Two phase multiple-tube thermosiphon loop an experimental study.

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LA THÈSE A ÉTÉ MICROFILMÉE TELLE QUE NOUS L'AVONS REçUE
TWO PHASE MULTIPLE-TUBE THERMOSIPHON LOOP
(AN EXPERIMENTAL STUDY)

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

SAMPATH, SRINIVAS

WINDSOR, ONTARIO, CANADA
1984
ABSTRACT

An experimental investigation has been carried out at the University of Windsor to determine the operating characteristics of a recirculation two-phase thermosiphon loop in which both the evaporator and condenser consist of a bank of tubes running from an inlet to an outlet header. The recirculation is achieved through the use of a separator at the evaporator outlet header which permits the liquid collected to be recirculated directly to the evaporator inlet header. Each heat exchanger consisted of three identical copper tubes, 3/8 inch diameter in one unit and 1/4 inch in the other. Either unit could be used as the evaporator or the condenser. The system was instrumented to permit the measurement of tube surface temperatures, fluid temperatures, fluid flow rates and heat transfer rates. The length of the heat exchanger tubes was fixed, however all other parameters could be varied independently of each other including tube diameters, the angle of inclination of either heat exchanger, the operating temperature difference, the mean temperature of the loop, the elevation of one heat exchanger with respect to the other, the working fluid and the charge in the loop.

The experimental results were used to develop a computer simulation program. The simulation and the experimental work were designated as ASHRAE PROJECT 188. The experimental parameters were varied to give a severe test to the simulation program.
It was concluded that the simulation program was able to accurately predict the behaviour of such systems. Also the recirculation two-phase multi-tube thermosiphon loop was found to yield higher loop conductances and to be much less sensitive to operating conditions than a similar non-recirculation loop.
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NOMENCLATURE

A  Surface area (m²).
D  Diameter (cm).
h  Heat transfer coefficient (W/m² °K).
L  Length (m).
Q  Heat flow rate (W).
R  Resistance (m² °K/W).
S fg  Change in specific entropy between saturated liquid and saturated vapour state (J/Kg °K).
T  Temperature (°C).
U  Conductance (W/m² °K).
V fg  Change in specific volume between saturated liquid and saturated vapour (m³/Kg).
Z  Elevation of condenser base above the evaporator base (m).
θ  Angle of inclination of the evaporator (degree).
φ  Angle of inclination of the condenser (degree).
∠  Angle.
Δ  Difference.

SUBSCRIPTS

e  Evaporator
c  Condenser
a  Average of evaporator and condenser parameters
CHAPTER I
INTRODUCTION

1.1 GENERAL

Projections by the Federal Government for the 1980's have revealed that 53% of Canada's total energy consumption will be rejected in the form of waste. Because this will generally be in the form of thermal energy, heat reclaiming equipment and systems are becoming more important and are receiving increased attention.

Heat reclamation can be divided into three major categories: 1. Heat is recovered from process flue gases and is recycled directly into the process for re-use; 2. the recovered waste heat from process exhausts is used to preheat make-up air during the heating seasons; and 3. the use of comfort exhaust to preheat ventilation fresh air during the heating season, and to precool it during summer months. The thermosiphon loop is the most recent addition to heat reclaiming equipment - it offers a number of features and advantages making it a practical and efficient method of heat reclamation in both industrial and commercial applications.

A thermosiphon loop heat-exchanger is a device which permits heat energy to be absorbed from a warm region (e.g. the warm gas in an exhaust duct) by an intermediate working fluid and be transported by natural circulation of the working fluid to a remote cool region (e.g. a cold air intake duct) where it gives up its heat. In thermosiphon loops
where there is no phase change for the working fluid, a significant elevation and temperature difference between hot and cold regions is required to generate buoyancy forces sufficient to circulate the working fluid, whereas in loops where there is a phase change of the working fluid, the buoyancy forces are much higher, arising from the large decrease in density which occurs when the liquid vaporizes. Attention could be brought to the operation of a 'Heat Pipe' in this context. The present study is concerned only with thermosiphon loops where there is a phase change of the working fluid at the warm and cool regions and hence are known as 'two phase natural circulation thermosiphon loops'.

In a two-phase thermosiphon loop, the working fluid is sealed within the loop system and does not come into direct contact with either the hot source or cold sink fluids. In a typical air to air heat exchanger installation a thermosiphon loop system would consist of an evaporator and condenser finned tube coil, each mounted in the exhaust and supply ducts respectively, with interconnecting vapour and liquid return lines as shown in Figure 1. Provided that liquid is present in the warm region, the working fluid will circulate through the coils and interconnecting tubing by virtue of the saturation pressure difference between the two coils. It should be observed that if the orientation of the system permits liquid to be present at all times in both coils, then the loop can transfer heat in either direction (bi-directional). If the liquid working fluid can drain completely from one coil then the system can transfer heat only from the wet to the dry coil (uni-directional).
Fig. 1 Schematic elevation view of coil loop thermosiphon heat exchangers.
(a) Bidirectional installation. Loop can transfer heat in either direction.
(b) Unidirectional installation. Loop can transfer heat only from B to A.
Such an energy transfer from the warm to the cool region is also obtainable from recuperative heat exchangers (used in steam power plants) and in rotary regenerative heat exchangers (used in gas turbine power plants). These are direct transfer heat exchangers having the limitation that the two gas streams (warm and cool) have to be in close proximity. Forced circulations indirect transfer heat exchangers have no such deficiencies but they need energy consuming pumps to circulate the working fluid between the source and sink regions. A natural circulation two-phase thermosiphon loop does not need a pump to circulate the working fluid. Also since the essential heat transfer phenomena occur during boiling and condensing processes, high heat exchanger conductance values could be expected. Such a loop will be light in weight, as it is only partially filled with working fluid, quiet in operation, and can transfer heat in either direction in a bidirectional configuration, a feature that is ideal for winter heating and summer cooling.

1.2 OPERATING PRINCIPLE OF A THERMOSIPHON LOOP

In a thermosiphon loop such as shown in Figure 2, a working fluid is sealed within the tube in a two-phase state. Under static (isothermal) conditions, the liquid collects in the lower portion of the loop and the vapour occupies the remaining volume. The vapour pressure in the loop is equal to the saturation pressure at the Liquid-vapour interface, which in turn is dependent upon the temperature at the interface.
FIG. 2: SCHEMATIC ELEVATION OF THERMOSIPHON LOOP
If a temperature difference is created between the liquid-vapour interfaces (in the evaporation and condenser regions) of the loop, a corresponding saturation pressure difference is established which causes vapour to flow from the warm to the cool region where it condenses, thus transferring its latent heat to the cool region. The condensate returns to the bottom of the evaporator by gravity where the working fluid, heated in this warm region, vapourizes, thus completing a cycle. A more detailed description on the principle of operation of thermosiphon loops is available in references 4, 7, 8 and 9.

1.3 PERFORMANCE RATING

The ability of such a loop to transfer energy from one region to another may be expressed in terms of the ratio of heat transfer rate per unit tube surface area, Q/A, to the temperature difference, ΔT, between the two regions, a rating which will be referred to as the loop conductance, U. Thus

\[
U_{a,a} = \frac{Q_a}{A_a \cdot \Delta T}
\]

where

- \( U_{a,a} \) = loop conductance based on average heat transferred across the two heat exchangers and the average temperature difference between the two heat exchangers.
- \( A_a \) = average surface area of the two heat exchangers.
- \( \Delta T \) = difference in the average surface temperatures of the tubes in the two heat exchangers.
Appendix 'A' discusses the other possible ways of defining U and a sample print-out of results in Appendix 'C' shows seven such combinations.

1.4 INVESTIGATION AT THE UNIVERSITY OF WINDSOR

The initial study on thermosiphon loops was sponsored by ASHRAE as Research Project 140 entitled 'Development of Performance Characteristic for Two-Phase Thermosiphon Loops'. The report on that project (9) was completed in 1975. Papers related to this study were subsequently published in the ASHRAE Transactions (4,5,6).

The RP 140 investigation consisted of the development of a computer program to simulate a single tube thermosiphon loop having the configuration shown schematically in Figure 3. The validity of this program was checked by comparing the simulation results with those from experimental studies carried out on two loops of the configuration shown in Figure 4 (8,9). This dual approach, together with the opportunity to visually observe the evaporator and condenser in operation through the glass outer tubes used in the system as shown in Figure 4(b) provided the research team with a very good insight into the system behaviour.

