature, \( T_c \).

ii) That boiling takes place on all the wetted surface areas in the evaporator.

iii) That the liquid level on the cool side of the loop is set equal to the static liquid level. The wetted level on the hot side of the loop is given by its static liquid level divided by a mean liquid fraction for the wetted region in the evaporator, or the height of the loop, whichever is the least.

iv) The cross connecting tubes are adiabatic.

2.2 CRITIQUE OF MODEL:

The assumption that the temperature of the working fluid does not change as it is circulated around the loop requires that all energy transfers are accommodated by changes in phase.
This would correspond to an operation with no superheating in the dryout region, no transport resistance, $R_t$, and no heat transfer in any submerged region of the condenser. The requirement that there is no heat transfer in the dryout region is not critical, since the heat transfer coefficient, $h_d$, is approximately two orders of magnitude smaller than the boiling heat transfer coefficient.

The transport resistance is a function of the working fluid saturation temperature difference between the evaporator and the condenser. The saturation temperature difference is a function of the pressure drop between the evaporator and the condenser which in turn is dependent on; the tube size, the loop geometry, the mass flow rate of the working fluid and the properties of the working fluid. The net effect of these variables on the transport resistance is complex and not easily predicted. It is expected that the transport resistance is negligible when only vapour with a moderate velocity is present. However, the transport resistance will be significant when liquid must be pumped from the evaporator to the condenser.

The assumption that the submerged region in the condenser is adiabatic is reasonably good, since the heat transfer coefficient in this region is one order of magnitude smaller than the film condensation heat transfer coefficient.

The assumption that boiling starts at the bottom of the evaporator will be valid only when the wall temperature is higher than the saturation temperature of the working fluid at the bottom of the evaporator. Since the pressure at the
bottom of the evaporator increases with liquid depth, the saturation temperature also increases. Thus the assumption that boiling starts at the bottom of the evaporator would not be valid for long evaporators or small temperature differences.

The assumption that condensate liquid level in the condenser is based on the static liquid level is thought to be reasonably valid. Two opposing factors affect the liquid level in the condenser. One is that when the loop is operating the liquid levels in both the evaporator and the condenser drop below these static liquid levels, because some liquid is now present as condensate coating the tube in the vapour region of the condenser. The other is that a hydrostatic pressure difference is required between the condenser and the evaporator to cause the working fluid to circulate in the loop. This increases the liquid level in the condenser.

The assumption that the wetted boiling length is longer than the static wetted length is reasonable, since the density of the two-phase mixture in the evaporator is less than the density of the liquid.

2.3 MODEL DEVELOPMENT

Application of the modelling assumptions to equation 1-2a, b reduce them to the form

$$R = \frac{1}{UA_e} = R_{\text{evaporator}} + R_{\text{condenser}} \quad \ldots 2-1a$$

and

$$\frac{1}{UA_e} = \frac{1}{h_b A_b} + \frac{1}{h_c A_c} \cdot \frac{K}{h} \quad \ldots 2-1b$$
respectively, where $A_e$ is the evaporator tube inside surface area, $A_b$ is the wetted area in the evaporator, and $A_c$ is the film-condensation area.

By rearranging Eqn. 2-1b it is possible to solve for the loop conductance $U$ as a function of $h_c$ and two dimensionless groupings.

$$U = \frac{h_c}{\frac{h_c A_e}{h_b A_b} + \frac{A_e}{A_c}} \text{ w/m}^2\text{k} \quad ...2-2$$

where

$$\frac{A_b}{A_e} = \frac{Z_{bs}}{Z_e} \cdot (\frac{Z_b}{Z_{bs}}) \quad ...2-2a$$

$$\frac{A_c}{A_e} = \frac{Z_c}{Z_e} \quad ...2-2b$$

where $Z_e$ is the length of the evaporator, $Z_b$ is the wetted length in the evaporator, $Z_{bs}$ is the static wetted length in the evaporator, and $Z_c$ is the film condensation length.

The ratios $(Z_{bs}/Z_e)$ and $(Z_c/Z_e)$ are functions of percentage charge of the loop and the rotation angle of the loop. The functional relationship is developed in Appendix H.

The ratio of $(Z_b/Z_{bs})$, which represents the amount by which the wet mixture expands due to the presence of vapour bubbles, is a function of the temperature difference between the wall and the working fluid, saturation temperature, the physical properties
of the working fluid, the percent charge and the rotation angle. For this model \( Z_b/Z_{bs} \) was assumed to be only a function of the rotation angle since none of the other parameters were varied sufficiently to determine their effect. This empirical functional relationship is presented and discussed in Appendix J. All other effects were neglected.

The evaporator heat transfer coefficient, \( h_b \), is a function of two phase forced convection with boiling. The heat transfer coefficient for nucleate pool boiling varies directly with the excess temperature between the wall of the evaporator and the working fluid saturation temperature. The heat transfer coefficient for two phase forced convection is a function of the mass flow rate and the void fraction of the working fluid. Two-phase forced convection becomes the dominant mode of heat transfer for large mass flow rates. The mass flow rate in the loop increases with the amount of heat being transferred from the evaporator to the working fluid. For low % charges when dryout occurs, more liquid is located in the evaporator when the loop is rotated. This causes the super heated region (dryout) to decrease and hence increase the heat transfer rate, and hence mass flow rate.

When the loop rotation angle is near the horizontal position, two factors may cause a decrease in the mass flow rate within the loop. One is that some liquid is carried over from the evaporator to the condenser to cause the pressure drop in the cross connecting tube to change. The other is that
the working fluid may flow in a slug or stratified manner. Thus the evaporator heat transfer coefficient, \( h_b \), may be expected to increase with the rotation angle and then decrease again. The functional relationship is developed in Appendix J.

In the condenser, the film condensation coefficient, \( h_c \), is dependent on the film thickness of the condensate, \( \frac{K_p}{\delta} \). A vertical cross section through an inclined condenser tube is shown in figure (3). Condensate runs down from the top of the ellipse collecting on the bottom. This passage is shorter than the condenser tube length. Thus the distance that the condensate must travel is much shorter when it flows over an elliptical path to the collecting stream than when the film must flow vertically down the tube wall with an ever increasing thickness until it reaches the level of the liquid condensate pool. The condensate flow path length continues to decrease as the tube reaches a horizontal orientation. On the other hand when the slope of the condenser tube is very small the depth of condensate on the bottom of the tube becomes greater thus inhibiting the heat transfer. Thus the condenser heat transfer coefficient, \( h_c \), may be expected to increase with rotation angle and then decrease again near the horizontal.

Chato [12] also found experimentally that at low vapour velocity (inlet Reynolds No. < 35,000), the condensation coefficient is a maximum between 70° and 80° from the vertical and the value is 10 to 20% higher than the horizontal.

The functional relationship between \( h_c \) and \( \theta_R \) is developed in Appendix J.
CHAPTER III

APPARATUS AND INSTRUMENTATION

3.1 LOOP ASSEMBLY

Since the experimental study was to be used to verify a computer simulation, the size and geometry of the loop were arbitrarily selected.

In order to investigate as many parameters as possible the loop was mounted on a plane which was capable of any orientation in a gravity field and capable of operating with any predetermined percent charge of working fluid.

The general configuration of the loop is shown in figure 4.

3.1.1 Main flow loop

The main flow loop was made of 7.9 mm inside diameter soft copper tubing (0.311 in. I.D., 3/8" O.D.) in a rectangular shape 1.22 meters (4 feet) long by 0.61 meters (2 feet) wide.

The evaporator and condenser tubes were joined to the two cross connecting tubes as shown in figure 5. The end caps were sealed by means of Neoprene Rubber-0-rings on which Leak Lock, a Freon sealant, was painted before assembly.

Use of these end caps facilitated interchangeability of evaporator, condenser or either of the two cross connecting tubes. Three thermocouples (24 gauge) were imbedded in the outside surface of both the evaporator and the condenser in order to measure the temperature drop between the evaporator
Fig. 4 LOOP ASSEMBLY
Fig. 5 End Cap Assembly
and the condenser tube surfaces.

Thermocouples were located at the mid point of each tube and 0.46 M above and below the mid point as shown in figure 4.

In order to ensure accurate readings, grooves were milled circumferentially on the tubes. The two thermocouple heads were fixed in the groove as shown in figure 5 about 2 mm apart by pressing the copper tube over the head. They were then silver soldered in place and the surface smoothed with a fine emery cloth to reduce surface variation.

The two thermocouples, which were used to measure the temperature of the working fluid between the evaporator and condenser at the top and at the bottom of the loop were mounted within the cross connecting tubes at their mid point using the thermocouple lead wire holder "A" shown in figure 7.

To prevent leakage of the working fluid between the thermocouple wire and its insulation the insulation was removed over a 1/8" length within holder "A" then the region surrounding the wires filled with Epoxilite 813 supplied by William T. Bean Inc. This ensured that a positive seal could be achieved and maintained.

3.1.2 Source and Sink Heat Exchanger Design

For ease of thermal control and construction it was decided to jacket the evaporator and condenser tubes with water flowing in an annular section. The assembly is shown in figure 4.

The thermocouple wires from the evaporator and con-
Fig. 6 Thermocouple tube installation
denser tubes were each bundled together and lead out through the top end tap using the thermocouple lead wire holder "B" shown in figure 7, which was then filled with G.E. Silicone Sealant.

The water jacket supply and return tubes were directed toward the center of the mounting board as shown in figure 4. The tubes were passed through the board and then connected to flexible rubber hoses. This was done to provide flexibility when the loop orientation was changed.

In order to measure the circulating water temperature at the inlet and outlet of each heat exchanger, a thermocouple (24 gauge) was mounted on each tube approximately 13 cm (5") away from the end caps as shown in figure 4.

3.1.3 Dial Pressure Gage

In order to check for non-condensing gases in the loop and also check for leaks of the loop, a dial gage made by J/B Industries was mounted 7.6 cm (3") below the middle of the bottom cross connecting tube. This gage is a refrigeration pressure gage specially designed for use with R-21, R-12 and R-502. Its reading range was from 30 inches $^\text{Hg}$ (760 mm Hg) vacuum to 120 psig (8.4 Kg/cm$^2$) with an accuracy of $\pm1/2\%$ of full scale.

3.1.4 Auxiliary Vacuum Chamber

An Auxiliary Vacuum Chamber was installed as shown in figure 4. This provides a chamber outside of the
FIG. 7 THERMOCouple Seal plugs

A - Freon-air

B - Water-air

Silicon Sealant

9/8" Copper Tube

3/8" Compression fittings Assembly

813 Epoxy

Body

Cap Screw

Dimensions: 12" and 35/"
test loop for gases, carried into the loop with the working fluid when charging, to collect.

In addition it acted as a protective device for the vacuum pump because it was not necessary to leave the vacuum pump coupled to the system during charging and perhaps draw harmful Freon vapours into the pump.

3.1.5 Insulation

In order to reduce heat losses from the loop the outside of the evaporator and condenser assemblies were covered with 25.4 mm thickness of molded fiber glass insulation (1 1/2" I.D., 3 1/2" O.D.), thermal conductivity, 0.018 Btu/ft² Hr °F. All other tubes were wound first with 1.6 mm (1 1/16") layer of soft fiber glass insulation and then covered with a 25.4 mm (1") thickness of Armstrong Armaflex Pipe Insulation (12.7 mm (1/2") I.D., 38 mm (1 1/2") O.D., thermal conductivity; 0.253 Btu/ft² °F Hr at 90°F.

3.2 LOOP ORIENTATION MEASUREMENT DEVICES

The loop was mounted on a plywood sheet (4'x8'x1/2") and a support stand constructed which allowed angular rotation about two axes denoted as the rotations and inclination angles. A protractor was installed as shown in figure 8 with a plumb bob suspended from the radius center of the protractor by means of a ring and pin connector. The string indicates the loop orientation on the protractor. The accuracy of rotation of the inclination angle is ±1/2 degrees.
3.3 LEAKAGE TEST

3.3.1 Flow Loop

The loop was carefully tested for leaks over a pressure range from near zero to fifty psia. In the initial leakage test the loop leakage was small resulting in a pressure variation of only 0.2 psig per hour. This was so small that the leak location could not be detected by using a soap solution. The leak was found at the Neoprene rubber "O"-ring seat and 3/8" evaporator tube at the top end cap by partially submerging the loop in a water tank.

Each time the loop was to be charged a leakage check was carried out by sealing the loop under vacuum for approximately 15 minutes prior to charging it and observing the loop pressure gage.

3.3.2 Annulus

This was checked by circulating water through the annulus at a high flow rate (about 1000 lbs/hour). Leakage from the thermocouple lead wire hold "A" on the evaporator side was detected and eliminated by replacing the thermocouple wire holder "A" with the longer version "B" shown in figure 7.

3.4 HEAT SOURCE AND SINK

Distilled hot water was circulated through a calibrated Brook Rotameter Co. flowmeter from a LAUDA constant temperature bath that could be controlled accurately to ±0.01°C.

The rotameter was used to set the hot water to approximately 275 lb/hr for all of the test runs. The flow rate
was measured manually during every run by collecting a water sample over a timed period using a 500 ml beaker. A CENCO precision balance scale with a minimum scale division of 0.01 ounce, and a LAB-CHROH electric timer, capable of measuring elapsed time to the nearest one hundreth of a second, were used for these measurements.

Once set, the hot water flow rate measurement varied by no more than 1.1%.

A special constant temperature bath, shown in figure 9, was constructed using a soap stone sink and the components taken from a LAB-LIN constant temperature bath. The temperature variation was found to be less than ±0.03°C over our range of operating temperatures.

The cold water flow rate was also measured using the mass balance and timer, but with a one gallon container. The flow rate remained steady within 1.0% after steady state conditions were reached.