The performance of any given single tube thermosiphon loop was found to be sensitive to the operating temperature difference imposed upon it. Peak performance for these loops occurred when the vapour quality at the outlet end of the evaporator section was close to 1. When operating under these conditions, any increase or decrease in the applied temperature difference caused the performance to fall off.
The evaporator, condenser, vapour and condensate tube lengths and diameters must be specified as well as $\alpha$, $\theta$ and $\Delta$, the angle of inclination of the evaporator and of the condenser from the vertical and the elevation of the condenser base above the evaporator base respectively.

**FIG. 3**: THERMOSIPHON LOOP SIMULATED BY THE COMPUTER PROGRAM FOR RP 140
1 and 2: Hot water (source) inlet and outlet, respectively.
3 and 4: Cold water (sink) inlet and outlet, respectively.

FIG. 4: EXPERIMENTAL THERMOSIPHON LOOPS SIMULATED BY THE COMPUTER
Evaporator dry out occurred when the temperature difference was increased. A reduced temperature difference caused an increased vapour header pressure drop and condenser 'flooding' due to an increase in liquid carryover.

Based on the results of the first project, a further study was initiated to investigate the behaviour of thermosiphon loops with liquid-vapour separation at the evaporator exit, as shown in Figure 5. In such a loop, the 'flooding' liquid would be directly bypassed to the evaporator inlet header thus allowing only vapour to flow to the condenser. This modification eliminates both of the primary causes for the drop off in peak performance when the operating temperature difference deviates from the design peak performance value. It also facilitates the use of multiple tubes running between common inlet and outlet headers as would be the case in using standard commercial single pass, multi-tube evaporator and condenser coils. This study, involving both the experimental investigations and the computer simulations, were designated as RP 188 by ASHRAE. The experimental results were used by Ali (7) in his development of a computer simulation program. The experimental setup had the following:

1) Use of precision flow meters to measure working fluid flow rates and flow oscillations,
2) Facility to set desired elevations and angle of inclinations of evaporator and condenser heat exchangers,
3) Ability to charge the system with the required amount of working fluid,
FIG. 5: SCHEMATIC ELEVATION OF COIL LOOP THERMOSIPHON SHOWING SEPARATOR AND LIQUID RECIRCULATION
4) Use of temperature measuring probes at different locations in the system and,

5) Use of a mini-computer data acquisition system to calculate the experimental results.

More information on the experimental apparatus is given in Section 1.6 'Design Objectives' and Chapter II.

1.5 LITERATURE SURVEY

To the author's knowledge, the only background information available on the performance of two-phase thermosiphon loops as waste heat recovery system has been the work done at the University of Windsor. This was dealt with in Section 1.4 above.

The other aspect of the literature survey was to determine whether or not flow instability conditions would exist and cause performance problems in multiple tube thermosiphon loop systems for typical operating conditions encountered in air to air heat exchangers. Almost all the published literature dealt with nuclear reactor stability and all of them devoted to the case of constant heat flux situations. Wallis and Hensley (17) reported that a flow restriction upstream of the entrance to an evaporator tube stabilizes the flow around a natural circulation loop whereas any flow restriction between the evaporator outlet and condenser inlet promotes unstable operation. Thus pulsating flows can be controlled by inserting a flow restriction at the inlet of each evaporator tube. Such a restriction will also promote boiling and hence provide a secondary benefit, but will reduce the maximum heat recovery
capacity of the system. However, this is not a problem since
the loops are not designed for maximum performance but for
economical and reasonable performance over a wide range
of conditions. Tong (18) reported that inlet sub-cooling
increases stability but Koshelev (14) pointed out that sub-
cooling up to a critical point does prevent oscillations but
over and above that point, does not stabilize the flow.
Mathison (19) reported that an increase in system pressure
increases the stability of the system. More information on
the phenomena of two-phase flow instability is presented in
Appendix 'B'.

1.6 DESIGN OBJECTIVES

Based on the information presented so far, an
experimental study was initiated to investigate the perform-
ance of a multiple-tube thermosiphon loop with liquid-vapour
separators. Since the primary purpose of the experimental
study was to provide data for comparison with the computer
simulation program results, it was important to be able to
experimentally vary as many parameters as possible. Toward
this end, the test loop shown in Figure 6 was designed which
would permit the study of loop performance as a function of
the charge in the loop, the relative elevation of condenser
with respect to the evaporator, the angle of inclination of
the condenser and of the evaporator tubes and the source and
sink temperatures, for any given working fluid. The design
also permitted two different combinations of evaporator and
condenser tube diameters to be used. The only independent
Fig. 6  Uninsulated multiple tube evaporator and condenser apparatus

A  Vapor header
B1, B2  Evaporator and condenser liquid bypass lines, respectively
C  Vapor line to reservoir
D  Working liquid reservoir
E  Condensate return line
F  Bypass, condensate and evaporator tube flow meters
G  Liquid return lines to the evaporator tubes
H  Evaporator assembly
J  Condenser assembly
K  Liquid line to reservoir (with valve at connection to the loop)
L  Tube wall thermocouple terminals
X  Reservoir vapor line valve
variable which the apparatus could not vary was the length of the evaporator and condenser tubes, which were set equal to 0.61 m (2 ft.). Results from previous work on 1.22 m (4 ft.) long evaporator/condenser loop (8) and the present work were used to verify the computer simulation program's capability to predict the loop performance for an arbitrarily chosen length of an evaporator/condenser loop (1,2).

Water was selected as the source and sink fluid since it had a low thermal resistance. Thus isothermal conditions at the loop tube walls were achieved. This enabled the experimental loop conductance value at a given temperature difference to be used for comparison with the corresponding simulated value.

In order to investigate the flow instabilities in a multi-tube system, the loop was designed with three tubes in parallel in each of the heat exchanger units. The recirculation tube and the loop itself provided two additional flow paths between the evaporator vapour and liquid headers. Three tubes per heat exchanger adequately represented a typical system where the stability problems could be either compounded or compensated by virtue of sharing the flow among them. Flow variations were observed in the flow meters. The data-acquisition system used had forty channels available for temperature measurements.

Complete details of the design of the experimental apparatus are given in Chapter II. The following, however, will provide the reader with a general overview of the loop and its ability to impose a severe test on the
computer simulation program predictions.

Each heat exchanger consisted of three identical tubes interconnected by a common vapour header. Each tube had five thermocouples imbedded in its surface at locations 10, 30, 50, 70 and 90% along its length. Although each tube was individually water jacketted in this study, the three annulii in each heat exchanger were supplied from a common header where the water inlet temperature was measured. The three outlet temperatures were individually measured and then averaged. The temperature difference between inlet and outlet, together with the measured flow rate, were used to calculate the heat transferred in both the evaporator and the condenser. The fifteen surface temperatures for each heat exchanger were individually recorded and subsequently averaged to provide a mean surface temperature. These temperatures were recorded and subsequent calculations were made for the loop conductance by an HP 2100 mini-computer controlled data acquisition system which scanned the temperatures for any specified number of times, then calculated the loop conductances together with their coefficient of variance.

The two heat exchangers differed only in that 1/4-in. (4.83 mm I.D.) copper tubes were used in one, and 3/8 in. (7.90 mm I.D.) tubes were used in the other. The loop assembly permitted either exchanger to be elevated with respect to the other by more than one meter. In addition, the plane of the tubes of each could be independently inclined from the vertical up to 85 degrees.
Rotameters with inlet needle valves were installed to observe flow pulsations and to measure the fluid flow rates through each evaporator tube. The needle valves were used to study the flow instability problems. Two more rotameters were used to measure condensate return and liquid recirculation flow rates. The working fluid temperature was measured in the evaporator outlet and condensate inlet headers and in the mixing liquid header upstream of the evaporator tube flow meters.

The static charge in the loop could be varied between tests by adjusting the elevation of a glass reservoir and noting the static liquid level. When the loop was under test, the reservoir was isolated from the loop by closing the interconnecting valves. The dynamic liquid level in the condensate return line and in the liquid recirculation line could be observed in the interconnecting teflon tubes when their insulation was removed temporarily.

The amount of insulation used was found to be adequate since the experimental tests showed that the difference between the measured heat given up by the source hot water and the measured heat absorbed by the sink cold water was no more than 2% of the heat transferred.

For both the evaporator and the condenser, the three water jacket outlet temperatures were in close agreement, thus indicating that each heat exchanger tube contributed equally to the total energy transfer. This conclusion was further supported by the fact that corresponding tube surface temperatures were also in very close agreement.
The flow rate of the working fluid around the loop was steady, even with the flow control needle valves wide open, for the operating conditions studied and reported. There was, however, some indication that for temperature differences of 5°C or less, the flow rate would be less steady.