3.5 AUXILIARY CHARGING APPARATUS

One of the most difficult problems encountered was the design of a system to charge the loop with one component without introducing unwanted noncondensable gases of working fluid. Preliminary tests indicated that careful purging of gases from the loop and from the working fluid would be necessary to ensure proper operation.

To solve this problem an auxiliary vacuum chamber, a deaeration unit and a distillation apparatus were constructed.

The unit shown in figure 10 was used to evacuate the loop. The vacuum pump was protected from the working fluid
FIG. 9 CONSTANT TEMPERATURE BATH FOR SINK

FIG. 8 LOOP ORIENTATION MEASUREMENT DEVICES
Fig. 10 Schematic Diagram of Deaeration Unit
vapour by using two liquid Nitrogen vapour traps in series. A mercury-U-tube was used to check the vacuum drawn in the loop.

The distillation apparatus shown in figure 10 was constructed both to purify previously contaminated working fluids and to remove any air dissolved in the working fluid.

All of these safeguards were found to be necessary in order to eliminate gases from the loop when charging with a specified mass of working fluid.

3.6 DATA ACQUISITION SYSTEM

A convenient, fast, reliable on stream data analysis method was developed using a Hewlett Packard 2752A Teleprinter, a 2570A Coupler-Controller, a 2402A Integrating Digital Voltmeter and a 2100 Computer. The system is shown in Figure 11.

Two methods of on stream data analysis were used: an indirect analysis method and a direct analysis method.

3.6.1 Indirect Analysis Method

This method was used initially because of a malfunction in one of our interface cards which required considerable time to locate and repair. The coupler-controller pinboard was programmed to read and print one set of temperature readings every minute. This data was punched on paper tape, then read into the computer. The computer program shown in Appendix B was used to calculate the conductance "U" for each set of data, and a running average value of "U"
FIG. 11  SCHEMATIC DIAGRAM OF DISTILLATION UNIT
and its variance for each of the data sets for a given test. After a specified minimum number of sets of readings were taken the computer stopped accepting new data when two successive variance calculations differed by less than 3%.

A typical computer printout is shown in Appendix C.

3.6.2 Direct Analysis Method

The Coupler-Controller output was directly connected to the computer using serial interface cards and programmed as shown in Appendix D to accept any arbitrarily specified number of sets of data \( \leq 999 \), then calculate and print out the average value of "U" and its coefficient of variation of the temperatures. This process was repeated after 5-10 minutes and if no significant change was noted the test run was considered to be complete.

A sample printout is shown in Appendix E.

3.7 THERMOCOUPLE INSTRUMENTATION CALIBRATION

All thermocouples were made using 24 gage copper-constantan thermocouple wire taken from the same stock. The thermocouple junctions were made with a thermocouple welder. These junctions had an average diameter of 0.08 inches.

In order to check for consistency between the various thermocouple outputs, the loop was made isothermal by filling it with R-113 and supplying water for both the source and sink heat exchangers from a single constant temperature source set at a room temperature. It was found that the
first two switches in the scanner channel card yielded readings slightly but consistently higher (approximately 5 microvolts) than all the others regardless of which thermocouples were connected to it. It was decided to use these two switches for the mid point evaporator and condenser thermocouple so that their errors would tend to cancel out when calculating the mean temperature difference between the two surfaces.

Appendix F shows the final set of readings as well as the average of 9 sets of readings taken at 2 minute intervals during one of these tests. A comparison of the outputs for each run revealed that the variation in emf for each output was no more than ±1 μv about its mean value. The error in the average value of the heat transferred is of the order of a few watts. This error was considered to be negligible for most runs where the energy transferred was in the thousands of watts.

Similarly, the thermocouples were checked for consistency using the Direct Analysis Method. All working fluid was removed from the loop. The loop, containing air, was allowed to stand so that the thermocouples reached room temperature. As shown in Appendix G, each thermocouple was read five hundred times within a time span of five minutes, the average variation of the temperature from the mean of five hundred readings was no more than 0.02°C. This error in temperature measurement results in an error in the average value of the heat transferred.
is of the order of a few watts. This error was considered to be negligible for most runs where the energy transferred was in the thousands of watts.

Figure 13 shows the entire experimental test apparatus.
CHAPTER IV

EXPERIMENTAL PROCEDURE

4.1 TESTING PROCEDURE

Prior to conducting a test the following steps must be carried out to prepare the overall system:

a) Load the appropriate Fortran program into the computer (Appendix B for indirect data analysis program and Appendix D for direct analysis program).

b) Set the necessary coupler-controller to their required modes.

c) The thermocouple reference chamber must be warmed up for more than six hours prior to testing.

d) Warm up and calibrate the digital voltmeter.

e) Set the thermostats in the two constant temperature water baths to the temperature level desired for the tests and permit them to come to a steady operating state.

f) If the loop must be charged then the vapour trap dewar flasks of the evacuation assembly have to be filled with liquid nitrogen.

g) Distil approximately 400 ml of the chosen working fluid as follows:

(1) Circulate tap water in annulus of condensers of distillation unit.
(2) Fill flask A (see figure 10) with about 500 ml of the chosen working fluid.
(3) Gently boil the fluid in flask A.
(4) When a small quantity of condensate has collected in flask B, start heating the fluid slightly so that it is barely boiling. This is done so that air is not dissolved in the condensate during distillation and degasing.

h) Charge the working fluid in the loop as follows:

(1) After approximately 400 ml of condensate is collected in the flask B, connect its outlet tube to valve #3 at the bottom of the loop. (See figure 4)
(2) In order to purge all air from the inlet line and the valve the loop was rotated approximately 30° from the vertical, valves #1 to #4 were opened, and a small quantity of working fluid was allowed to flow through the line and valve 3 into the graduate flask in such a way that all gages were purged from the charging line and valve.
(3) Close valve #3 then drain all the working fluid from the flask through its bottom valve.
(4) Close the drain valve and valve #1.
(5) Record barometric pressure and room temperature.
(6) Connect the evacuation assembly suction line to valve #1, turn on the vacuum pump, then, after steady state is achieved gradually open valve #1.
(7) After approximately 30 minutes, when the system vacuum within accuracy of the gages, close valve #1 and then turn off the vacuum pump.
(8) Check the loop for leaks by waiting for about 15 minutes to see if the loop pressure changes.
(9) If no leaks are detected, close valve #4, then open valve #3.
(10) Allow the pressure in flask B in the charging unit to force the working fluid into the loop. Close valve #3 when liquid appears in the upper auxiliary vacuum chamber. In order to overcome the higher loop pressure with R-11 it was necessary to cool the loop as much as possible using cold tap water. Pressurize flask B in order to force the working fluid into the loop.
(11) Check for pressure of gases in the system by comparing the sum of the hydrostatic
and saturation pressures with the loop gage pressure.

(12) If pressures are checked, close valve #2. Loop now has a 100% charge. To set any lower charge open valve #4 slightly and bleed off the required amount of working fluid into its lower graduated flask. The % charge is given by

% charge = \frac{\text{loop volume} - \text{volume discharged}}{\text{loop volume}}

Close valve #4 and the loop is ready for testing.

i) Test procedure

(1) Rotate and incline the loop according to the desired orientation.

(2) Turn on the source and sink circulation pumps for both the evaporator and condenser and adjust their flow rate. Allow at least 20 minutes for the system to stabilize.

(3) Measure the source and sink water flow rates.

(4) Record system pressure, barometric pressure, room temperature, hot and cold water flow rate, rotation and inclination angles, percent charge, test number, run
number and date. Input the necessary data to the computer program.

(5a) When using the indirect data analysis method turn on the coupler-controller and its teleprinter, set to punch, when the test is to start press the start button on the coupler-controller. This will produce a data tape. Turn on the computer teleprinter, and data tape in its tape reader then press "run" on the computer panel.

(5b) When using the direct data analysis program press "run" on the computer panel, then input the requested data using the computer teletype according to the printed instructions on the teleprinter.

4.2 DETAILED DESCRIPTION OF PARAMETER VARIATION

4.2.1 Data Acquisition Tests

Appendix L defines the actual values of the parameters used in each test. It also identifies each test by test number. This provides for easy cross reference with the computer output data log books, and provides a synopsis of the results for each test.

4.2.2 Test Sequence I

This was performed to study the effect of the rotation angle and the percent charge on the loop conductance for two different sets of source and sink water temperatures ($T_{hw} = 54^\circ C\ (130^\circ F), T_{cw} = 25^\circ C\ (77^\circ F), T_{hw} = 54^\circ C\ (130^\circ F), T_{cw} = 32^\circ C\ (90^\circ F)$).
The loop was charged with the working fluid R-113 and the inclination angle set to zero degrees (vertical) with the source and sink temperature held constant. The rotation angle was varied from zero degrees (vertical) to near to 88° for charges which varied 90% to 30%.

4.2.3 Test Sequence II

In order to study the effect of the sink temperature on the loop conductance for various loop rotation angles, the charge (R-113) was held constant at 31% and tests carried out with $T_{\text{hw}} = 54^\circ\text{C}$ and $T_{\text{cw}} = 32^\circ\text{C}, 46^\circ\text{C}, 49^\circ\text{C}$ and $52^\circ\text{C}$, using R-113.

4.2.4 Test Sequence III

The study described in sequence I was continued except the source and sink temperatures were set at $T_{\text{hw}} = 54^\circ\text{C}$, $T_{\text{cw}} = 46^\circ\text{C}$ and the charge was varied from 27% to 9%.

4.2.5 Test Sequence IV

This test sequence was performed to study the effect of the mean operating temperature (system pressure) on the loop conductance for various angles of rotation for the two working fluids used. The % charge was held constant at 31% and the temperature difference, $(T_{\text{hw}} - T_{\text{cw}})$, for R-115 was held constant at 8.3°C (15°F) and for R-11 was held constant at 11.1°C (20°F). The loop conductance was then determined as a function of rotation angle for each different mean operating temperature level, $(T_{\text{hw}} + T_{\text{cw}})/2$. 
4.2.6 Test Sequence V

The purpose of this sequence of tests was to study the effect of inclination angle for a fixed charge and the loop at various angles of rotation of 30\(^\circ\) and \(T_{hw} = 44^\circ C\), \(T_{cw} = 36^\circ C\). For various inclination angles between 0\(^\circ\) and 80\(^\circ\) from the vertical plane, rotation angle was varied from 0\(^\circ\) to 88\(^\circ\) using R-113.

4.2.7 Test Sequence VI

To determine the effect of temperature difference (heat flux) on \(U\) for various rotation angles for a given mean operating temperature (system pressure). This was carried out by holding \((T_{hw} + T_{cw})/2\) constant for various \((T_{hw} - T_{cw})\) between 11\(^\circ\)C and 56\(^\circ\)C and varying the rotation angle with a 30\% charge of R-11.

4.2.8 Test Sequence VII

In order to determine the effect of the working fluid on the loop conductance test sequence I was repeated using R-11. In these tests the charge was varied between 84\% to 11\% and a set of source and sink water temperatures \(T_{hw} = 54^\circ C\), \(T_{cw} = 32^\circ C\) was used.

4.3 CALCULATION PROCEDURE

4.3.1 Loop Conductance

The loop conductance is determined by dividing the energy transport rate by the interface area between the working fluid and the inside wall of the evaporator (or the
condenser) and by the surface temperature difference between the evaporator and the condenser. Unfortunately, the energy transport from the evaporator to the condenser could not be calculated directly from the measured data. In addition the surface temperature difference between the evaporator and the condenser was subject to some error as a result of using only three thermocouples. The energy transport and the temperature difference were estimated experimentally as follows:

1) Calculation of energy transport.

The most realistic measure of the energy transport would be to take the average of the energy absorbed in the evaporator, $Q_e$, and the energy rejected in the condenser, $Q_C$. However, since these values could not be measured directly, an average energy transport was evaluated by using the mean value of the heat given up by the hot water flowing through the evaporator jacket, $Q_{ew}$, and the heat absorbed in the cold water flowing through the condenser jacket, $Q_{cw}$. The difference between this value of the loop conductance and the "ideal" value may be estimated by noting that:

$$Q_{ew} = Q_e + Q_{el}$$

where $Q_{el}$ is the heat loss from the evaporator to the surrounding air,

$$Q_{cw} = Q_c - Q_{cl}$$

where $Q_{cl}$ is the heat loss from the condenser. Thus

$$\frac{Q_{ew} + Q_{cw}}{2} = \frac{Q_e + Q_c}{2} + \frac{Q_{el} - Q_{cl}}{2}$$
The second term represents the amount by which the energy transport measured experimentally exceeds the ideal energy transport.

Since the source temperature is always higher than the sink temperature, the heat losses from the evaporator, $Q_{el}$, is always greater than that from the condenser, $Q_{cl}$.

In order to determine the magnitude of this error, loop heat loss tests were carried out with the loop empty of working fluid and supplying water at constant temperature to both source and sink heat exchangers. These tests were carried out for various temperature levels and the magnitude of the heat losses versus the temperature levels above ambient was plotted as shown in figure 14. The heat loss from either the evaporator or condenser jackets may now be estimated from figure 14 at the appropriate temperature. The magnitude of $Q_{el} - Q_{cl}$ was found to be approximately 5 watt when the source and the sink temperatures were 54°C, and 32°C respectively a 8 Btu/hr for 54°C and 46°C respectively. As a result the values of the energy transport used in the calculation of average loop conductance, $U_{avg}$ are high by 2.5 to 10% for R-113 for high and low performance and 2 to 8% for R-11 for high and low performance condition. This error results $1 \leq |U_{ideal} - U_{avg}| \leq 23 \text{ w/m}^2 \cdot ^\circ\text{K}$ for 40°F source and sink temperature difference and 0.5 $\leq |U_{ideal} - U_{avg}| \leq 12 \text{ w/m}^2 \cdot ^\circ\text{K}$ for 8°C difference.
ii) Calculation of temperature difference, \( (T_e - T_c) \)

The average temperature difference between the evaporator and the condenser, \( (T_e - T_c) \) should be calculated as

\[
T_e - T_c = \frac{1}{A_e} \int_0^{T_e} T_e \, dA_e - \frac{1}{A_c} \int_0^{T_c} T_c \, dA_c
\]

However, due to use of only three thermocouples to determine the average surface temperature of the evaporator and the condenser, the temperature difference was actually estimated as

\[
T_e - T_c = \frac{(T_{e1} + T_{e2} + T_{e3}) - (T_{c1} + T_{c2} + T_{c3})}{3}
\]

where \( T_{e1}, T_{e2}, T_{e3} \) and \( T_{c1}, T_{c2}, T_{c3} \) represent the temperatures obtained from the evaporator and the condenser thermocouples respectively.