For a typical test run, the coefficient of variation for the loop conductance as calculated for 10 sets of data scanned over a period of approximately 10 seconds, rarely exceeded 0.5%. With these measurements, the experimental loop was assumed to have met the design objective of testing the computer simulation program developed under RP 188.
CHAPTER II

APPARATUS AND INSTRUMENTATION

2.1 LOOP ASSEMBLY

The two primary criteria that were to be satisfied by the loop assembly are:

1) Flexibility to vary different parameters such as operating temperatures, elevation, charge, orientation, etc. of the source and sink heat exchangers to provide enough information to test the simulation program, and

2) use of commercial size tubing to duplicate a typical standard single pass, multiple tube air to air heat exchanger coils with inlet and outlet headers for the working fluid.

Item 1) is subdivided as follows:

a) GEOMETRY:
The evaporator and condenser tube assemblies were mounted on different planes which could be elevated and inclined from the vertical to any desired angle independently of one another. To achieve this, flexible teflon tubes of adequate length were used to interconnect the evaporator and the condenser. Also, teflon being inert to R-11 and R-113 refrigerants, which were used as working fluids, could also be used to observe liquid levels by temporarily removing a portion of its insulation.
b) STATIC CHARGE: The static charge in the evaporator tubes could be adjusted to any desired value through the use of a supplementary reservoir which could be isolated during loop operation.

c) FLOW MEASUREMENT: Rotameters were installed in all the liquid lines to measure the flow rates and to observe flow pulsations.

d) OPERATING TEMPERATURE: To maintain isothermal conditions, constant temperature water baths were used. The loop conductance was based on the condenser to evaporator tube surface temperature difference to eliminate all external source-sink effects.

The variation of these parameters coupled with the ability to measure inner surface temperatures along the length of each tube was found to be sufficient to provide a severe test to the simulation program.

The general configuration of the loop was shown in Figure 6 with the insulation removed.

2.1.1 MAIN FLOW LOOP

To duplicate a typical small scale commercial multiple tube heat exchanger application, the main flow loop consisted of 3 tubes in each heat exchanger with interconnecting teflon tubes in the liquid and vapour lines to complete the loop. One set of 3 tubes was made up of 1/4 inch (4.83 mm I.D.) commercial copper tubes, 2 feet (0.61 m) long and the other set was made up of 3/8 inch (7.90 mm I.D.) tubes 2 feet (0.61 m) long. A 3/4" thick x 1' wide x 2½' long wooden panel was used for
mounting each heat exchanger. Either set could be used as an evaporator or a condenser heat exchanger by reversing the three liquid line connections.

2.1.2 SEPARATORS AND HEADERS

Figures 7 and 8 show the liquid header distributor in the bottom of the loop between the flow meters. A screen was installed in this header to ensure thorough mixing and uniform conditions at the entrance to each of the evaporator tube flow meters. The header also had two fittings to accommodate the thermocouple and dial pressure gauge.

Figure 9 shows the vapour header and separator arrangement at the top of evaporator and condenser. Initially, the header itself was to be the liquid vapour separator, but at high charges (50% and above) some liquid was found to back up into the vapour line. So an additional separator was installed. The header liquid return line and the second separator were mounted offset at 45° to the vapour inlet lines so that when the heat exchanger was tilted, they would still be operational.

2.1.3 FLOW METERS

Five identical Skan-Flo Rotameters were used to measure the various working fluid flow rates as shown in Figure 8. In these experiments a black glass float was used in flow meters of the three evaporator tubes and in the recirculation return line. These four flow meters each had a range of 0 to 30 cc per minute (0 to 30 x 10^-6 cu.m/min.). The fifth rotameter, used in the condenser return line, utilized a stainless steel float to give a range of 0 to 75 cc per
FIG. 7 : Lower Mixing Header for Liquid
Fig. 8: LOWER MIXING HEADER AND FLOWMETER ASSEMBLY
minute (0 to 75 x 10^-6 cu.m/min.). These flow meters also had precision needle valves at their inlet to regulate the flow rate if it was necessary to control instabilities.

2.1.4 LOOP ORIENTATION DEVICES

Figure 10 shows the arrangement that was used to provide the desired angle of inclination from the vertical for the plane of each heat exchanger assembly. The protractor was used to set the desired angle, upto 85°, from the vertical.

The vapour and liquid lines were supported using flexible thin ropes from the ceiling so that they would not pinch or sag during testing.

2.1.5 RESERVOIR ARRANGEMENT

The static charge in the loop could be varied with the use of the reservoir arrangement shown in Figure 11. The total volume of the reservoir was approximately equal to the total volumes of evaporator and condenser tubes. So if the reservoir was positioned centrally with respect to the evaporator and if it was filled half full while charging, then by raising or lowering the reservoir, the static charge in the evaporator could be varied between 100 and 0 percent respectively. This reservoir was isolated from the operating loop by a valve near the vapour header and another valve beneath the reservoir.

In actual operation the desired static evaporator charge was set when the loop was not operating by adjusting the elevation of the reservoir with respect to the bottom of the evaporator until the liquid level was at the desired height and then the reservoir was isolated from the loop by closing the
FIG. 10: HEAT EXCHANGER INCLINATION AND SUPPORT ASSEMBLY
FIG. 11: RESERVOIR ASSEMBLY
valves mentioned above, which interconnect the reservoir to the loop.

2.1.6 DIAL PRESSURE GAUGES

In order to check for the presence of non-condensable gases in the loop during operation and to find leaks in the system during assembly, three Marsh Instrument Co. pressure gauges were installed. One was located in the liquid header between the flow meters and the other two were used in the evaporator and condenser vapour headers. These gauges were selected for use with refrigerants R-11, R-113 and had a range of 30"_h (vacuum) to 200 psi (gauge) to an accuracy of 2"_h (vacuum) and 1 psi (gauge) respectively.

2.1.7 SOURCE AND SINK HEAT EXCHANGERS

For ease of thermal control and construction, it was decided to jacket the evaporator and condenser tubes with water flowing in the annular sections. The water flowed upward in each in order that air could be easily purged from the unit. Nominal 3/8" and 1/2" copper tubes were used to jacket the 1/4" and 3/8" inner tubes respectively. The resulting small annular space in each heat exchanger provided turbulent flow conditions for the design mass flow rate of 5 kg/min. of water. This flow rate was chosen to give a temperature difference of not more than 1°C for either of the heat exchangers so that each heat exchanger essentially operated at constant temperature conditions. This decision was made since, in a typical commercial multiple tube air to air heat exchanger, each parallel flow path of the working fluid sees approximately a constant temperature air flow over it. Also any attempt to gain more accuracy
in the calculation of heat flow, Q, resulted in a higher temperature difference in the source/sink water.

The decision to individually water jacket each heat exchanger tube would permit a further experimental study with unequal heating of thermosiphon tubes. For this study, each heat exchanger was supplied with water from a common inlet header. The outlet water temperatures from each of the three tube jackets were measured before they were mixed to return to the respective baths. The average of these three outlet temperatures and one inlet temperature were used to determine the heat transferred. A comparison of the three outlet temperatures was to give an insight into the possible variations in the performance among the three tubes during flow pulsations.

Figure 12 shows the type of custom fittings used to jacket the heat exchanger tubes. The fittings were such that one could easily get to the heat exchanger inner tubes for repairs, if necessary, by simply removing from the jackets from below.

2.1.8 INSULATION

The heat exchangers and all interconnecting tubing were insulated using foam rubber tube insulation with 1/2" wall thickness supplied by Messrs. Armstrong Ltd. The insulation over the condensate return line and recirculation lines could be removed temporarily to observe dynamic liquid levels under operating conditions. Figure 13 shows the tube insulation.

In addition, each heat exchanger assembly was further insulated by wrapping it in sheets of 2" foam insulation.
FIG. 13: Thermosiphon Loop
Showing the Tube Insulation
2.2 INSTRUMENTATION

The instrumentation of the experimental set-up included the temperature measuring devices, hot and cold water baths and the data acquisition system used. Maximum and efficient utilization of the capabilities of available equipment was the basis for all measurements. The temperature measurements were to provide an insight into the different flow conditions of the working fluid within the system.

2.2.1 HOT AND COLD WATER SUPPLY

Distilled water, circulated from a LAUDA constant temperature bath that could be controlled accurately to ±0.01°C was used as a heat source reservoir. A calibrated Brooks rotameter with a scale range of .5 - 10 Kg/min. was used to measure the flow rate. The calibration was checked periodically by collecting a sample over a measured time interval.

A Lab-Line constant temperature controller was used for the heat sink water temperature control. A submersible pump was used for circulation. The sink reservoir flow rate was measured manually before each run by collecting a sample over a measured time interval.