The maximum error which arises in this approximation occurs when there is a shift between a wetted and a dryout condition in the evaporator or a shift between a film wetted and a flooded condition in the condenser. Such changes cause a jump in a thermocouple reading of \( \Delta T_e \) or \( \Delta T_c \) (the change in temperature reading of the particular thermocouple, which contributes an error of \( \Delta T_e/3 \) or \( \Delta T_c/3 \) in the calculation of the temperature difference, \( (T_e - T_c) \). The maximum relative magnitude of \( \Delta T_e/3 \) to \( (T_e - T_c) \) and \( \Delta T_e/3 \) to \( (T_e - T_c) \) for a 22°C
source to sink temperature difference were found to be approximately ±7% and ±9% respectively. Thus the corresponding possible error in the loop conductance, due to dryout in the evaporator or flooding in the condenser, would be approximately ±7% and ±9% respectively. The calculations are shown in Appendix I.

4.3.2 Percentage Charge

Due to the installation of valves, a pressure gage, and the charge measurement equipment, the experimental test rig volume was larger than that of the loop proper.

The % charge of the loop proper was determined as follows:

\[
\text{% charge} = \frac{V_T - V_W - V_B}{V_L} \quad \ldots 4-2
\]

where \(V_T = V_{\text{top}} + V_L + V_B\)

\(V_{\text{top}}, V_L\) and \(V_B\) are the volumes of the auxiliary tubing at the top of the loop, the main loop, and of the extra tubing etc. at the bottom of the loop respectively. \(V_W\) is the volume of the working fluid withdrawn from the loop.

The volumes of \(V_{\text{top}}, V_L\) and \(V_B\) were estimated by individual calculations. The total volume obtained by adding \(V_{\text{top}}, V_L, V_B\) agreed well with the volume which was measured experimentally.
CHAPTER V

RESULTS AND DISCUSSION

5.1 EFFECT OF CHARGE AND LOOP ORIENTATION

5.1.1 Effect of % Charge and Rotation Angle

Figures 15 to 16 for R-113, and figure 17 for R-11 show the effect on the loop conductance of % charge and the loop rotation angle for zero inclination with a constant temperature difference.

It may be noted that for both fluids the optimum loop conductance occurs with approximately a 20% charge with the loop between 60° and 70° from the vertical.

Figures 18 and 19 are cross plots of the smoothed data curves shown in figure 15, 16 and 17.

A careful study of the experimental data shows that the rapid decrease of loop conductance found to the left side of the optimum loop conductance in figures 18 and 19 is caused by dryout occurring in the evaporator. The dryout area increases rapidly with decreasing % charge. The decrease to the right side of the optimum loop conductance coincides with an increasing quantity of liquid being pumped from the evaporator to the condenser. The liquid carryover from the evaporator to the condenser increases both the transport resistance in the cross connecting tube and the condenser resistance. The former is increased because of extra pumping losses associated with the two phase flow and the latter because of the increased film in the condenser.

When the loop is rotated the vertical, working fluid flows into the evaporator from the condenser. This
Fig. 15 Loop conductance vs rotation angle for R-113

\[ T_{HN} = 130^\circ F \ (54.4^\circ C), \quad T_{CW} = 90^\circ F \ (32.2^\circ C) \]
Fig. 16 Loop conductance vs rotation angle for R-113

$T_{HN} = 136^\circ F \ (54.4^\circ C), \ T_{CW} = 115^\circ F \ (46^\circ C)$
Fig. 17: Loop conductance vs rotation angle for R-11

$T_{HW} = 130^\circ (54.4^\circ C), \ T_{CW} = 90^\circ F (32.2^\circ C)$.
Fig. 18 LOOP CONDUCTANCE VS Charge for R-113

(A) \( T_{HN} - T_{CW} = 8.3^\circ C \) (18\(^o\)F)

(B) \( T_{HN} - T_{CW} = 22.2^\circ C \) (40\(^o\)F)
decreases any flooded area in the condenser and any dryout area in the evaporator, thus causing the loop conductance to increase. For the smaller charges, less than 50%, this trend ceased, then reversed when the condenser was no longer flooded and liquid carryover commenced.

In figures 15 to 16, the use of only three thermocouples to determine the average temperature of the evaporator and the condenser caused some of the experimental points on the graphs to deviate from a smooth curve. This was most apparent when a change between boiling and dryout occurs over a thermocouple in the evaporator. The resulting abrupt change in a temperature reading could cause the loop conductance to jump by as much as 30%.

5.1.2. Inclination Angle of the Loop

Figure (19) shows the effect of the inclination angle and the rotation angle on the loop conductance for a 30% charge of R-113 with a constant temperature difference.

This figure reveals that the loop conductance is not significantly affected by inclination angle except for the case of zero rotation angle.

For the zero rotation angle case it was observed experimentally that, between 0° and 25° angle of inclination, the heat flux and the temperature difference between the evaporator and the condenser, did not change significantly. For both of these tests the upper two thermocouples in the evaporator were in a dryout region. However, from 25° to 45° angle of loop inclination from the vertical plane
Fig. 19 Effect of Inclination Angle for R-113

$T_{HN} = 44.4^\circ C (112^\circ F)$  $T_{CH} = 35.5^\circ C (96^\circ F)$
the heat flux increased by approximately 21% and the middle thermocouple became wetted, thus causing a decrease in the temperature difference of approximately 14%. From 45° to 80°, angle of inclination, the heat flux and the temperatures did not change significantly.

The order in which these tests were carried out was 0°, 45°, 80° and then 25° angle of inclination.

Neither the lack of any significant change of the loop conductance for the zero rotation angle between 0° and 25° angle of inclination and between 45° and 80° angle of inclination, nor the very abrupt change between 25° and 45° angle of inclination can be satisfactorily explained. A steady increase in the heat flow rate should be expected.

5.2 COMPARISON OF R-113 and R-11

Figure 20 shows a comparison of the values of loop conductance for R-113 with those for R-11 for a 30% charge at various angles of rotation. The performance for R-11 is significantly better than that for R-113. As may be noted, the difference generally increases with the loop rotation angle.

In order to better understand the reason for the difference, the effect of property changes upon the evaporator resistance, the condenser resistance and the transport resistance will be investigated individually.

For the near vertical position of the loop, where dryout occurs in the evaporator, the transport resistance is
FIG. 20 COMPARISON OF R-113 AND R-11

$T_{in} = 130^\circ (54.4^\circ C)$; $T_{cw} = 90^\circ F (32.2^\circ C)$
negligible. For this case the effect of property changes on the heat transfer coefficients can be estimated using the nucleate pool boiling equation (1-7) and the condensation equation (1-8) evaluating the properties at mean operating temperature levels used in the experiments.

The boiling and film condensation heat transfer coefficients were approximately 13% and 15% respectively higher for R-11 than with R-113 for the range of operating temperatures used. Thus the improvement of the loop performance should be between 13 and 15%. This agrees well with the 15% increase measured experimentally.

When liquid carries over from the evaporator to the condenser, the transport resistance has a significant effect on the loop performance, which can be explained using the transport resistance equation (1-9e).

\[ R_t = \frac{K(\Delta T_{sat})^{1-n}}{T_s} \cdot \left( \frac{V_{fg}}{S_{fg}} \right)^{n+1} \]

The ratio, \((\frac{V_{fg}}{S_{fg}})^{n+1}\), for R-11 is approximately 35% less than that for R-113 for the range of operating temperatures used. Thus the transport resistance for R-11 should improve by a similar amount, assuming that \(K\) does not vary significantly.

Thus the improvement in the loop conductance for R-11 with rotation angle may be explained by the fact that the increase in the transport resistant terms is twice as great as the increase in the evaporator and condenser resistances.
5.3 EFFECT OF PRESSURE

Figures (21, 22) show the effect of the system pressure (mean operating temperature level) on the dimensionless loop conductance \((U'(p, \theta_R)/U (p = 32 \text{ psia}, \theta_R))\) for R-113 and R-11 for a 30\% charge at various angles of rotation and constant temperature difference. These graphs show that the loop conductance increases slightly with an increase in the system pressure.

This variation may be explained by considering the effect of the property changes, brought about by the change in the operating pressure (temperature) level, on the evaporator, condenser, and the transport resistances. According to the convection and the nucleate pool boiling equations, an increase in pressure from 15.7 to 43 psia would cause the forced convection heat transfer coefficient and the boiling heat transfer coefficient to increase. The convection and the boiling heat transfer coefficients increase by approximately 2\% and 135\% respectively. This will cause a decrease in the evaporator resistance which should fall between those limits. The actual value will be dependent upon which heat transfer mode is dominant for the given conditions.

The condenser heat transfer coefficient and hence resistance was found to be insensitive to the system pressure over the pressure range studied.

The effect on the transport resistance due to property changes may be estimated by considering their effect on the ratio, \(\frac{V_{fg}}{S_{fg}}^n + 1\) in the transport equation (1-9e) while con-
CHARGE = 30 \% \quad T_{HN} - T_{CW} = 8.9^\circ C

\textbf{I} \quad \text{INDICATES RANGE OF DATA POINTS FOR}
\theta_{\rho P} = 0, 40, 50, 60, 70, \text{ and } 80 \text{ degrees}

\textbf{SYSTEM PRESSURE}

\textbf{FIG. 21 EFFECT OF SYSTEM PRESSURE FOR R-113}
Fig. 22 Effect of Loop System Pressure for R-11
sidering the flow resistance term, $\tau$, to be essentially constant. The transport resistance for R-11 was found to decrease by approximately 78% when the pressure increased from 15.7 to 43 psia.

Since the experimental performance of the loop increases by only 10% for a pressure rise from 15.7 to 43 psia, it would appear that the forced convection must dominate in the evaporator and the sum of the evaporator and condenser resistances outweigh the transport resistance by approximately seven to one at 15.7 psia and by approximately thirty-five to one at 43 psia.

5.4 EFFECT OF TEMPERATURE DIFFERENCE

Figure 23 shows the effect of the temperature difference on the dimensionless loop conductance $U((T_e - T_c, \theta_R)/U(T_e - T_c) = 19.7^\circ C, \theta_R))$ for various angles of rotation for R-11 when operating at a constant mean temperature. The graph shows that over the five fold increase in temperature difference over the studied from 4 to 19.7$^\circ$C, the temperature difference does not have a significant affect on the loop conductance, except for the vertical position of the loop.

Some insight into this behaviour may be gained by considering the effect of a change in the temperature difference on the transport, the evaporator and the condenser resistance.

The transport resistance varies as $(T_{ef} - T_c)^{1-n}$ where "n", defined by flow varies from 1 for laminar flows to 0.5 for fully turbulent flows. For this calculation it was assumed that
\[(T_{ef} - T_{C}) \sim (T_e - T_C)\]. The vapour flow Reynolds number in the
top cross-connecting tube may be estimated by using experi-
mental data to calculate the flow rate, \(\dot{m} = \frac{Q}{x H_{fg}}\). It
would appear that the vapour flow Reynolds number would fall
between approximately 10,000 and 50,000. For this range the
power, \((1-n)\) would be approximately 0.43. Thus a five fold
increase in the fluid saturation temperature difference be-
tween the evaporator and the condenser would cause the trans-
port resistance to approximately double.

According to the film condensation equation, the
condensation resistance varies as \((T_{cf} - T_C)^{1/4}\). Since working fluid
temperature in condenser, \(T_{cf}\) was not measured experimentally,
the temperature difference was assumed to be proportional to
\((T_{top} - T_C)\). Based on this assumption, the condenser resistance
increased approximately 69% when \((T_e - T_C)\) increased from 4 C
to 19.7 C.

The effect on the evaporator resistance may be esti-
mated by considering both the convection equation (1-6) and
the nucleate pool boiling equation. The forced convection
heat transfer coefficient is a function of Reynolds number
which is a function of the mass flow rate. The mass flow rate
is essentially linearly proportional to the heat flow rate
which in turn was found experimentally to be approximately
linearly proportional to the temperature difference, \((T_e - T_C)\).
Thus the forced convection heat transfer coefficient varies
approximately as \((T_e - T_C)^{0.8}\). The boiling heat transfer co-
efficient, \(h_b\) varies as \((T_{ef} - T_{ef})\). Since \(T_{ef}\) was not measured
experimentally it was assumed that \((T_e - T_{ef})\) or \((T_e - T_{top})\) and hence, the boiling heat transfer coefficient, \(h_b\), varies as \((T_e - T_{top})\). Based on these assumptions it was found that the forced convection and boiling heat transfer coefficients both increased by approximately 270% when the temperature difference, \((T_e - T_c)\), increased from 4°C to 19.7°C. The evaporator resistance is affected not only by the heat transfer coefficient but also by any change which may occur in the dryout area. For the vertical orientation of the loop the lower temperature differences caused less vigorous boiling which in turn resulted in dryout occurring in the evaporator. Thus any change in temperature difference when the loop is operating in a dryout condition will substantially affect the evaporator resistance.