2.2.2 DATA ACQUISITION SYSTEM

A convenient, fast, reliable on-line data analysis method was developed using the Hewlett Packard 2752A Tele-printer, a 2590A coupler-controller, a 2402A integrating Digital Voltmeter and a 2100 computer. The system is shown in Figure 14. Four scanner cards in the coupler-controller, with a capacity of 40 channels, were used to measure and process the temperatures.
FIG. 14: DATA ACQUISITION UNIT

(1) Digital Voltmeter
(2) Computer
(3) Coupler Controller/Scanner Unit
(4) Photo Reader
The coupler-controller was controlled by the computer which was programmed to scan the temperatures for any arbitrarily specified number of times, then calculate and print out various values of the loop conductance and also to print the final set of all temperatures. These sets of data were taken repeatedly at 5 to 10 minute intervals. If no significant change was noted, then steady state was presumed and the test run was considered to be complete. Appendix 'C' describes the program used in this computer and lists a sample output.

2.2.3 THERMOCouple INSTRUMENTATION

The four scanner boards in the coupler-controller of the data acquisition system are capable of scanning a maximum of 40 channels. These were identified as shown in Figure 15.

Each heat exchanger tube had five 36 gauge copper-constantan thermocouples imbedded in its surface at locations 10, 30, 50, 70 and 90% along its length as shown in Figure 16.

In order to ensure accurate readings, grooves were milled circumferentially on the tubes. The ends of two thermocouple wires were peened into a groove 1 to 2 mm apart, then they were silver soldered in place. The surface was then smoothed with a fine emery cloth.

Figure 17 shows the terminal board arrangement for the 5 thermocouples of each heat exchanger tube. The numbers 1 to 5 on each terminal board represent the thermocouples from top to bottom of a vertical heat exchanger.

The working fluid temperature was measured at the mixing header upstream of the evaporator flow meters and at the
FIG. 15: NUMBERING SYSTEM USED TO IDENTIFY THERMOCOUPLES
FIG. 16: THERMOCOUPLE TUBE INSTALLATION
FIG. 17: THERMOCOUPLE TERMINAL BOARD ARRANGEMENT
vapour headers for the evaporator and the condenser tubes. The measurement was done manually using a potentiometer and recorded. It could also be read through the data acquisition system by temporarily changing one of the permanent connections. Figure 18 shows how the thermocouples were mounted for both water and refrigerant temperature measurements. Epoxy type 813 was specially selected for refrigerant usage.

Figure 19 shows the ice bath arrangement used as a reference temperature for all the thermocouples. A paddle wheel was used to circulate the cold bath. A fixed baffle arrangement was designed to create a more complete upward circulation and mixing of the bath. This baffle arrangement solved the initial difficulties with stratification of water and inadequate vertical circulation of water necessary to keep in contact with the layer of ice at the bottom. A STIR-COOL unit with a magnetic stirrer was able to create a fine layer of ice at the bottom of the reservoir. Ice cubes were added before each experiment to make sure that the bath was at 0°C throughout.

2.3 OPERATING PROCEDURES

The following describes the checks that were carried out, as required, in order to charge and operate the system.

2.3.1 LEAKAGE TEST FOR THE WORKING FLUID LOOP

The refrigerant loop was initially pressurized to 4 atmospheres with air and the pressure was observed. If any pressure drop was detected, a soap solution was used to locate the leak. Subsequently, whenever a loss of refrigerant was noted, the leak was located using a Halogen or electronic...
FIG. 18: FITTINGS TO MOUNT THERMOCOUPLES
FIG. 19: THERMOCOUPLE ICE BATH
detector or by using a soap film.

As the saturation pressure of R-113 is around 16" Hg vacuum at room temperatures, vacuum tests were also carried out to ensure that air did not leak into the system. For this test, the system was evacuated to 29" Hg and the pressure observed for a day or two. The vacuum pump arrangement is shown in Figure 20. Two liquid nitrogen-cooled traps were used to condense any refrigerant vapour so that it would not contaminate the vacuum pump oil.

2.3.2 THERMOCOUPLE TESTS

The STIR-COOL unit and the paddle wheel were started at least 3 hours before the test so that a fine layer of ice was formed at the bottom of the reservoir. Crushed ice, made from distilled water, was added to the reservoir. A check was made on the bath temperature with a mercury precision thermometer to ensure 0°C at the thermocouple level.

With the loop in thermal equilibrium with its surroundings, the data acquisition system was used to scan all the thermocouples to ensure uniformity of readings. This also checked out the electrical circuit of the data acquisition system and as well ensured that all components were operating properly.

2.3.3 CHARGING PROCEDURE

A refrigerant charging cylinder was connected to the loop through a 3/8" needle valve and a charging valve at the junction of the condensate return line and flow meter. The cylinder was placed at an elevation of 2 m to provide an additional hydrostatic head for charging since the system was
FIG. 20: SCHEMATIC DIAGRAM OF DEARATION UNIT
close to its saturation pressure. Another charging valve just above the reservoir was used to evacuate the system utilizing the vacuum pump arrangement shown in Figure 20. In addition, cold water was circulated through both heat exchangers so that the loop saturation pressure could be kept at a low value. The system was charged until the liquid occupied approximately 60 percent of the reservoir when it was centrally placed with respect to the evaporator. This procedure permitted 10% of the charge to be used to purge non-condensible gases from the system. This purging operation was done after running the system for 2 hours. Complete degasification of the non-condensible gases was assumed when the loop performance remained unchanged after a purge operation was carried out.

To set a desired percentage static charge of the system (the evaporator flooded under static conditions), the experiment was stopped for 1/2 to 1 hour so that it attained equilibrium. Then the two isolating valves of the reservoir were opened and the reservoir was raised or lowered (with respect to the bottom of the evaporator tubes) to increase or decrease static charge respectively. The height of the liquid column in the evaporator tubes with respect to the total length of evaporator tubes (0.51 m) was used as the percent static charge of the system. Before resuming the experiment, the isolating valves were closed.
CHAPTER III

EXPERIMENTAL PROCEDURE

3.1 START UP PROCEDURE

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

a) The thermocouple reference temperature bath must be switched on at least six hours prior to test runs.

b) Set the hot and cold water temperatures required in the respective thermostats and start the baths at least two hours before test runs.

c) If the loop must be charged, then use the Procedure outlined in Section 2.3.3.

d) If the computer program is already stored in the Data Acquisition System, then go to the starting address. If not, load the final binary tape version of the Fortran Program given in Appendix "C".

e) Warm up the digital voltmeter for at least half an hour and then calibrate it.

f) Switch on the coupler-controller.

3.2 TESTING PROCEDURE

The following steps are to be followed to complete a test run:

a) Set the desired elevation and inclination angle of the evaporator and condenser.
b) Set the required evaporator static charge by using the charging reservoir and then isolate it.

c) Turn on the hot and cold water pumps and measure the respective flow rates.

d) Allow the loop to stabilize for at least 20 minutes.

e) Record the system pressure, barometric pressure, room temperature, hot and cold water flow rates, percent static charge, elevation of condenser, angle of inclination of each heat exchanger, test number, run number and date.

f) Press 'run' on the computer panel, then punch-in the requested data from item e) to the computer using the teletype. Follow the instructions on the teleprinter as received from the computer, to enter these data.

g) The digital voltmeter will indicate the scan of the temperature measurements and then the results will be printed. During this time, record the working fluid flow rates and flow oscillations, if any, as a percentage of a mean value.

h) Follow instructions on the teleprinter to restart, repeat or to continue with new data and for stopping the test run.

3.3 GENERAL DESCRIPTION OF PARAMETER VARIATION

This section describes the general approach used to test the computer simulation program developed under RP 188.
This program, which was a modified version of a single tube thermociphon loop (RP 140), was checked in stages of increasing severity as the experiments progressed.

In the experiments, the following were the different parameters that could be varied:

a) Working fluid = R-11 or R-113

b) Evaporator tube diameter, \( D_e \) = 4.83 (or) 7.90 mm (I.D.)

c) Condenser tube diameter, \( D_c \) = 7.90 (or) 4.83 mm (I.D.)

d) Evaporator inclination, \( \beta_e \) = 0°, 15°, 30°, 45°, 60°, 75°

e) Condenser inclination, \( \beta_c \) = 0°, 15°, 30°, 45°, 60°, 75°

f) Evaporator static charge = 20% to 70%

g) Condenser Elevation, \( z \) = 0 m (bidirectional) = up to 1 m (unidirectional)

h) Temperature Difference, \( \Delta T \) = 5°, 10°, 20°, 30° C

The different stages of the experiments are listed below. A detailed description of parameter variation is given in Appendix 'D':

a) The tests started with a unidirectional loop configuration. With R-11 as working fluid and 4.83 mm (I.D.) as evaporator tube diameter, the effects of charge, angle of inclination of condenser and temperature difference were investigated.

b) The diameter of the evaporator tube was changed to 7.90 mm (I.D.) and the experiments were repeated as in item a).

c) The next step was to investigate the bidirectional loop performance.
d) The working fluid was changed to R-113 and the same tests as above were repeated. The working fluid change to R-113 required 'Property Evaluation Subroutines' to be changed in the computer simulation program.