Since the loop conductance remained essentially constant when the temperature difference, \((T_e - T_c)\), increased from 4°C to 19.7°C, it would appear that the decrease in the evaporator resistance balanced the increase in the condenser and transport resistance except for the vertical case where the added dryout effect was noticeable.

5.5 EFFECT OF CHANGING THE SINK TEMPERATURE

Figure 24 shows the effect on the loop conductance of varying the sink temperature for various angles of rotation, with a 30% change of R-113.

For small temperature differences, \(T_{hw} - T_{cw} < 8.3°C\), the loop conductance decreases as the sink temperature decreases. This may be caused by the effect of dryout in the
Fig. 24: Effect of changing sink temperature for R-113

\[ T_{HN} = 54.4^\circ C \ (130^\circ F) \]
evaporator due to less vigorous boiling. For large temperature differences, $T_{hw} - T_{cw} > 8.3^\circ C$, the loop conductance is insensitive to the sink temperature. This trend may be caused by the combination of the following two opposing effects. An increase of the sink temperature, which causes an increase in the system pressure (mean operating temperature level), decreases the evaporator and the transport resistances, but a decrease of the sink temperature, which causes an increase in the temperature difference, increases the transport and the condenser resistances.

5.6 SIMULATION OF THE EXPERIMENTAL LOOP

This section describes the specific model used to simulate the experimental loop.

The experimental loop was simulated using the model developed in Chapter II with additional input for the evaporator and the condenser heat transfer coefficients, and the reciprocal of the average liquid fraction for the wetted length. The equations used for this simulation as well as their development are given in Appendix J and H.

Figure 25 and 26 show a comparison of the simulated and the experimental loop conductance as a function of % change and rotation for R-113 and R-11 respectively.

The study of the model simulation shown by the graphs indicate that for low charges where dryout occurs in the evaporator the agreement is very good. However for larger charges the loop conductance predicted is higher than the experimental value due to the model neglect of the transport resistance.
FIG. 26 COMPARISON OF THE SIMULATED AND EXPERIMENTAL U FOR R-11
The decreases in the simulation conductance with increasing charge is due to liquid flooding in the condenser. This effect can also be seen in the experimental data points on the right side of the optimum loop conductance.

It should be noted that for angles of rotation about 80°, the agreement between the simulation and experimental results is very poor because of the sudden departure of the experimental results from their usual slope.

5.7 RELIABILITY OF THE EXPERIMENTAL RESULTS

5.7.1 Reproducibility

Figure 27 illustrates the reproducibility of the experimental results through a comparison of two test runs made four months apart. The second test was made with a fresh charge of R-11. The maximum difference between the two sets of results is within the limits of the coefficient of the variation of each test.

Since all the data were analysed using the on-stream method, misreading of data due to human error was eliminated.

The coefficient of variation of the average loop conductance for each test was usually less than 5%. The maximum coefficient of variation of any of the data used was 10%.

5.7.2 Probable error in calculated results

A detailed probable error analysis is available in Appendix I.

Errors arise due to three factors:

1) inaccuracy in an experimental reading

2) the assumption that the mean surface temperatures of the
Rotation angle from the vertical, degrees

Fig. 27: Reproducibility

Charge = 30%

JAN. 10/75

MAY 11/75

U, W/m² K

U, BTU/hr ft² °F
evaporator and of the condenser can be obtained by averaging the output of three thermocouples,

3) the value calculated for loop conductance does not precisely represent the correct loop conductance due to heat losses.

i) Error due to inaccuracies in the experimental measurements

This error occurred due to thermocouple reading errors caused by the equipment used, inaccuracy in the measurement of water flow rates, the manufacturing tolerance of the diameter, and the length of the evaporator and the condenser tubes. The probable error in the loop conductance due to these errors was found to be \( \pm 4.8\% \).

ii) Approximation of mean surface temperatures using three thermocouples.

This error, already discussed in Chapter IV, may become significant if dryout occurs in the evaporator or if flooding occurs in the condenser. For R-11, for a 22.2°C and a 8.3°C source and sink temperature difference the maximum possible error is approximately \( \pm 7 \) and \( \pm 9\% \) respectively.

iii) Error due to the heat losses

The reason this error occurred, is mentioned in detail in Chapter IV. This error is caused by the heat losses through the insulation of the evaporator and the condenser jackets. The values of the loop conductance, \( U \) calculated by the program and plotted on the various diagrams are high by 2.5 to 10% for R-113 for the high and low loop
conductance respectively and 2 to 8% high for R-11 for high and low performance. This results in the reported loop conductance being high by from 16 to 23 w/m² K for a source and sink temperature difference of 22.2°C and high by 9 to 11 w/m² K for a 8.3°C difference.

iv. Charge

The probable error in the percent charge was calculated as:

For charge = 90%: \(+ 0.4 \) (-+ 0.43%)
charge = 73%: \(+ 0.4 \) (-+ 0.55%)
charge = 63%: \(+ 0.43 \) (-+ 0.58%)
charge = 52%: \(+ 0.46 \) (-+ 0.88%)
charge = 41%: \(+ 0.492 \) (-+ 1.2%)
charge = 30%: \(+ 0.54 \) (-+ 1.8%)
charge = 20%: \(+ 0.58 \) (-+ 2.9%)

v. Error in loop orientation

The loop rotation angle and the inclination angles were estimated to be \( \pm 0.5 \) degree; that is plus or minus one half of the smallest division.
CHAPTER VI
CONCLUSION

As a result of the experimental study carried out, it is apparent that the optimum loop conductance is achieved when dryout occurs at the top of the evaporator and when there is no flooding in the condenser. The loop can be improved by using a working fluid which has a higher thermal conductivity, higher latent heat of vaporization, smaller ratio of specific volume of vaporization to the entropy of vaporization, \( \frac{V_{fg}}{S_{fg}} \), and lower viscosity. Increased system pressure also results in better performance.

The maximum loop conductance achieved (operating with mean operating temperatures of 39°C for R-113 and 43°C for R-11, and with a source-sink temperature difference of 14°C for R-113 and 25°C for R-11) was approximately 850 W/m²·K for R-113 and 1150 W/m²·K for R-11. This occurred with a 20% charge and a 60° angle of loop rotation.

On the basis of the price comparison made in Appendix K between a two phase thermosiphon loop with a direct gas-to-gas heat exchanger, it was found that the thermosiphon loop would cost about one half as much as the latter. In addition the thermosiphon loop has the advantage of flexibility.

For future experimental investigations, the following recommendations are suggested:

1) The number of thermocouples mounted on the surfaces of the evaporator and condenser should be increased. This will decrease the average
surface temperature error which occurs when there is dryout in the evaporator or flooding in the condenser.

2) A smaller annulus cross-sectional area in the evaporator and condenser heat exchangers would yield better outside heat transfer coefficients and minimize possible natural convection effects for low water flow rates.

3) Temperature and pressure measurements should be made at the top of the evaporator and condenser. This will provide significant information on both the state of the working fluid and pressure drop in the interconnecting vapour tube.
REFERENCES


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6. McDonald, T.W. Personal communication


17. Diciccio, R. Two-phase flow Visualization Thermosiphon Loop, University of Windsor, Mechanical Engineering Report, H.T. 1-75 (1975)

APPENDIX A

LIST OF EQUIPMENT AND INSTRUMENTATION

1. Armstrong Armflex Pipe Insulation, 1/2" I.D., 1/2" wall thickness, thermal conductivity: 0.258 BTU/ft²-°F at 90°F. Armstrong Cork Co., Lancaster, Pennsylvania, U.S.A.

2. Computer, Model 2100A, Memory storage to 8192 words (8K) with 980 nanosecond memory cycle time. Program languages: HP Fortran, HP Fortran IV, HP Algol and HP Basic. Hewlett Packard, U.S.A.

3. Coupler-Controller, Model 2570A, consists of two scanner cards, BCD Voltmeter, Serial Interface Card, Teleprinter Card, Clock timer control card. A scanner card is up to 10 data-channels. Hewlett Packard, U.S.A.

4. Dial Pressure Gage, 120 psig to 30 inches Hg vacuum, for R-12, R-22 and R-502. J/B Industries, Aurora, Ill., U.S.A.

5. Epoxilite 813, Temperature Setting Epoxy Bond Integrity up to 600°F, William T. Bean Inc., Detroit, Mich., U.S.A.

6. Epoxy, Use at 75°F. Lepages, Canada.

7. Freon-113, see table , Du Pont of Canada.

8. Freon-11, " "

9. Flowmeter, Rotameter, type 841110. 0 to 250 graduated with an increment 10, Hatfield, Penn., U.S.A.
10. **Integrating Digital Voltmeter**, Model 2402A, 0.001% reading accuracy after 60 minute warm up. 43 measurements per second (triggered externally). One measurement every 10 seconds to 10 measurements per second. Hewlett Packard, U.S.A.


13. **Lab-Chron Timer**, Model 1401.100th second. Division Lab-Line Instruments Inc., Chicago, Ill., U.S.A.

14. **Molded Fiber Glass Insulation**, 1 1/2" I.D., 3 1/3" O.D., thermal conductivity; 0.018 BTU/ft²°F at 100°F, Fiber Glass Insulation Co., Canada.


16. **Silicone Seal**, adhesive; Canadian General Electric, Canada.

17. **Prex Glass Funnel**, 0 to 250 graduated with an increment 2 ml.

18. **Teleprinter**, Model 2752A. Typing speed; 100 words per minute, punching and reading speed; 10 words per second. Hewlett Packard, U.S.A.

19. **U-tube Manometer**, 36" the Meriam Instrument Co., Cleveland, Ohio, U.S.A.

PROGRAM THERM

MADE BY K. S. HUANG

TAPE NO. 18, OCTOBER 2, 1974

DATA ANALYSIS FOR THE THERMOSIPHON LOOP

DIMENSION DATA(23), TEMP(23), TAVG(23), TSUM(23)

500 VARU=0.0
    HFSUM=0.0
    CFSUM=0.0
    ZN=1.0
    CP=1.0
    DO 301 I=1,23
    TSUM(I)=0.0
    TAVG(I)=0.0
    301 CONTINUE
    WRITE(2,4)
    4 FORMAT("INPUT USING THE TAPE PUNCH OR THE KEYPAD")
    READ(1,5) CHARG, ROTAT, CFL, HFL, DAV, TEST, DIFF, SKIP, SJUMP, ANGL
    5 FORMAT(F4.1, IX, 12, 1X, F5.1, IX, F5.1, IX, 12, 1X, F6.4, 1X, F2.0, 1X,
    #F3.0, IX, 12)

600 WRITE(2,7)
    7 FORMAT(4/I)
    WRITE(2,8) TEST, ZN
    8 FORMAT(38X,"TEST NO ",13.3X,"RUN NO ",12)
    WRITE(2,9)
    9 FORMAT(4/I)
    WRITE(2,10)
    10 FORMAT(" DATA INPUT(VOLT)"
    READ(1,*) (DATA(I), I=1,7)
    IF (ABS(6.-DATA(I)) .GT. 3) GOTO 300

300 READ(1,*) (DATA(I), I=8,11)
    READ(1,*) (DATA(I), I=12,15)
    READ(1,*) (DATA(I), I=16,19)
    READ(1,*) (DATA(I), I=20,23)
    DO 200 I=1,23
    TEMP(I)=32.081111971+46638.749761*DATA(I)-1321291.435*
    #DATA(I)**2+59109465.*DATA(I)**3
    200 CONTINUE
    THAVE=(TEMP(1)+TEMP(2)+TEMP(19))/3.
    TCAVE=(TEMP(21)+TEMP(4)+TEMP(23))/3.
    TDIF=ABS(TEMP(9)-TEMP(11))
    TCDIF=ABS(TEMP(13)-TEMP(15))
    QIN=HFL*CP*TDIF
    QOUT=CFL*CP*TCDIF
    TMDIF=ABS(THAVE-TCAVE)
    FLUXA=(QIN+QOUT)*6.)/(8.311*3.141593*4.)
    FLUX=(QOUT*12.)/(8.311*3.141593*4.)
    UAVE=FLUXA/TMDIF
    U=FLUX/TMDIF
    DO 302 I=1,23
    TSUM(I)=TSUM(I)+TEMP(I)
    TAVG(I)=TSUM(I)/ZN
    302 CONTINUE
HFSUM=HFSUM+HFLOW
CFSUM=CFSUM+CFLOW
HFAVG=HFSUM/ZN
CFAVG=CFSUM/ZN
THAVG=(TAVG(17)+TAVG(2)+TAVG(19))/3.
TCAVG=(TAVG(21)+TAVG(4)+TAVG(23))/3.
HDIFA=ABS(TAVG(9)-TAVG(11))
CDIFA=ABS(TAVG(13)-TAVG(15))
GIAVG=HFAVG*CP*HDIFA
QOAVG=CFAVG*CP*CDIFA
TDIFA=ABS(THAVG-TCAVG)
FAVG=((GIAVG+QOAVG)+6.)/(6.311*3.141593*4.)
FAVG=(QOAVG*12.)/(6.311*3.141593*4.)
UAVG=FAVG/TDIFA
UAVG=FAVG/TDIFA
VARUN=(VARUN*(ZN-1.))+(U-UAVG)**2)/ZN
STDUN=SRT(VARUN)
WRITE(2,101)
101 FORMAT(3/
 IF(3.-SKIP) 900,706
706 IF(ZN-SJUMP) 910,266
266 IF(ABS(VARUN-VARU)-(DIFF*VARU)) 900,900,910
910 WRITE(2,111)
111 FORMAT(16X,"QUANTITY",19X,"VALUE",5X,"AVGE")
GO TO 999
900 WRITE(2,12)
12 FORMAT(16X,"DATA ANALYSIS FOR THE THERMOSIPHON LOOP")
WRITE(2,13)
13 FORMAT(16X,38"*/")
WRITE(2,14) DAY
14 FORMAT("DATE",21X,"OCT.",1X,12,"","74")
WRITE(2,15) DATA(5),DATA(6),DATA(7)
15 FORMAT("TIME",20X,12,":",12,":",12).
WRITE(2,16)
16 FORMAT("WORKING FLUID","1X,"RT13")
WRITE(2,17)
17 FORMAT("LOOP VOLUME","13X,"219 MILLI LITERS")
WRITE(2,18) CHARG
18 FORMAT("Z CHARGE( BY VOLUME)","4X,12,1X,"Z")
WRITE(2,19) ROTAT
19 FORMAT("LOOP ROTATION ANGLE","5X,12,1X,"DEGREES")
WRITE(2,20) ANGLI
20 FORMAT("INCLINATION-LOOP PLANE","2X,12,1X,"DEGREES")
WRITE(2,20)
25 FORMAT("RESULTS")
WRITE(2,51)
51 FORMAT("---------")
WRITE(2,52)
52 FORMAT(16X,"QUANTITY",19X,"VALUE",6X,"AVGE")
WRITE(2,53) HFLOW,HFAVG
53 FORMAT("FLOW RATE OF HOT WATER(LBS/HF)","11X,F6.1,4X,F6.1)
WRITE(2,54) CFLOW,CFAVG
FORMAT("FLOW RATE OF COLD WATER (LBS/HR)\\n", 1FX, F6.1, 4X, F6.1)
WRITE(2, 55) TEMP(9), TAVG(9)

FORMAT("TEMPERATURE OF HOT WATER IN (F)\\n", 11X, F6.1, 4X, F6.1)
WRITE(2, 56) TEMP(11), TAVG(11)

FORMAT("TEMPERATURE OF HOT WATER OUT (F)\\n", 1FX, F6.1, 4X, F6.1)
WRITE(2, 57) TEMP(13), TAVG(13)

FORMAT("TEMPERATURE OF COLD WATER IN (F)\\n", 16X, F6.1, 4X, F6.1)
WRITE(2, 58) TEMP(15), TAVG(15)

FORMAT("TEMPERATURE OF COLD WATER OUT (F)\\n", 9X, F6.1, 4X, F6.1)
WRITE(2, 59) TEMP(17), TAVG(17)

FORMAT("TEMP OF HEATER SURFACE-TOP (F)\\n", 12X, F6.1, 4X, F6.1)
WRITE(2, 60) TEMP(2), TAVG(2)

FORMAT("TEMP OF HEATER SURFACE-MIDDLE (F)\\n", 9X, F6.1, 4X, F6.1)
WRITE(2, 61) TEMP(19), TAVG(19)

FORMAT("TEMP OF HEATER SURFACE-BOTTOM (F)\\n", 9X, F6.1, 4X, F6.1)
WRITE(2, 62) TEMP(21), TAVG(21)

FORMAT("TEMP OF CONDENSER SURFACE-TOP (F)\\n", 9X, F6.1, 4X, F6.1)
WRITE(2, 63) TEMP(4), TAVG(4)

FORMAT("TEMP OF CONDENSER SURFACE-MIDDLE (F)\\n", 6X, F6.1, 4X, F6.1)
WRITE(2, 64) TEMP(23), TAVG(23)

FORMAT("TEMP OF CONDENSER SURFACE-BOTTOM (F)\\n", 6X, F6.1, 4X, F6.1)
WRITE(2, 65) TDIF, HDIF

FORMAT("TEMP DIFF BETWEEN HOT WATER IN & OUT (F)\\n", 2X, F6.1, 4X, F6.1)
WRITE(2, 66) TCDIF, CDIF

FORMAT("TEMP DIFF BETWEEN COLD WATER IN & OUT (F)\\n", 1X, F6.1, 4X, F6.1)
WRITE(2, 67) TMDIF, TDIF

FORMAT("SURFACE TEMP DIFF EVAP.-COND. (F)\\n", 9X, F6.1, 4X, F6.1)
WRITE(2, 68) FLUXA, FAUVG

FORMAT("HEAT FLUX BASED ON AVG. (BTU/HR FTSC)\\n", 3X, F6.1, 4X, F6.1)
WRITE(2, 69) FLUX, FAUVG

FORMAT("HEAT FLUX (BTU/HR FTSC)\\n", 19X, F6.1, 4X, F6.1)
WRITE(2, 70) QIN, QIAVG

FORMAT("ENERGY IN (BTU/HR)\\n", 24X, F6.1, 4X, F6.1)
WRITE(2, 71) QOUT, GOAVG

FORMAT("ENERGY OUT (BTU/HR)\\n", 23X, F6.1, 4X, F6.1)
WRITE(2, 72) UAVE, UAAVG

FORMAT("U BASED ON AVG. C(BTU/HR FTSC)\\n", 9X, F6.1, 4X, F6.1)
WRITE(2, 73) U, UAUVG

FORMAT("OVERALL HEAT TRANS. COEFF. (BTU/HR FTSC)\\n", 1X, F6.1, 4X, F6.1)
WRITE(2, 74)

FORMAT("VAR. OF OVERALL HEAT TRANS. COEFF.\\n", 4X, F9.4)
WRITE(2, 76) STDNU

FORMAT("STANDARD DEVIATION OF OVERALL COEFF.\\n", 4X, F9.4)
WRITE(2, 77)

FORMAT(8/)

IF(ABS(VARUN-VARU)-(DIFF*VARU)) 261, 261, 263

IF(ZN-SJUMP) 263, 767

WRITE(2, 262)

FORMAT(10/)
WRITE(2,320)
320 FORMAT(62HPAUSE WITH 66(OCTAL) IN THE A-REGISTER. TO EXECUTE THE PR
ограм, \43HLOAD DATA TAPE FOR NEXT TEST AND PRESS RUN.)
PAUSE 66
GO TO 500
263 VARU=VARUN
ZN=ZN+1.
GO TO 600
211 WRITE(2,321)
321 FORMAT(62HPAUSE WITH 77(OCTAL) IN THE A-REGISTER. TO EXECUTE THE PR
ограм, \48HLOAD NEXT DATA TAPE FOR THIS TEST AND PRESS RUN.)
WRITE(2,322)
322 FORMAT(///)
PAUSE 77
GO TO 600
STOP
END
ENDS
DATA INPUT(VOLT)

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<td>1</td>
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<td>01:17:00</td>
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<td>4</td>
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<td>01:17:14</td>
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<td>5</td>
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<td>6</td>
<td>0.1913E-6</td>
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<td>7</td>
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<tr>
<td>8</td>
<td>0.1592E-6</td>
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DATA ANALYSIS FOR THE THERMOSIPHON LOOP

*******************************

DATE: SEPTEMBER 2, 1974
TIME: 1:17:0
WORKING FLUID: R113
LOOP VOLUME: 201 MILLI LITERS
% CHARGE BY VOLUME: 30%
LOOP ROTATION ANGLE: 50 DEGREES
INCLINATION LOOP PLANE: 0 DEGREES

RESULTS

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Value</th>
<th>Ave</th>
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<tr>
<td>FLOW RATE OF HOT WATER(LBS/HR);</td>
<td>275.6</td>
<td>275.6</td>
</tr>
<tr>
<td>FLOW RATE OF COLD WATER(LBS/HR);</td>
<td>264.2</td>
<td>264.2</td>
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<td>TEMPERATURE OF HOT WATER IN(F);</td>
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<td>TEMPERATURE OF HOT WATER OUT(F);</td>
<td>127.8</td>
<td>127.7</td>
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<td>TEMPERATURE OF COLD WATER IN(F);</td>
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<td>89.9</td>
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<tr>
<td>TEMPERATURE OF COLD WATER OUT(F);</td>
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<td>119.0</td>
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<tr>
<td>TEMP OF HEATER SURFACE-BOTTOM(F);</td>
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<td>123.5</td>
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<td>TEMP OF CONDENSER SURFACE-TOP(F);</td>
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<td>103.3</td>
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<td>SURFACE TEMP DIFF EVAP.-COND.(F);</td>
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<td>17.6</td>
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<td>HEAT FLUX(BTU/HR FTSC);</td>
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<td>ENERGY IN(BTU/HR);</td>
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<td>ENERGY OUT(BTU/HR);</td>
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<td>U BASED ON Ave @C(BTU/HR FTSC F);</td>
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<td>127.6</td>
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<tr>
<td>OVERALL HEAT TRANS. COEFF(BTU/HR FTSC F);</td>
<td>124.6</td>
<td>123.2</td>
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</table>

VARIANCE OF OVERALL HEAT TRANSFER COEFF : 3.2624
STANDARD DEVIATION OF OVERALL COEFF : 1.8662
APPENDIX D

FTN.B.L

PROGRAM THERM

C MADE BY K. S. HVANG

C THERMOSIPHON LOOP, PROGRAM NO. 2, TAPE NO. 2F, NOV. 25, 7A

DIMENSION DATA(12), TEMB(12), TEHI(12), TSUMB(12), TAB(12), TAI(12)

201 IRUN=1

WRITE(2,1)

1 FORMAT("INPUT THE FOLLOWING DATA USING TELETYPewriter OR READER;

/MONTH, DAY, CHARGE, INCLINATION, DATA SETS/AVG.","FORMAT(A4, IX, I

2, IX, F4. 1, IX, I2, IX, F4. 0")

READ(1, 3) MONTH, DATE, CHARGE, IANGLE, DSETS

3 FORMAT(A2, A2, IX, I2, IX, F4. 1, IX, I2, IX, F4. 0)

204 WRITE(2, 42)

42 FORMAT("INPUT THE FOLLOWING DATA: HW FLOW, CW FLOW, TEST NO, ROTATION

ANGLE","FORMAT(F5. 1, IX, F5. 1, IX, I3, IX, I3")")

READ(1, 4) HFLOB, CFLOB, ITEST, ROTAT

4 FORMAT(F5. 1, IX, F5. 1, IX, I3, IX, I3)

205 WRITE(2, 5)

5 FORMAT(4/)

IDVN=11B

CALL L66(2, IDVN)

CALL STATS(IDVN)

WRITE(IDVN, 301)

301 FORMAT("050")

CALL STATS(IDVN)

WRITE(IDVN, 302)

302 FORMAT("051070")

CALL STATS(IDVN)

READ(IDVN, 303) ISHR, ISMIN, ISSEC, ISMSC

303 FORMAT(I4, I2, I2, I2)

CALL STATS(IDVN)

VRCUB=0. 0

VRAUB=0. 0

ZNI=1. 0

CP=1. 0

DO 100 I=1, 12

TSUMB(I)=0. 0

100 CONTINUE

210 DO 200 J=1, 6

WRITE(IDVN, 304)

304 FORMAT("051020")

CALL STATS(IDVN)

WRITE(IDVN, 305)

305 FORMAT("01E05I#100")

CALL STATS(IDVN)

I=2+J-1

READ(IDVN, 306) DATA(I)

306 FORMAT(2X, F18. 0)

CALL STATS(IDVN)

WRITE(IDVN, 307)

307 FORMAT("05102D")

CALL STATS(IDVN)

WRITE(IDVN, 308)