For the first few test sequences, sufficient data points were obtained to show the behaviour of the system. When it was apparent that the computer program could satisfactorily simulate the experimental results, the subsequent experimental tests were less exhaustive, but covered a wider range. In this way, the simulation program was tested more severely. Care was taken, however, to vary the evaporator static charge sufficiently to cause the operating conditions to range between evaporator and condenser flooding.

3.4 CALCULATION PROCEDURE

The simulation program output listing, among other results, mainly showed the variation of loop conductance with respect to evaporator static charge. This was plotted as a curve. The experimental loop conductance and the corresponding coefficient of variation for ten scans of data (typical) as shown in Appendix 'C' were used to superimpose the experimental loop conductance values on the simulated curve for the same.

3.4.1 LOOP CONDUCTANCE

The loop conductance was 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 condenser. The simulation program loop
conductance was based on Q evaporator, inside surface area of evaporator and the mean surface temperature difference. Such a value in the experiment could be shown as

\[ U_{e,e} = \frac{Q_{\text{evap.}}}{A_{\text{evap.}} \times \Delta T_{\text{surface}}} \]

where \( Q_{\text{evap.}} \) = (Mass flow rate of water in the evaporator annulus) \( \times \)
(Temperature difference across the evaporator) \( \times \)
(Specific heat of water),

\[ A_{\text{evap.}} = 3 \times \pi \times (\text{inside diameter of evaporator tube}) \times \]
(Length of evaporator tube),

and \( \Delta T_{\text{surface}} = \left( \frac{15}{1} T_{\text{evap.surface}} - \frac{30}{16} T_{\text{cond.surface}} \right) / 15 \)

The surface temperature points are shown in Figure 15.

This experimental value of \( U_{e,e} \) was used for comparison with the simulated value. The experimental output also consisted of 'U' values like \( U_{c,c} \), \( U_{a,a} \), etc. These are explained in Appendix 'A' and Appendix 'C' shows one set of such results.

3.4.2 PERCENT EVAPORATOR STATIC CHARGE

The percentage evaporator static charge, frequently called 'charge' in this thesis and references, is the ratio of the height of flooded evaporator column to the total height of evaporator under static, equilibrium conditions. In this experimental loop, the height of the evaporator was 2 feet (0.61 meter). During the experiments, the evaporator static charge
was measured before the start of each run and inputted to the data acquisition system.
CHAPTER IV
EXPERIMENTAL RESULTS AND DISCUSSIONS

4.1 INTRODUCTION

The experimental results of $U_{e,e}$ with the corresponding coefficient of variation were plotted as points in a graph. The simulated results were plotted as a curve. If the simulated curve deviated more than the measured coefficient of variation, the simulation program was modified with different heat transfer equations by Ali (7). The experiment was also repeated, if necessary, until the simulation and the experimental results were well within the allowable deviation.

4.2 RESULTS

Figure 21 shows a comparison between the simulated loop conductance, shown by curves, and the experimental performance, indicated by test points, for a study using R-11 as the working fluid with the system operating in a unidirectional mode. The bottom of the condenser was maintained 0.9 m above the bottom of the evaporator. Several different condenser angles were investigated with the evaporator tubes being vertical. In addition, the system was studied with both heat exchangers in turn acting as the evaporator. This figure shows that recirculation in thermosiphon loops produce flat performance profiles so that variation of static charges would not be too critical on the performance of an actual proto-
Fig. 21: Comparison of Experimental Points with Simulation

Curves for R-11, h = 0.61 m, Elev. = 0.9 m, L = 0 deg.,
T_h = 35 C, T_c = 75 C
type system. It should also be noted that inclining the condenser 45° or more from the vertical produces loop conductance values that are almost double those with a vertical condenser. This happens because the condensate liquid occupies the lower part of the condenser tubes and hence more free area is available for condensation.

Figure 22 shows a comparison similar to that in Figure 21 except that the working fluid was R-113 and the source and sink fluid temperatures were 60°C and 50°C respectively. Not shown in this figure are two additional test points for the curve at the bottom (De = 0.790 cm) at 40% charge with the evaporator inclined at 45 and 75 degrees to the vertical. No significant variation in the loop conductance was noted from the value shown in Figure 22 for a vertical evaporator.

Figure 23 shows the results of tests carried out to investigate the effect of varying the applied temperature difference across a unidirectional loop while maintaining the mean temperature constant at 30°C. The working fluid was R-11 and the small tube heat exchanger was used as the evaporator. Both the evaporator and condenser tubes were vertical. Again, the points represent the experimental results whereas the curves represent the simulated performance.

Figure 24 shows the effect of the temperature difference on the loop conductance with percent static charge in the evaporator as a parameter. The test used R-11 as the working fluid and the smaller tube heat exchanger as
FIG. 22: COMPARISON OF EXPERIMENTAL POINTS WITH SIMULATION CURVES FOR R-13, L., 0.01 m s⁻¹, RELEVANT reflux.
FIG. 23: COMPARISON OF EXPERIMENTAL POINTS WITH SIMULATION CURVES: EFFECT OF TEMPERATURE DIFFERENCE FOR R-11, $D_e = 0.483$ cm, $D_c = 0.790$ cm, $L = 0.61$ m, Elev. = 0.9 m
$L_e = L_c = 0$ deg., $(T_h + T_c)/2 = 300$
FIG. 24: COMPARISON OF EXPERIMENTAL POINTS WITH SIMULATION CURVES: EFFECT OF TEMPERATURE DIFFERENCE FOR R-11.
L = .61 m, \( \theta_e = 0 \) deg., \( \theta = 45 \) deg., Elev. = .9 m
\( (h + \mu) / 2 \) = 450, \( D_e = .483 \) cm, \( D_c = .750 \) cm.
the evaporator with the evaporator vertical and the condenser inclined at 45 degrees.

For the range of variables considered in the tests represented by Figures 21, 22, 23 and 24, it is seen that the simulated curves closely agreed with the experimental loop conductance values (points) for a unidirectional mode of operation. It may also be noted that other than a significant decrease in performance when the condenser tubes are essentially vertical, the loop performance in this mode is relatively insensitive to both charge and to the precise tube inclination angle (two variables which are dependent upon installation conditions) and to the applied temperature difference (an operating condition which may be variable).

Figures 25 and 26 provide a comparison between the experimental performance points and the simulated loop conductance curves for a bidirectional mode of operation with both the evaporator and condenser tubes at the same elevation and inclined at the same angle from the vertical. In these studies, the working fluid was R-11. Figure 25 shows the performance when the small tube heat exchanger is used as the evaporator. Figure 26 shows the corresponding results when the larger tube heat exchanger is used as the evaporator.

The comparison shown in Figure 27 is for a study similar to that of Figure 25 except that the working fluid was R-115.
Fig. 25 Comparison of experimental points with simulation curves: effect of angle of inclination for R-11, \( L = 0.61 \text{ m} \), \( L_e = L_c \), Elev. = 0 m, \( T_R = 50^\circ C \), \( T_C = 40^\circ C \), \( D_c = 0.790 \text{ cm} \), \( D_e = 0.483 \text{ cm} \).
Fig. 26 Comparison of experimental points with simulation curves: effect of angle of inclination for R=11, L = .61 m, L_e = L_c, Elev. = 0 m, T_{D_0} = 50°C, T_c = 40°C, D_e = .790 cm, D_c = .493 cm
Fig. 27 Comparison of experimental points with simulation curves: effect of angle of inclination for R-113, \( L = .61 \text{ m} \), \( \epsilon = .9 \), Elev. = 0 m, \( T_h = 60^\circ \text{C} \), \( T_c = 50^\circ \text{C} \), \( D_e = .483 \) cm, \( D_c = 0.790 \) cm.
Figure 28 shows the behaviour of a bidirectional loop (inclined at 75° from the vertical) with respect to applied temperature difference and R-113 as working fluid. The behaviour is similar to that shown in Figures 25, 26, and 27. Figure 28 also shows the excellent reproducibility of experimental results for $\Delta T = 10^\circ$C over a period of 3 months.

Once again, for the range of variables studied in tests represented by Figures 25, 26, 27 and 28, the simulated curves closely matched the experimental data points for a bidirectional loop.

At low applied temperature differences of $5^\circ$C and below, the bidirectional system exhibited flow pulsations of the order of $\pm 5\%$ from the mean flow rate at low evaporator static charges. However, it was possible to reduce such oscillations by introducing inlet restriction to individual evaporator tubes, by partially closing their respective flow meter valves.

A further check of the validity of the simulation program was made by comparing the simulated temperature and flow rates of working fluid within the loop, to the experimentally measured values. Table I shows such a comparison for four representative simulations which closely match the experimental evaporator static charges. Two are bidirectional and two are unidirectional. One run for each mode of operation used R-11 and the other R-113. In addition, different combinations of angles of inclination and charges were used for each test. One of the tests utilized the larger tube heat
Fig. 28 Effect of temperature difference for R-11. Comparison of experimental points with simulation. $L = .61 \text{ m}$, Elev. = .0 m, $L_c = L_e = 75 \text{ deg}$. $T_{\text{mean}} = 45^\circ \text{C}$, $D_e = 0.483 \text{ cm}$, $D_c = 0.790 \text{ cm}$
TABLE 1
Comparison of Experimental and Simulated Tests

<table>
<thead>
<tr>
<th>Run No.</th>
<th>TYPE</th>
<th>EVAPORATOR TUBE</th>
<th>Cond Temp + (°C)</th>
<th>TEMPERATURES OF Source</th>
<th>TEMPERATURES OF Sink</th>
<th>HEAT FLOW Source</th>
<th>HEAT FLOW Sink</th>
<th>LOOP Cond Cond ( \text{UA} ) (kW/m²K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>E</td>
<td>50</td>
<td>40.70</td>
<td>50.11</td>
<td>49.49</td>
<td>40.24</td>
<td>40.80</td>
<td>216.6</td>
</tr>
<tr>
<td>S</td>
<td>49.6</td>
<td>.0024</td>
<td>48.70</td>
<td>50.11</td>
<td>49.48</td>
<td>40.32</td>
<td>40.91</td>
<td>216.3</td>
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<tr>
<td>E</td>
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<td>.0015</td>
<td>48.70</td>
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<td>51.43</td>
<td>50.25</td>
<td>207.5</td>
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<td>.0068</td>
<td>58.70</td>
<td>51.43</td>
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<td>51.76</td>
<td>51.07</td>
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</tr>
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<td>.0014</td>
<td>58.90</td>
<td>51.43</td>
<td>50.25</td>
<td>51.24</td>
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</tr>
<tr>
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<td>.0018</td>
<td>58.45</td>
<td>51.43</td>
<td>50.25</td>
<td>51.24</td>
<td>51.07</td>
<td>268.1</td>
</tr>
<tr>
<td>E</td>
<td>60.0</td>
<td>.0057</td>
<td>48.50</td>
<td>51.45</td>
<td>50.25</td>
<td>51.24</td>
<td>51.07</td>
<td>265.6</td>
</tr>
<tr>
<td>S</td>
<td>62.4</td>
<td>.020</td>
<td>48.60</td>
<td>48.60</td>
<td>48.00</td>
<td>48.00</td>
<td>48.00</td>
<td>260.0</td>
</tr>
</tbody>
</table>

E - Experimental Results, S - Simulation Results,

1 - R-11, \( \phi_c = \phi_e = 15 \text{ deg.} \), Elevation = 0 m, \( D_e = 0.483 \text{ cm}, D_c = 0.790 \text{ cm} \)
2 - R-113, \( \phi_c = \phi_e = 45 \text{ deg.} \), Elevation = 0 m, \( D_e = 0.483 \text{ cm}, D_c = 0.790 \text{ cm} \)
3 - R-113, \( \phi_e = 0, \phi_c = 45 \text{ deg.} \), Elevation = 0.9 m, \( D_e = 0.483 \text{ cm}, D_c = 0.790 \text{ cm} \)
4 - R-11, \( \phi_e = 0, \phi_c = 75 \text{ deg.} \), Elevation = 0.9 m, \( D_e = 0.790 \text{ cm}, D_c = 0.483 \text{ cm} \)

* Average Value of 15 readings
exchanger as the evaporator.

As may be seen, the program simulated data very close to the experimentally measured values for all of these diverse test conditions.

4.3 DISCUSSIONS

The system behaviour discussed here is based on the successful experimental results and further computer simulations carried out (Ref. 1, 2 and 3) for other geometries and operating conditions (not done experimentally) which would yield flow conditions similar to those studied experimentally. Reference 9 is also used here for comparison to a non-recirculation thermosiphon loop.

For a unidirectional recirculating thermosiphon loop, as shown in Figure 24, the larger charges have the flatest response to the temperature difference, however, this is accompanied by a slightly reduced peak performance at certain $\Delta T$'s. The shape of these curves is dictated by low heat transfer co-efficients due to suppressed boiling for low temperature differences and by evaporator tube dryout for larger temperature differences.

In order to study the effect of the separator on the loop performance, the limitations of a non-recirculation thermosiphon loop are dealt with first (Ref. 9).

Without a separator, the transition from liquid carryover region to the dryout region causes the slope of the curve (similar to plot as shown in Figure 24) to suddenly change. Also, for temperature differences greater or less than the maximum performance temperature difference, the system suffers
from dryout in the evaporator or liquid carryover to the condenser, respectively. Peak performance for a no-separator loop occurs when there is negligible dryout and negligible liquid carryover.

The difference between these two systems is most noticeable when dealing with a 50% charge curve as shown in Figure 24. For this separator case, the loop conductance continues to rise gently from approximately 5°C temperature difference, then fall to a temperature difference of approximately 30°C. This is due to dryout being suppressed due to greater rate of recirculation through the evaporator tubes. Thus the separator permits the system to operate, not only with low temperature differences without liquid carryover, but also with much higher temperature differences without dryout because of the significantly higher evaporator flow rates (consequent high two phase heat transfer co-efficients) with a liquid recirculation system.

A study of Figures 21 and 22 reveals the effect of evaporator tube diameter on the loop conductance. For identical orientations of the loop, for each diameter of evaporator tube, there exists a charge which yields maximum loop conductance. Reference 3, using further simulations, shows that for a given system with a large condenser surface area (so that only evaporator resistance effectively controls the performance of the loop), satisfactory performance may be expected over a modest range of evaporator tube sizes (since maximum loop conductance value is only mildly dependent on the evaporator
tube diameter) provided the correct charge is used.

Figures 25, 26 and 27 illustrate the characteristic behaviour of bidirectional thermosiphon loops. Unlike unidirectional loops, in order to maximize loop conductance, it is important that the charge be set within reasonably close tolerances because dryout occurs if the charge is too low and the condenser performance is severely impaired due to flooding if the charge is too high. Inclining the loop from the vertical causes two effects. The maximum performance increases with angle of inclination up to approximately 45 deg. then starts to fall. It should be noted, however, that the performance with inclined tubes is, in all cases, much better than that for the vertical tubes. In addition, the performance becomes more sensitive to charge as the angle of inclination increases.

For all configurations shown in Figures 25, 27, and 28 (smaller tube evaporator), the maximum conductance occurred with an evaporator static charge between 44 - 50%. This is a result of the evaporator slightly outperforming the condenser. This also explains why the maximum performance decreases in magnitude and shifts to a still lower charge when the smaller tubes are used in the condenser as shown in Figure 26.

Figure 28 shows that for an inclined bidirectional loop, there is little difference in performance with respect to charge for small temperature differences.

Further discussions on the computer simulation results of bidirectional and unidirectional loops are available
in Ref. 1 and 2 respectively.

Figure 29 reproduced from Ref. 1 shows a typical simulated comparison of unidirectional and bidirectional loop performance and illustrates the effect of increased sensitivity to loop charge due to the inherent condenser flooding and the decrease in the driving potential and the consequent drop in loop conductance for a bidirectional loop.

The flow oscillations ± 5% from mean flow rate observed in the inclined bidirectional mode of operation at low temperature differences (below 5°C) and at low charges, are attributed to the following:

a) Low flow rate in the system due to low temperature difference,

b) Smaller tube evaporator outperforming larger tube condenser and consequent higher dynamic liquid head in the evaporator than the condenser,

c) Liquid sub-cooling exceeding the critical limit (for stable flow) due to low flow rates (more retention time for liquid in the return circuit), and

d) Weak driving potential due to bidirectional mode.

In inclined bidirectional loop operation, flow oscillations can be reduced by introducing flow restrictions at inlet to the evaporator tubes. Vertical bidirectional loop operation has the inherent characteristic to self-limit the flow oscillations since the buoyancy of the vapour aids upward flow in the vertical evaporator. Also, multiple tube evaporator
Figure 29  Comparison of Uni- and Bidirectional Thermosiphon Loop Performance

$L = .61m  \quad \angle e = \angle C = 45^\circ  \quad T_m = 30^C  \quad \Delta T = 8^C  \quad D_e = D_c = .483$
configuration has the built in capability to share the flow among the parallel paths for small pulsations unlike a single tube evaporator system.
CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

For a multiple tube, two phase, recirculation
thermosiphon loop, the experimental results were in a very
close agreement with the simulated values for all diverse
test conditions. Further simulations carried out on the
basis of this success and reported in References 1, 2 and
3 showed a recirculation loop to have a higher loop
conductance and significantly reduced sensitivity to
operating conditions (the source and sink temperature) than
the non-recirculation loop studied in RP 140 (Ref. 9).