308 FORMAT("051030")
CALL STATS(IDVN)
WRITE(IDVN,309)
309 FORMAT("#1E5I#10"")
CALL STATS(IDVN)
I=2*J
READ(IDVN,310) DATA(I)
310 FORMAT(2X, F10.0)
CALL STATS(IDVN)
WRITE(IDVN,311)
311 FORMAT("#5I#3D"")
CALL STATS(IDVN)
200 CONTINUE
IF(ZN-DSET5)225,226
220 WRITE(IDVN,312)
312 FORMAT("#5I#7O")
CALL STATS(IDVN)
READ(IDVN,313) IFHR,IMIN,IFSEC,IFMSC
313 FORMAT(I4, I2, I2, I2)
CALL STATS(IDVN)
225 DO 300 I=1,12
TEMP(I)=32.08111197146638.749761*DATA(I)-1321251.435*
DATA(I)**2-59109465.0*DATA(I)**3
TEM(I)=(TEM(I)-32.6)/5.0/9.0
TSUMB(I)=TSUMB(I)+TEMP(I)
TAB(I)=TSUMB(I)/ZN
TAF(I)=(TAB(I)-32.0)*C.555556
300 CONTINUE
HDB=ABS(TEMP(5)-TEMP(7))
CDB=ABS(TEMP(11)-TEMP(9))
DMB=ABS((TEMP(6)+TEMP(3)+TEMP(12))-(TEMP(8)+TEMP(1)+TEMP(10)))/3.
QEB=HFLOB*CP*HDB
QCB=CFLOB*CP*CDF
FCB=QCB*C.55493/C.311
FAB=(QEB+QCB)*C.47746/C.311
UCB=FCB/DMB
UAB=FAB/DMB
HADB=ABS(TAB(5)-TAB(7))
CADB=ABS(TAB(11)-TAB(9))
DMAB=ABS((TAB(6)+TAB(3)+TAB(12))-(TAB(8)+TAB(1)+TAB(10)))/3.
QEAB=HFLOB*CP*HADB
QCAB=CFLOB*CP*CADB
FCAB=QCAB*C.95493/C.311
FAB=(QEAB+QCAB)*C.47746/C.311
UCAB=FCAB/DMAB
UAAB=FAB/DMAB
HFLOI=HFLOB*C.453592
CFLOI=CFLOB*C.453592
HADI=HADB*C.55556
CA DI=CADB*C.55556
DMAI=DMAB*C.55556
QEAI=QEAB*C.2928751
QCAI=QCAB*C.2928751
FCAI=FCAB*C.151481
FAAI=FAAB+3.15481
UCAI=UCAB*5.674466
UAII=UAAB*5.674466
VCUNB= ((VRCUB*(ZI-1.0))+(UCB-UCAB)**2)/ZI
VAUNB= ((VRAUB*(1-1.0))+(UAB-UAAB)**2)/ZI
VRCUB=VCUNB
VRAUB=VAUNB.
IF(ZN-DSET5) 700, 701
700 ZN=ZN+1.0
GO TO 210
701 VCUNB=SORT(VCUNB)*100.0/UCAB
VAUNB=SORT(VAUNB)*100.0/UAAB
WRITE(2, 8)
8 FORMAT(16X,"DATA ANALYSIS FOR THE THERMOSIPHON LOOP"/16X,38"**"16X)
WRITE(2, 10) ITEST, IRUN, MONTTH, IDATE
10 FORMAT("TEST NO.:", IX, 13, IX, "RUN NO.:", IX, 12, /A2, A2, IX, 12, IX, "74")
WRITE(2, 12) ISHR, ISMIN, ISSEC, ISMSC, IFHR, IFMIN, IFSEC, IFMSC
12 FORMAT("START TIME: ", 14X, 12, ":", 12, ":", 12, ", ", 12, "/FINISH TIME: ", 14X, 12, ":", 12, ",", 12, ",", 12, ", ", 12)
WRITE(2, 14) CHARG, ROTAT, ANGL
WRITE(2, 16) DSETS
16 FORMAT(12X, "AVERAGE RESULTS OF ", IX, F4.6, IX, "SETS OF DATA"/12X, "OU
TILITIE", 22X, "VALUE", 6X, "VALUE"/38X, "(ENG FOR UNIT)"/2X, "(S. UNIT)")
WRITE(2, 18) HFLOB, HFLOI, CFLOB, CFLOI
18 FORMAT("LOW RATE OF HOT WATER(LBS/HR, KG/HR): ", 3X, F7.2, 4X, F7.2, /"F
LOW RATE OF COLD WATER(LBS/HR, KG/HR): ", 2X, F7.2, 4X, F7.2, 2X, F7.2, 4X, F7.2)
WRITE(2, 20) TAB(5), TAI(5), TAB(7), TAI(7)
20 FORMAT("TEMP OF HOT WATER IN (F, C): ", 14X, F7.2, 4X, F7.2, /"TEMP OF HOT
WATER OUT (F, C): ", 13X, F7.2, 4X, F7.2)
WRITE(2, 22) TAB(11), TAI(11), TAB(9), TAI(9)
22 FORMAT("TEMP OF COLD WATER IN (F, C): ", 13X, F7.2, 4X, F7.2, /"TEMP OF CO
LD WATER OUT (F, C): ", 12X, F7.2, 4X, F7.2)
WRITE(2, 24) TAB(6), TAI(6), TAB(3), TAI(3), TAB(12), TAI(12)
24 FORMAT("TEMP OF EVAP SURF-TOP (F, C): ", 13X, F7.2, 4X, F7.2, /"TEMP OF EV
WRITE(2, 26) TAB(8), TAI(8), TAB(1), TAI(1), TAB(10), TAI(10)
26 FORMAT("TEMP OF COND SURF-TOP (F, C): ", 13X, F7.2, 4X, F7.2, /"TEMP OF CON
WRITE(2, 28) TAB(2), TAI(2), TAB(4), TAI(4)
28 FORMAT("TEMP OF R113-TOP (F, C): ", 15X, F7.2, 4X, F7.2, /"TEMP OF R113-BOT
TM (F, C): ", 15X, F7.2, 4X, F7.2)
WRITE(2, 30) HADH, HADI, CABD, CADI, DHAD, DHAI
WRITE(2, 32) OABE, OAIE, OCAE, OCAI
32 FORMAT(’CC(BTU/HR,WATT)',24X,F7.2,4X,F7.2,’/CC(BTU/HP,WATT)’,24X
0,F7.2,4X,F7.2) WRITE(2,34) FCAB,FCAI,FAAB,FAAI
34 FORMAT(’FLUX(CC/A)(BTU/HR SOFT,WATT/SCN)’,7X,F7.2,4X,F7.2,’/FLUX(2)
#0A/2A)(BTU/HR SOFT,WATT/SCN)’,6X,F7.2,4X,F7.2) WRITE(2,36) UCAE,UCAB,UAAB,UAAB
36 FORMAT(’U(CC/ADT)(BTU/HR SOFT F,WATT/SCN K)’,4X,F7.2,4X,F7.2,’/U(2)
#0A/2ADT)(BTU/HR SOFT F,WATT/SCN K)’,3X,F7.2,4X,F7.2,’//’) WRITE(2,38) VCUNI,VAUNI
38 FORMAT(’42X,’COND’’,7X,’AUGE’’/’’COEF OF VARIATION OF U(%)’’,13X,F7.1,
#4X,F7.1,8/) WRITE(2,40)
40 FORMAT(’TO RESTART; PUSH RUN, THEN SET SKIP=4. TO CONTINUE RUNS, SKI
P=3. TO ”/’’CONTINUE WITHOUT NEW FLOW DATA, SKIP=2. TO STOP. USE FOR
MAT(F4.8)’’) PAUSE 55 IRUN=IRUN+1 READ(1,2)SKIP
2 FORMAT(F4.8)
IF(3.,SKl) 204,205,203
203 WRITE(2,49)
49 FORMAT(’TEST FINISHED. FOR NEXT TEST PRESS RUN’) PAUSE 66 
GO TO 281
END
SUBROUTINE STATS(IDVN)
350 CALL L66(l,IDVN,ISTAT,ITRLG) IF(ISTAT) 350,351
351 IF(l AND (ISTAT,37B)-1) 350,352,353
353 PAUSE 77
352 RETURN
END
END3
APPENDIX B

4.
INPUT THE FOLLOWING DATA: HV FLOW, CV FLOW, TEST NO., ROTATION ANGLE
FORMAT(F5.1, I4, F5.1, I4, I3, I4, I3)
277.8 277.0 681 068

DATA ANALYSIS FOR THE THERMOSYPHON LOOP
********************************************

TEST NO 681 RUN NO 3
MAY 11 74
START TIME: 7:48:21.5
FINISH TIME: 7:48:56.9
WORKING FLUID: R11
LOOP VOLUME: 217 MILLI LITERS
% CHANGE BY VOLUME: 26.0%
LOOP ROTATION ANGLE: 60 DEGREES
INCLINATION-LOOP PLAN: 0 DEGREES

AVERAGE RESULTS OF 100 SETS OF DATA

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<th>VALUE (ENGL UNIT)</th>
<th>VALUE (S.I. UNIT)</th>
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<td>125.64</td>
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<tr>
<td>FLOW RATE OF COLD WATER (LBS/HR, KG/HR)</td>
<td>277.00</td>
<td>125.64</td>
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<tr>
<td>TEMP OF HOT WATER (°F, °C)</td>
<td>129.72</td>
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<td>TEMP OF HOT WATER OUT (°F, °C)</td>
<td>126.51</td>
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<td>TEMP OF COLD WATER IN (°F, °C)</td>
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<td>TEMP OF COLD WATER OUT (°F, °C)</td>
<td>93.44</td>
<td>34.13</td>
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<tr>
<td>TEMP OF EVAP SURF-TOP (°F, °C)</td>
<td>115.17</td>
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<td>TEMP OF EVAP SURF-MIDDLE (°F, °C)</td>
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<td>TEMP OF EVAP SURF-BOTTOM (°F, °C)</td>
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<td>1.78</td>
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<tr>
<td>TEMP DIFF COLD WATER IN &amp; OUT (°F, K)</td>
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<td>1.66</td>
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<td>CE (BTU/HR, WATT)</td>
<td>880.80</td>
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<tr>
<td>QC (BTU/HR, WATT)</td>
<td>829.74</td>
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<td>FLUX (QC, A) (BTU/HR SQFT, WATT/SQFT)</td>
<td>2547.73</td>
<td>8316.92</td>
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<tr>
<td>U (QC/ADT) (BTU/HR SQFT, WATT/SQFT K)</td>
<td>195.29</td>
<td>1168.19</td>
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<tr>
<td>U (QC/2ADT) (BTU/HR SQFT F, WATT/SQFT K)</td>
<td>282.15</td>
<td>1147.69</td>
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COE OF VARIATION OF U(°C): COND AVERE
4.9 3.9
APPENDIX F

DATA INPUT (VOLT)

0 000988E-6 1 000987E-6 14:54:00
2 000983E-6 3 000984E-6 14:54:07
4 000982E-6 5 000981E-6 14:54:14
6 000982E-6 7 000982E-6 14:54:21
8 000981E-6 9 000984E-6 14:54:28
E-7 OF

DATA ANALYSIS FOR THE THERMOSIPHON LOOP

***************************************************************

DATE: AUGUST 12, 74
TIME: 14:54:0
WORKING FLUID: R113
LOOP VOLUME: 210 MILLI LITERS
% CHARGE (BY VOLUME): 99%
LOOP ROTATION ANGLE: 0 DEGREES
INCLINATION - LOOP PLANE: 0 DEGREES

RESULTS

<table>
<thead>
<tr>
<th>QUANTITY</th>
<th>VALUE</th>
<th>AVGE</th>
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<tbody>
<tr>
<td>FLOW RATE OF HOT WATER (LBS/HR)</td>
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<td>255.0</td>
</tr>
<tr>
<td>FLOW RATE OF COLD WATER (LBS/HR)</td>
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<td>258.0</td>
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<tr>
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<td>76.6</td>
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<td>76.7</td>
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<td>76.9</td>
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<tr>
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<td>76.9</td>
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<td>76.7</td>
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<td>HEAT FLUX (BTU/HR FTSO)</td>
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<td>U BASED ON AVGE Q (BTU/HR FTSO F)</td>
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<td>OVERALL HEAT TRANS. COEFF (BTU/HR FTSO F)</td>
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<td>169.8</td>
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</table>

VARIANCE OF OVERALL HEAT TRANSFER COEFF: 843028
STANDARD DEVIATION OF OVERALL COEFF: 918.17
DATA ANALYSIS FOR THE THERMOSIPHON LOOP

TEST NO 0 RUN 1
MARCH 15 74
START TIME 0:15:3 8
FINISH TIME 0:19:34:7
WORKING FLUID R113
LOOP VOLUME 219 MILLI LITERS
% CHARGE (BY VOLUME) 0%
LOOP ROTATION ANGLE 0 DEGREES
INCLINATION-LOOP PLANE 0 DEGREES

AVERAGE RESULTS OF 500 SETS OF DATA

<table>
<thead>
<tr>
<th>QUALITY</th>
<th>VALUE (ENGL UNIT)</th>
<th>VALUE (SI. UNIT)</th>
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<tbody>
<tr>
<td>FLOW RATE OF HOT WATER (LBS/HR, KG/HR)</td>
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<td>117.48</td>
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<tr>
<td>FLOW RATE OF COLD WATER (LBS/HR, KG/HR)</td>
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<td>117.48</td>
</tr>
<tr>
<td>TEMP OF HOT WATER IN (F, C)</td>
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</tr>
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<td>TEMP OF COLD WATER IN (F, C)</td>
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<td>TEMP OF COLD WATER OUT (F, C)</td>
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<td>TEMP DIFF COLD WATER IN &amp; OUT (F, K)</td>
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<td>QC(BTU/HR, WATT)</td>
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<td>FLUX(QA/2A)(BTU/HR SOFT, WATT/SM)</td>
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<td>U(OC/ADT)(BTU/HR SOFT F, WATT/SM K)</td>
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<td>6282.04</td>
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</table>

COEF OF VARIATION OF U(%) COND AUGE

$\text{COND}$ $\text{AVUGE}$
APPENDIX H

CALCULATIONS OF THE BOILING AND CONDENSATION LENGTHS

The wetted evaporator static length, $Z_{bs}$, and the film condensation length, $Z_c$, can be calculated in terms of % charge, rotation angle, $\theta_R$, the evaporator length, $L$, and the length of cross connecting tube, $L_p$ as follows:

1) The vertical loop \((\theta_R = 0)\)
   (see figure 29-a))
   a) \(0.5 L_s \leq C \leq 1 - 0.5 L_s\)

\[
Z_{bs} = C (L + L_p) - 0.5 L_p \quad \text{A-1a}
\]
\[
Z_c = L - Z_{bs} \quad \text{A-1b}
\]

where

\[
L_s = \frac{L_p}{L + L_p}, \quad C = \frac{\% \text{ charge}}{100}
\]

$L_p$ is the length of the cross connecting tube.

b) \(0.5L_s \geq C\)

\[
Z_{bs} = 0 \quad \text{A-2a}
\]
\[
Z_c = L \quad \text{A-2b}
\]

c) \(1 - 0.5 L_s \leq C\)

\[
Z_{bs} = L \quad \text{A-3a}
\]
\[
Z_c = 0 \quad \text{A-3b}
\]

2) The rotated loop \((\theta_R > 0)\)
   a) High % charge
   i) \(\tan \theta_R < \frac{L}{L_p} \text{ and } (1 - 0.5 L_s (1 + \tan \theta_R)) \leq C\)
ii) \( \tan \theta_R \geq \frac{L}{L_p} \), and \( \{1 - 0.5 \ L_s (1+\cot \theta_R)\} \leq C \)

(see 60% charge in figure (29-c))

\[ Z_{bs} = L \]  \hspace{1cm} \text{A-4a} \\
\[ Z_c = 2(1-C)(L+L_p) \tan \theta_R / (\tan \theta_R+1) \]  \hspace{1cm} \text{A-4b}

where

\[ L_r = \frac{L}{(L+L_p)} \]

b) Intermediate % charge

i) \( \tan \theta_R \leq \frac{L}{L_p} \), and \( 0.5 \ L_s (1+\tan \theta_R) \leq C \)

\( \leq \{1-0.5 \ L_s (1+\tan \theta_R)\} \)

(see 40, 50 and 60% charge in figure (29-c))

\[ Z_{bs} = \{C (L+L_p) - 0.5 \ L_p\} + 0.5 \ L_p \tan \theta_R \]  \hspace{1cm} \text{A-5a} \\
\[ Z_c = L - (2*C/L2-1) + 0.5*L1*\tan(R) \]  \hspace{1cm} \text{A-5b}

ii) \( \tan \theta_R > \frac{L}{L_p} \), \( 0.5 \ L_r (1+\cot \theta_R) \leq C \), and

\( \{1-0.5 \ L_r (1+\cot \theta_R)\} \geq C \)

(see 50% charge in figure (29-c))

\[ Z_{bs} = L \]  \hspace{1cm} \text{A-6a} \\
\[ Z_c = L \]  \hspace{1cm} \text{A-6b}

c) Low % charge

i) \( \tan \theta_R < \frac{L}{L_p} \), and \( 0.5 \ L_s (1+\tan \theta_R) \leq C \)

(see 20% charge in figure (29-b))

\[ Z_{bs} = 2.C(L+L_p) \tan \theta_R / (\tan \theta+1) \] \\
\[ Z_c = L \]

ii) \( \tan \theta_R > \frac{L}{L_p} \), and \( \{1 - 0.5L_r (1+\cot \theta_R)\} \geq C \)

(see 20% and 40% charge in figure (29-c))
\[ Z_{bs} = 2(L+L_p)(1-C) \tan \theta_R/(\tan \theta_R+1) \]  
\[ Z_c = L \]
APPENDIX I
ERROR ANALYSIS

The uncertainty in the calculation of the loop conductance and % charge are estimated in this appendix.