Recirculation in the evaporator essentially con-
verts it into a thermosiphon reboiler. The resulting high
flow rate through the evaporator tubes yields a very high
two phase heat transfer co-efficient for a wide range of
temperature differences, without dryout of the evaporator
even at high temperature differences.

The liquid/vapour separator at the evaporator
outlet allows the condenser to operate with a high-condens-
ation co-efficient over a wide-range of temperature differ-
ences, by preventing liquid carryover to the condenser even
at low temperature differences.

For a unidirectional loop, the condenser tubes
should be mounted so that they are inclined to the vertical
by at least 15 degrees to take advantage of the significant
improvement this provides in the condensation co-efficient.
The effect of diameter of the evaporator tube, based on Reference 3 and the two different sizes used in experiments indicates that satisfactory performance may be obtained over a modest range of diameters provided correct charge is used, since maximum loop conductance is only mildly dependent on evaporator tube diameter.

Inclination of the evaporator tubes from the vertical does not become an important factor for the small temperature differences (10°C and below) and for short tubes (0.61 m) used in experiments, since the hydrostatic head variation along the tube did not result in saturation temperature variation in excess of the applied temperature difference.

Flow oscillations observed in multiple tube evaporator, bidirectional loop operation at low temperature differences should not pose a problem in real systems. During experiments, it was possible to reduce such oscillations by introducing inlet restrictions at the entrance to individual evaporator tubes.

The two variables which have the greatest effect on the conductance of the bidirectional loop are: the angle of inclination of the tubes and the charge. For small temperature differences dealt with in the experiments, there was little difference in performance for 45° and 75° angles of inclination, however, their performance was approximately double that for vertical tube case. The more rapid decrease in performance with charge for the bidirectional loop as compared to the unidirectional loop is due to the inherent
condenser flooding which occurs in the former. Hence the bidirectional system, even though reasonably less sensitive to temperature differences, was found to yield a poorer loop conductance than the corresponding unidirectional system.

The loop conductance for a vertical bidirectional loop, based on Reference 1 and the two diameters used in experiments, is found to be relatively insensitive to diameter at moderate temperature differences (10°C typical). The small decrease in loop conductance at high charges is attributed to reduction in convection coefficients due to low linear velocities with larger evaporator tubes and condenser flooding.

Finally, through a careful selection of the loop charge as well as the coil tube diameter and length, multiple tube thermosiphon loops can provide very good loop conductance values over a wide range of temperature differences. The range is limited by evaporator tube dryout for large temperature differences and by boiling suppression for small temperature differences.

5.2 RECOMMENDATIONS

An experimental flow visualization study on a large scale multiple tube thermosiphon reboiler evaporator system should be carried out in order to provide an insight into the factors which cause and suppress instabilities and flow pulsations in the tubes.

Full scale experimental tests on a recirculation, two phase, coil loop thermosiphon heat exchanger should be carried out, preferably with commercial heat exchangers.
A further full scale experimental study on the use of thermosiphon loop as a solar heat recovery device is suggested.
REFERENCES


APPENDIX A

A.1 PERFORMANCE RATING

The ability of a thermosiphon loop heat exchanger to transfer energy from one region to another may be expressed in terms of the ratio of the heat transfer rate per unit tube surface area \( Q/A \), to the temperature difference \( \Delta T \) between the two regions, a rating which will be referred to as the loop conductance. In order to eliminate external influences from the rating such as fin tube configuration, exterior gas properties and gas flow rates, contamination, etc., \( \Delta T \) is selected to be the difference between the evaporator tube inner surface average temperature and the condenser tube inner surface average temperature. \( Q \) is chosen to be:

- \( Q_e \) if \( Q_{\text{evaporator}} \) is used (or)
- \( Q_c \) if \( Q_{\text{condenser}} \) is used (or)
- \( Q_a \) if \( Q_{\text{avg.}} \) of evaporator and condenser is used.

(Note: \( Q_a \) allows for any heat loss or gains associated with the interconnecting piping.)

Area 'A' is set to be

- \( A_e \) if inner tube surface area of evaporator is used (or)
- \( A_c \) if inner tube surface area of condenser is used (or)
- \( A_a \) if average inner tube surface area of condenser and evaporator are used.

(Note: \( A_a \) is useful when the diameters of evaporator and condenser tubes are different)

Then \( U_{e,c} = \frac{Q_e}{A_e \cdot \Delta T} \)
\[ U_{a,a} = \frac{Q_a}{A_a \cdot \Delta T} \]

and \[ U_{c,c} = \frac{Q_c}{A_c \cdot \Delta T} \]

This experimental \( U_{e,e} \) is used to compare with the simulated result. The experimental results also consist of \( U \)'s based on other combinations of \( Q \)'s and \( A \)'s as defined below:

\[ U_{e,a} = \frac{Q_e}{A_a \cdot \Delta T} \]

\[ U_{a,e} = \frac{Q_a}{A_e \cdot \Delta T} \]

\[ U_{a,c} = \frac{Q_a}{A_c \cdot \Delta T} \]

and

\[ U_{c,a} = \frac{Q_c}{A_a \cdot \Delta T} \]

A.2 FACTORS AFFECTING PERFORMANCE

A brief generalized approach to analyze the performance of a thermosiphon loop heat exchanger system is given here. Let \( T_{cf} \) and \( T_{hf} \) be the temperatures of cold (SINK) fluid and hot (SOURCE) fluid respectively. Let \( T_c \) and \( T_e \) be the average inner surface temperatures of condenser and evaporator tubes respectively. For such assumptions, Figure A1 symbolically illustrates the thermal resistances which contribute to the overall thermal resistance, \( R_{HF} \), of a thermosiphon heat exchanger. If the thermal resistances between the source and sink fluids and the evaporator and condenser tube inner surfaces \( R_{HF} \) and \( R_{CF} \) respectively, are eliminated from consideration, we are left with the resistance for the loop itself, \( R_L \). The loop resistance, with which this thesis is concerned, can be sub-divided into three parts: the evaporator resistance, \((1/hA)e\), the transport
HEAT EXCHANGER CONDUCTANCE \((UA)_{HE} = \frac{1}{R_{HE}}\)

THERMOSIPHON LOOP CONDUCTANCE \((UA)_{L} = R_{L}^{-1}\)

FIGURE A1: ELECTRICAL ANALOG OF IDEALIZED THERMOSIPHON LOOP
resistance, $R_s$, and the condenser resistance, $(1/hA)c$. An understanding of how the loop behaves may be obtained by considering the factors which affect each of these resistances in turn.

The transport resistance arises as a result of the saturation temperature difference between the evaporator and condenser which is necessary to produce the saturation pressure difference which in turn is necessary to cause the circulation of the working fluid. This temperature drop can be reduced through the selection of a working fluid which has a saturation pressure vs saturation temperature curve with a very high slope, a large latent heat of vaporization and small specific volume of the saturated vapor. The high latent heat will require a lower mass flow rate for a given heat flow rate and the small specific volume will reduce the volume to be transported and hence will reduce the required pressure drop. A high saturation pressure vs saturation temperature curve slope will permit a large pressure drop to be sustained by a relatively small temperature difference. The pressure drop itself may be reduced by minimizing the quantity of liquid present in the vapor line and by ensuring that any liquid present is carried in a separated flow mode. The selection of a large diameter vapor pipe to reduce the flow resistance must be governed by the economics of balancing the increased initial cost and increased heat loss against reduced pressure drop and hence reduced temperature drop between the evaporator and the condenser coils.
The evaporator resistance is dependent on the properties of the working fluid and on the proportion of the surface area subject to single phase liquid forced convection, subcooled boiling, nucleate bulk boiling, two-phase forced convection and finally to single phase vapor forced convection. This resistance is most easily minimized by avoiding single phase convection as much as possible, particularly dryout, through proper design of the system. In general, the higher the proportion of two-phase convection, the lower will be the evaporator resistance. A working fluid with a high slope on the saturation pressure-saturation temperature curve would allow boiling to start closer to the bottom of the evaporator, since the saturation temperature would not be significantly increased in the bottom of the evaporator by hydrostatic pressure. In addition, a high value of thermal conductivity is desirable.