The probably error in a calculated value due to uncertainties in the quantities used in the calculations were estimated using the following equation:

$$W_z = \left[ \left( \frac{\partial z}{\partial x_1} \cdot W_1 \right)^2 + \left( \frac{\partial z}{\partial x_2} \cdot W_2 \right)^2 + \ldots \left( \frac{\partial z}{\partial x_n} \cdot W_n \right)^2 \right]^{\frac{1}{2}}$$

where $z$ is a function of the independent variables $x_1, x_2, \ldots, x_n$, $W_z$ is the probable error in $z$, and $W_1, W_2, \ldots, W_n$ are the estimated error of the respective independent variables.

i) Error in the loop conductance due to inaccuracies in the measurements.

The loop conductance was determined using the following equation:

$$U = \frac{\dot{m}_{hw} \cdot C_{hw} \Delta T_{hw} + \dot{m}_{cw} (C_{cw} \Delta T_{cw})}{2 \Delta T \text{ec} \pi \cdot \text{D.L}}$$

where the mass flow rates of the hot and the cold water ($\dot{m}_{hw}, \dot{m}_{cw}$) were measured to an accuracy of $\pm 2$ lb/hr. This accuracy was found experimentally by measuring the time for sampling & mass.

The variation of the specific heats of the hot and cold water ($C_{hw}, C_{cw}$) = 1 Btu/lbm$^o$F) with changing the temperature level was
so small that it was neglected.

$$\Delta T_{hw} = T_{hw_i} - T_{hw_o},$$
where $T_{hw_i}$ and $T_{hw_o}$ are temperature readings at the inlet and outlet of the evaporator respectively.

$$\Delta T_{cw} = T_{cw_i} - T_{cw_o},$$
where $T_{cw_i}$ and $T_{cw_o}$ are temperature readings at the inlet and outlet of the condenser respectively.

$$\Delta T_{e-c} = \Delta T_{ec} = \frac{(T_{e1}+T_{e2}+T_{e3}) - (T_{c1}+T_{c2}+T_{c3})}{3}$$

where $T_{e1}$, $T_{e2}$, $T_{e3}$ and $T_{c1}$, $T_{c2}$, $T_{c3}$ are surface temperature readings of the evaporator and the condenser respectively.

The manufacturing tolerance of the inside diameter of the tube, and the length of the evaporator and the condenser were 0.3" ±0.005" and 4' ±0.0052' respectively.

Then according to equation (AI -1) the probable error in the loop conductance, $U$ may be written as:

$$W_u = \left[ \left( \frac{\partial u}{\partial \Delta T_{hw}} \cdot W_{\Delta T_{hw}} \right)^2 + \left( \frac{\partial u}{\partial \Delta T_{cw}} \cdot W_{\Delta T_{cw}} \right)^2 \right]$$

$$+ \left( \frac{\partial u}{\partial W} \cdot W_{W} \right)^2 + \left( \frac{\partial u}{\partial L} \cdot W_{L} \right)^2$$

$$+ \left( \frac{\partial u}{\partial \Delta T_{ec}} \cdot W_{\Delta T_{ec}} \right)^2$$

where

$$W_{\Delta T_{hw}} = \left[ \left( \frac{\partial \Delta T_{hw}}{\partial T_{hw_i}} \cdot W_{T_{hw_i}} \right)^2 + \left( \frac{\partial \Delta T_{hw}}{\partial T_{hw_o}} \cdot W_{T_{hw_o}} \right)^2 \right]^{\frac{1}{2}}$$

$$W_{\Delta T_{cw}} = \left[ \left( \frac{\partial \Delta T_{cw}}{\partial T_{cw_i}} \cdot W_{T_{cw_i}} \right)^2 + \left( \frac{\partial \Delta T_{cw}}{\partial T_{cw_o}} \cdot W_{T_{cw_o}} \right)^2 \right]^{\frac{1}{2}}$$
\[ W \Delta T_{ec} = \left[ \left( \frac{\partial T_{ec}}{\partial T_{e1}} \cdot W T_{e1} \right)^2 + \left( \frac{\partial T_{ec}}{\partial T_{e2}} \cdot W T_{e2} \right)^2 \right. \]
\[ \left. + \left( \frac{\partial T_{ec}}{\partial T_{e3}} \cdot W T_{e3} \right)^2 + \left( \frac{\partial T_{ec}}{\partial T_{c1}} \cdot W T_{c1} \right)^2 \right. \]
\[ \left. + \left( \frac{\partial T_{ec}}{\partial T_{c2}} \cdot W T_{c2} \right)^2 + \left( \frac{\partial T_{ec}}{\partial T_{c3}} \cdot W T_{c3} \right)^2 \right] \]

or dividing this equation \((A I - 3 a)\) by \(u\), the % error can be obtained as:

\[ \frac{W_u}{U} = \left[ \left( \frac{\Delta T_{hw} \cdot W m_{hw}}{m_{hw} \Delta T_{hw} + m_{cw} \Delta T_{cw}} \right)^2 + \left( \frac{\Delta T_{cw} \cdot W m_{cw}}{m_{hw} \Delta T_{hw} + m_{cw} \Delta T_{cw}} \right)^2 \right. \]
\[ \left. + \left( \frac{m_{hw} \Delta T_{hw}}{m_{hw} \Delta T_{hw} + m_{cw} \Delta T_{cw}} \right)^2 + \left( \frac{m_{cw} \Delta T_{cw}}{m_{cw} \Delta T_{cw}} \right)^2 \right] \frac{1}{2} \]

A sample calculation of probable error

**Working fluid:** R-11

**Charge:** 30%

**loop orientation:** \(\theta_R = 60^\circ\) \(\theta_I = 0^\circ\)

\(\Delta T_{hw} = 4.64^\circ C\)

\(\Delta T_{cw} = 4.32^\circ C\)

\(\Delta T_{ec} = 22.31^\circ C\)

**water flow rates:** \(m_{hw} = m_{cw} = 275\) lb/hr

Then by using equation \((A I - 3 b)\):

\[ \frac{W_u}{U} = \pm 0.048 = \pm 4.8\% \]
ii) Error in the loop conductance due to three point temperature average.

The maximum error which may result from the use of only three thermocouples to determine the average surface temperature of the evaporator and the condenser may be calculated by using the following experimental data which yield the greatest possible errors for all the data studied:

a) For dryout in the evaporator

Test # 678 Run # 2
Working fluid: R-11
Charge: 20%
Loop orientation: $\theta_R = 0^\circ$, $\theta_I = 0^\circ$

Temperatures (°C):
Source and Sink: $T_{hw} = 54.2$, $T_{cw} = 32.6$
evaporator - top, $T_{e1} = 53.5$
" - middle, $T_{e2} = 53.3$
" - bottom, $T_{e3} = 46.8$
condenser - top, $T_{c1} = 37.3$
" - middle, $T_{c2} = 37.3$
" - bottom, $T_{c3} = 35.5$

Temperature difference, $T_{e} - T_{c} = 14.5$
Loop conductance, $U = 2664$ w/m°k

Since the temperatures at the top and the middle of the evaporator, $T_{e1}$ and $T_{e2}$ respectively, are approximately the same as the source temperature, $T_{hw}$, these thermocouples are located in the dryout region. The thermocouple at the bottom of the evaporator, $T_{e3}$
is in wetted region. When the region surrounding a thermocouple changes between wet and dry, the temperature indicated may then be expected to change by approximately 6.5°C. Since no other changes take place, \( U \) will also change by the same percentage as the mean temperature difference. Thus

\[
\frac{\Delta U}{U} = \frac{\Delta T_{\text{change}}}{3(T_e - T_c)} = \frac{6.5}{3 \times 14.5} \approx 14\% \quad (\pm 7\%)
\]

b) When flooding in the condenser

Test # 664  Run #3
working fluid: R-11
Charge: 50%
Loop Orientation: \( \theta_R = 0^\circ, \theta_I = 0^\circ \)
Temperatures (°C);
Source and Sink: \( T_{hw} = 54.3^\circ, T_{cw} = 32.2 \)
Evaporator- top, \( T_{e1} = 48.6 \)
Evaporator-middle, \( T_{e2} = 49.3 \)
Evaporator-bottom, \( T_{e3} = 51.0 \)
Condenser- top, \( T_{c1} = 40.3 \)
Condenser- middle, \( T_{c2} = 39.6 \)
Condenser- bottom, \( T_{c3} = 32.6 \)
Temperature difference, \( T_e - T_c = 12.1^\circ \)
loop conductance, \( U = 408.9 \text{ w/m}^2\text{°k} \)

The thermocouples at the top and the middle of evaporator, \( T_{c1} \) and \( T_{c2} \) respectively are located in film condensation condition, and the thermocouple, \( T_{c3} \) at the bottom of the condenser is in a flooding region. When the region surround-
ing a thermocouple changes between flooding and film condensation, the temperature indicated may then be expected to change by 7.0°C and thus the loop conductance will be changed as:

\[
\frac{\Delta U}{U} = \frac{\Delta T_{\text{change}}}{3(T e - T_c)} = \frac{7}{3.12.1} \approx 18\% (\pm 9\%)
\]

iii) Error in % charge calculation

The % charge was determined as:

\[
\% C = \frac{V_T - V_W - V_B}{V_L}
\]

or

\[
\% C = \frac{V_{\text{top}} + V_L - V_W}{V_L}
\]

where \( V_T = V_{\text{top}} + V_L + V_B \)

\( V_{\text{top}}, V_L \) and \( V_B \) were the volumes of the top, the main, and the bottom of the loop respectively. \( V_W \) was the working fluid volume withdrawn from the loop.

The accuracies of these volumes were:

\[
V_{\text{top}} = 10.5 \pm 0.5 \text{ ml}
\]

\[
V_L = 187 \pm 1 \text{ ml}
\]

\[
V_B = 20 \pm 1 \text{ ml}
\]

Then the probable error in % charge may be estimated, by using equation (AI - 1), as:

\[
W \% C = \left[(\frac{\partial \% C}{\partial V_{\text{top}}} \cdot W_{V_{\text{top}}})^2 + (\frac{\partial \% C}{\partial V_L} \cdot W_{V_L})^2 + (\frac{\partial \% C}{\partial V_W} \cdot W_{V_W})^2\right]^{\frac{1}{2}} \quad \text{AI-6a}
\]
or % error in % charge may be written as:

\[
\frac{W_{%C}}{\%C} = \left( \frac{W_{V_{top}}}{(V_{top} + V_L - V_W)} \right)^2 + \left( \frac{(V_W - V_{top}) \cdot W_{V_L}}{V_L (V_{top} + V_L - V_W)} \right)^2
\]

\[
+ \left( \frac{W_{V_W}}{(V_{top} + V_L - V_W)} \right)^2
\]

A sample calculation of 30% charge:

\[
\frac{W_{%C}}{\%C} = \pm 0.01.8 = \pm 1.8\%
\]
APPENDIX J
MODEL OF EXPERIMENTAL LOOP

In the loop simulation using equation (2-2), the heat transfer coefficients for the evaporator and the condenser and the reciprocal of the wetted liquid fraction in the wetted evaporator were found as follows:

i) The evaporator heat transfer coefficient, $h_b$.

For the vertical position of the loop the evaporator heat transfer coefficient, $h_b$, was calculated using Rohsenow's nucleate pool boiling equation (1-7) and found to be approximately 150 Btu/hr ft² °F and 170 Btu/hr ft² °F for R-113 and R-11 respectively. In making this calculation the difference between the evaporator wall and the working fluid temperature was taken as the difference between the mean wall temperature and the saturation temperature at the middle of the top cross connecting tube when peak performance occurred (no super heating and no excess pressure drop due to liquid carryover). The values used for the empirical constant, $C_{sf}$, $r$, and $s$ are shown in Table J-a. For R-11 the values of $C_{sf}$, $r$, and $s$ as determined by Ali [16] based on the data of Blatt and Adt were used.

<table>
<thead>
<tr>
<th></th>
<th>$C_{sf}$</th>
<th>$r$</th>
<th>$s$</th>
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<tr>
<td>R-113</td>
<td>0.0037</td>
<td>0.33</td>
<td>1.7</td>
<td>Tang</td>
</tr>
<tr>
<td>R-11</td>
<td>0.01</td>
<td>0.769</td>
<td>1.7</td>
<td>Ali</td>
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</table>
The evaporator heat transfer coefficient could be calculated for \( \theta_R = 60^\circ \) and 20\% charge where the measured maximum performance occurred (dryout at the top of the evaporator and no flooding in the condenser) by dividing the heat flow rate from the evaporator to the condenser by the temperature difference between the average evaporator surface temperature and the working fluid at the middle of the top cross connecting tube. This yielded values of 210 Btu/hr ft\(^2\) °F for R-113 and 310 Btu/hr ft\(^2\) °F for R-11.

In order to obtain a smooth variation of the evaporator heat transfer coefficient with rotation angle, a parabolic equation of the form,

\[
h_b = h_{bm} - \frac{(\theta_R - \theta_{bm})^2}{C_1}
\]

...J-1

was passed through the values of the heat transfer coefficient for the vertical and at 60°. It was found by trial and error that \( \theta_m = 57^\circ \) yielded the best fit. The equations for the evaporator heat transfer coefficient for R-113 and R-11 can be written as follows:

For R-113 \[ h_b = 210 - \frac{(\theta_R - 57)^2}{81.23} \text{ Btu/hr ft}^2 \text{ °F} \]  ...J-1a
For R-11 \( h_b = 310 - \frac{(\theta_R - 57)^2}{27.075} \) Btu/hr ft\(^2\) °F \( \ldots \) J-1b

where \( \theta_R \) is the angle of rotation from the vertical.

ii) The Condensation Heat Transfer Coefficients, \( h_c \)

For the condenser, the film condensation heat transfer coefficient varies with the loop rotation angle and the temperature difference between the working fluid saturation temperature and the condenser wall temperature. This temperature difference depends on the sink and source temperature difference and the loop rotation angle. The condensation heat transfer coefficient, therefore, is a function only of the angle of loop rotation when the source and sink temperature is kept constant.

As mentioned in Chapter II, the condensation heat transfer coefficient first increases with rotation angle and then decreases near to the horizontal position. According to the experimental study of Chato, the maximum value is expected to occur at a loop rotation angle of approximately 80°. In order to simulate this type of variation a parabolic equation of the form, was assumed as

\[
\frac{h_c}{h_{cm}} = \frac{(\theta_R - \theta_{cm})^2}{C_2}
\]

\( \ldots \) J-2

To evaluate the constants, \( h_{cm} \) and \( C_2 \), two data points are required; the value of the condenser heat transfer coefficient for the vertical position was calculated using equation (1-8). A second point was obtained by using the experimental data for a 60° angle of rotation with 20% charge where dryout occurred at the top of the evaporator.

For the vertical case, the experimental value
of the temperature difference between the average condenser surface and the working fluid at the middle of the top cross connecting tube was used. The values calculated were approximately 170 Btu/hr ft\(^2\) °F for R-113 and 185 Btu/hr ft\(^2\) °F for R-11.

The experimental values, which were calculated by dividing the flow rate from the evaporator to the condenser with the temperature difference between the average condenser surface and the working fluid at the middle of the top cross connecting tube were found to be approximately 480 Btu/hr ft\(^2\) °F for R-113 and 570 Btu/hr ft\(^2\) °F for R-11.

The equations which passed through the values of the heat transfer coefficient for R-113 and R-11 can be written as follows:

For R-113 \( h_c = 500 - \frac{(\theta_R - 80)^2}{18.29} \) Btu/hr ft\(^2\) °F. ..J-2a

For R-11 \( h_c = 600 - \frac{(\theta_R - 80)^2}{15.65} \) Btu/hr ft\(^2\) °F. ..J-2b

iii) The Reciprocal of the Liquid Fraction, \( \frac{Z_b}{Z_{bs}} \)

The reciprocal of the wetted length liquid average fraction, \( \frac{Z_b}{Z_{bs}} \) was found to be approximately 4 for the vertical case based on the fact that dryout just ceased with a 34% charge in the loop, and 2.54 for a 60° angle of rotation with a 20% charge for R-11. A plot of the reciprocal of the wetted length liquid average fraction \( \frac{Z_b}{Z_{bs}} \) versus the rotation angle, \( \theta_R \) for the optimum points on each curve
of constant charge in figure 28 indicated that the curve of the reciprocal of liquid fraction varied with loop rotation angle in the form of an "S" curve. This variation was reasonably represented by the equation

\[
\frac{Z}{Z_{bs}} = 2.54 - \frac{6R - 455}{41.66}
\]
Fig. 28. Reciprocal of the liquid-fraction vs rotation angle.
Fig. 29 Effect of rotation on static liquid levels in a rectangular loop
APPENDIX K

PRICE COMPARISON OF A TWO PHASE THERMOSIPHON
WITH A RECUPERATOR

A price comparison between a two-phase thermosiphon loop and a comparable direct gas-to-gas heat exchanger is made given the following operating conditions.

Recovery heat: 2700 Btu/hr
Exhaust air flow: 500 Scfm
Supply air flow: 500 Scfm
Exhaust air temperature at inlet: 75°F
Exhaust Air temperature at outlet: 70°F
Supply air temperature at inlet: 45°F
Supply air temperature at outlet: 50°F
Fin Material: Aluminum
Frontal dimension: 1' x 1'
Depth: 1'

A. Price of Waste Heat Recuperator

A waste heat recuperator which is operating with the above condition could be purchased at $216, f.o.b. price at General Motors Corporation, Lockport, New York, U.S.A. as of November 13, 1975.
B. Two phase Thermosiphon Heat Exchanger

The price estimation of the author's heat exchanger based on the characteristics, in figure 100, Compact Heat Exchanger by Kays and London [18], with the accompanying designs specifications which are;

Finned Circular Tubes

 Tube inside diameter: 0.311"
 Fin Pitch: 8.0 per inch
 Flow passage hydraulic diameter, 4 $h_r$: 0.01192"
 Fin thickness, $\delta$: 0.013"

Fire-flow area/frontal diameter $\sigma$: 0.534
Heat transfer area/total volume, $\alpha$: 179 ft$^2$/ft$^3$
Thermal conductivity, $K$: 134 Btu/hr ft$^2\circ$F
Heat exchanger frontal area: 1'x1'
Reynolds Number, $N_\text{ea}$

Air density, $\rho_a$ = 0.075 lbm/ft$^3$

Air viscosity, $\mu_a$ = 0.000744 lbm/ft min

$$G = \frac{\text{Air mass flow rate}}{\text{Free flow area}}$$

$$= \frac{500 \times 0.075}{1 \times 0.534} = 70.2 \text{ lbm/min ft}^2$$

$$N_{Ra} = \frac{4h_r G}{\mu} = \frac{0.01192 \times 70.2}{0.000744} = 1125$$

Unit air film conductance, $h_a$ from figure 100, Compact Heat Exchanger by Kays and London.
\[
N_{ST} \cdot \frac{P_r^{2/3}}{P_r} = 0.03
\]

where \( N_{ST} \): Stanton Number \((h / \rho C_p)\)

Pr: Prandtl Number \((\mu C_p / K)\)

\[
h_a = \frac{G C_p}{0.7887} = 4213.5 \times 24 \times \frac{0.01}{0.7887} 
\]

\( \approx 13 \text{ Btu/hr ft}^2 \circ \text{F} \)

Total inside tube area = 13.7 ft\(^2\)/ft\(^3\)

Fin Efficiency, \( \zeta_f \) (air side only)

\[
m = \sqrt{\frac{2h}{K \sigma}} = \sqrt{\frac{2 \times 13}{134 \times 0.0011}} = 13 \text{ ft}^{-1}
\]

\[
m_f = 13 \times \frac{0.299}{12} = 0.32
\]

\[
\zeta_f = \frac{\tanh(0.32)}{0.32} = \frac{0.31}{0.32} = 0.97
\]

Overall Surface Effectiveness, \( \zeta_o \)

\[
\zeta_o = 1 - \left( \frac{A_t}{A_a} \right) (1 - \zeta_f)
\]

\[
= 1 - 0.829 (1 - 0.97) \approx 0.98.
\]

From the calculation it has been determined that the required gas to liquid heat exchanger dimensions are 1" x 1" x 2 inches to satisfy the given operating conditions
where \( U_T \) is the conductance of the thermosiphon loop and \( A_T \) is the inside area of the thermosiphon loop.

\[
\frac{1}{AU} = \frac{1}{12 \times 0.98 \times 30} + \frac{1}{200 \times 2.3} + \frac{1}{13 \times 0.98 \times 30}
\]

Thus

\[
AU = 135 \text{ Btu/hr °F}
\]

\[
NTU = \frac{AU}{w} = \frac{135}{500 \times 0.045 \times 0.24 \times 60} = 0.25
\]

where NTU is number of heat transfer units of an exchanger, \( w \) is flow stream capacity \((m \cdot C_p)\), Btu/hr °F

From figure 5, Compact Heat Exchanger, cross flow heat exchanger with fluid unmixed. The effectiveness, \( \varepsilon \), is

\[
\varepsilon = 0.21
\]

\[
\varepsilon = \frac{T_{\text{fin}} - T_{\text{hout}}}{T_{\text{fin}} - T_{\text{cin}}} = \frac{75 - T_{\text{hout}}}{75 - 45} = 0.21
\]

\[
T_{\text{hout}} = 68.8 \text{ °F}
\]

Similarly, \( T_{\text{cout}} = 51.3 \text{ °F} \)

A gas to liquid heat exchanger could be purchased at \$1.72/ft^2 of heat exchanger area as of November. Thus the total heat exchanger cost for the two units required is \$103 where the units must still be interconnected and charged.
C. COMPARISON

DIRECT GAS-TO-GAS HEAT EXCHANGER

COLD AIR

HOT AIR

°F

50 75

70 45

AREA
Effectiveness, $\varepsilon = 17\%$
PRICE: $216$
at New York
U.S.A.

THERMOSIPHON LOOP HEAT EXCHANGER

COLD AIR

HOT AIR

°F

75 68.8

51.3 45°F

AREA
$\varepsilon = 21\%$
PRICE: $103$
at Windsor, Ont.
Canada

Due to difficulty of the manufacturing cost estimation of thermosiphon heat exchanger, the exact price comparison of two heat recovery systems is not easy. However, according to the above comparison, it is very feasible that the thermosiphon loop heat exchanger can be competitive with the Direct heat exchanger.
APPENDIX L

EXPERIMENTAL DATA

1) Table E-1  TEST SEQUENCE I

Effect of % Charge and Rotation Angle

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<th>$\theta_R^\circ$</th>
<th>$P_{psia}$</th>
<th>$U_{Btu/hr ft^2^\circ F}$</th>
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## 2) Table E-2

### TEST SEQUENCE II

Effect of Sink Temperature Change

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\[ \text{See Test Sequence I for this data.} \]
3) Table E-3

TEST SEQUENCE III

Continuation of Test Sequence I

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Table E-3 (Continued)

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4) **Table E-4**  
**TEST SEQUENCE IV**

**Effect of System Pressure for R-113**

a) For R-113

Charge = 30%  \[ \theta_I = 0^\circ \]

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Effect of Inclination Angle

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Table E-6

TEST SEQUENCE VI

b) For R-11 Effect of System Pressure for R-11

Charge = 30%

θ_I = 0°

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<th>Test #</th>
<th>T_{hw} °F</th>
<th>T_{cw} °F</th>
<th>θ_R °</th>
<th>P psia</th>
<th>U_{Btu/hr ft^{2}°F}</th>
<th>U w/m²°K</th>
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6) Table E-7  

TEST SEQUENCE VII

Effect of Surface Temperature Difference, $T_e - T_c$

R-11  
Charge = 30%  
$\theta_I = 0^\circ$

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<th>Test #</th>
<th>$T_{hw}$ °F</th>
<th>$T_{cw}$ °F</th>
<th>$\theta_R$ °F</th>
<th>$P$ psia</th>
<th>$U$ Btu/hr ft²°F</th>
<th>$U$ W/m²K</th>
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7) Table E-8

TEST SEQUENCE VIII

Effect of % Charge and Rotation Angle

\( T_{hw} = 130^\circ F \quad T_{cw} = 90^\circ F \quad \theta_I = 0^\circ \)

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<th>( \theta^\circ_R )</th>
<th>P (psia)</th>
<th>( U_{\text{Btu/hr ft}^2{^\circ F}} )</th>
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<td>$U_{w/m^2^\circ K}$</td>
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### APPENDIX M

**PROPERTIES OF REFRIGERANTS R-113 AND R-11**

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<td>Boiling temperature at 14.7 psia, °F</td>
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<td>Freezing temperature at 14.7 psia, °F</td>
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<td>Critical pressure, psia</td>
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<td>Critical density, lb/cu ft</td>
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<tr>
<td>Density of liquid, 86°F, lb/cu ft</td>
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<td>96.96</td>
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<td>Sp vol of sat vapor 5°F, cu ft/lb</td>
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<td>1.12</td>
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</table>
VITA AUCTORIS

1938  Born in South Korea on December 31.
1956  Graduate from Tae Jun High School, Choong Nam KOREA
1960  Received B.A.Sc. in Mechanical Engineering from the Hang Yang University, Seoul, KOREA
1961-1969  Employed by Bae Chang Ind. Co. Ltd., Pusan, KOREA
1976  Currently a candidate for the Degree of Master of Applied Science in Mechanical Engineering at the University of Windsor, Windsor, Ontario.