The condenser resistance can be reduced by minimizing the condensate film thickness. This can be achieved by ensuring no liquid carryover from the evaporator and by inclining the tubes so that the condensate film within the tube becomes assymetric. The thicker film on the bottom of the tube slightly retards the heat transfer, however the thinner film on the upper portion of the tube greatly enhances it thus yielding a net improvement in the condenser performance.

All three of these resistances are affected by the impressed temperature difference between the exhaust and supply ducts. It is highly desirable that the loop performance be as insensitive as possible to this temperature difference since the designer has no direct control over the operating conditions.
APPENDIX B

TWO PHASE FLOW INSTABILITY

B.1 INTRODUCTION

Flow instabilities are undesirable in boiling, condensing and other two-phase flow devices for several reasons. First, sustained flow oscillations may cause forced mechanical vibration of components. Second, flow oscillations may cause system control problems. Third, flow oscillations affect the local heat transfer characteristics and may induce boiling crisis (critical heat flux, burnout). The critical heat flux was found by Ruddick [10] to be reduced 40% when the flow was oscillating. Flow instability becomes of particular importance in water-cooled nuclear reactors (as well as in water-moderated nuclear reactors), steam generators and other types of heat exchangers like thermosiphon loops. The designer's job is to predict the threshold of flow instability so that he can design around it or compensate for it.

There are areas of confusion regarding the different types of flow instability, which is perhaps understandable since an instability which can start in a heated channel or in a two-phase flow system, can be a primary or secondary phenomenon, and it can be static or dynamic in nature. Since the behaviour and method of prediction of one type is different from another, the investigator must recognize the type of flow instability on hand before he can perform an analysis.

The purpose of this chapter is to categorize the types of flow instability, consolidate the basic understanding
and especially to consider the instabilities in natural circulation loops.

Instability in a very broad sense could be aperiodic (decaying or diverging transient) or periodic (oscillating).

The conditions to be satisfied for instabilities are:

1. For a given set of 'external parameters', the system could exist in more than one state.
2. An external energy source is required to account for frictional 'dissipation'.
3. Disturbances that can initiate the oscillations must be present.

All these conditions are satisfied for a two-phase flow with heat addition. If we compare these oscillations to a mechanical system, the

mass flow rate would be analogous to mass,
pressure drop would be analogous to existing force, and
void would be analogous to spring, respectively.

Therefore the relationship between mass flow rate and pressure-drop plays an important role. A purely hydrodynamic instability can occur because a change in flow pattern (and flow rate) is possible without a change in pressure-drop. The situation becomes complicated when there is thermohydrodynamic coupling of heat transfer, voids, flow pattern and flow rate. Even when heat source is held constant, flow oscillations could occur.

B.2 CLASSIFICATION OF FLOW INSTABILITIES

A flow is subject to static instability if, when the flow conditions change by a small step from the original steady-
state ones, another steady state is not possible in the vicinity of the original one. The cause of this phenomenon lies in the steady-state laws; hence, threshold of the instability can be predicted only by using steady-state laws. A static instability can lead either to a different steady-state condition or to a periodic behaviour.

A flow is subject to dynamic instability when the inertia and other feedback effects have an essential part in the process. The system behaves like a servo-mechanism and the knowledge of the steady-state laws is not sufficient, even for the threshold prediction. The steady-state may be a solution of the equations of the system, but it is not the only solution. The above mentioned fluctuations in a steady flow may be sufficient to start the instability.

The classification of various flow instabilities mostly dealing with constant heat-flux cases is shown in Table B1(11). The following parameters were considered:

Geometry - channel length, size, inlet and exit restrictions, single or multiple channels.

Operation Conditions - pressure, inlet subcooling, mass velocity, power input, forced or natural convection.

Boundary Conditions - axial heat flux distribution, pressure drop across channels.

The effects of geometry and boundary conditions are usually interrelated, such as in the case of flow redistribution among parallel channels. With common headers connected to the parallel channels, the flow distribution among channels
<table>
<thead>
<tr>
<th>Class</th>
<th>Type</th>
<th>Mechanism</th>
<th>Characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Static Instabilities</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.1 Fundamental</td>
<td>Flow excursion or</td>
<td>$-\frac{\Delta P_{int}}{\Delta P_{ext}} \leq 3$</td>
<td>Flow undergoes sudden, large amplitude excursion to a new, stable operating</td>
</tr>
<tr>
<td>(or pure)</td>
<td>Ledinegg instability</td>
<td></td>
<td>condition</td>
</tr>
<tr>
<td>Static</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Instabilities</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.2 Fundamental</td>
<td>Flow pattern transition</td>
<td>Bubbly flow has less</td>
<td>Cyclic flow pattern transitions and flow rate variations</td>
</tr>
<tr>
<td>relaxation instability</td>
<td>instability</td>
<td>void but higher $\Delta P$</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>than that of annular flow</td>
<td></td>
</tr>
<tr>
<td>1.3 Compound</td>
<td>Bumping, geysering,</td>
<td>Periodic adjustment of metastable</td>
<td>Period process of superheat &amp; violent evaporation with possible expulsion</td>
</tr>
<tr>
<td>relaxation instability</td>
<td>and chugging</td>
<td>condition, usually due to lack of</td>
<td>and refilling</td>
</tr>
<tr>
<td></td>
<td></td>
<td>nucleation sites</td>
<td></td>
</tr>
<tr>
<td>2 Dynamic Instabilities</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.1 Fundamental</td>
<td>Acoustic oscillations</td>
<td>Resonance of pressure waves</td>
<td>High frequencies (10-100 Hz) related to time required for pressure wave</td>
</tr>
<tr>
<td>(or pure)</td>
<td></td>
<td></td>
<td>propagation in system</td>
</tr>
<tr>
<td>Dynamic</td>
<td>Density wave oscillations</td>
<td>Delay and feedback effects in relationship</td>
<td>Low frequencies (1 Hz) related to transit time of a continuity wave</td>
</tr>
<tr>
<td>Instabilities</td>
<td></td>
<td>between flow rate, density, and pressure</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>drop</td>
<td></td>
</tr>
<tr>
<td>2.2 Compound</td>
<td>Thermal oscillations</td>
<td>Interaction of variable heat transfer</td>
<td>Occurs in film boiling</td>
</tr>
<tr>
<td>dynamic</td>
<td></td>
<td>coefficient with flow dynamics</td>
<td></td>
</tr>
<tr>
<td>Instabilities</td>
<td>BWR instability</td>
<td>Interaction of void reactivity coupling</td>
<td>Strong only for a small fuel time constant and under low pressures</td>
</tr>
<tr>
<td></td>
<td></td>
<td>with flow dynamics and heat transfer</td>
<td></td>
</tr>
<tr>
<td>2.3 Compound</td>
<td>Parallel channel</td>
<td>Interaction among small number of</td>
<td>Various modes of flow redistribution</td>
</tr>
<tr>
<td>dynamic</td>
<td>instability</td>
<td>parallel channels</td>
<td></td>
</tr>
<tr>
<td>Instability</td>
<td>Pressure drop oscillations</td>
<td>Flow excursion initiates dynamic</td>
<td>Very low frequency periodic process (0.1 Hz)</td>
</tr>
<tr>
<td>as secondary phenomena</td>
<td></td>
<td>interaction between channel and</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>compressible volume</td>
<td></td>
</tr>
</tbody>
</table>

**TABLE II: CLASSIFICATION OF FLOW INSTABILITY**
is determined by dynamic pressure variations in individual channels which are caused by flow instability in each channel. Thus, the boundary conditions of a heated channel separate the channel instability from system instability.

B.3 DYNAMIC INSTABILITY

Considering only the periodic nature of oscillations, it is customary to distinguish between two types of oscillations, depending upon whether the oscillation occurs in one of the parallel channels or in an entire loop.

B.3.1 PARALLEL CHANNEL OSCILLATIONS

When a large number of flow channels having common inlet and outlet plenums operate hydraulically in parallel, all have the same pressure drop. An instability in one or a few of the channels where boiling occurs does not significantly change the overall pressure drop because the flow is redistributed to a large number of other stable channels and the condition of constant imposed pressure drop is satisfied. A temporary reduction in the inlet flow rate to a boiling channel will increase the rate of evaporation, thereby raising the average void fraction. This disturbance affects the elevation, acceleration and frictional pressure drop as well as the heat transfer behaviour. For certain conditions of channel geometry, thermal properties of the heated wall, flow rate, inlet enthalpy, heat flux, etc., resonance may occur and sustained oscillations then result.

Parallel channel interactions were reported by Gouse and Andrysia (12) for a two channel Freon-113 system. The
flow oscillations, as observed through the electrically heated (constant heat flux case) glass tubing, were generally 180 degrees out of phase. Well within the stable region, the test sections began to oscillate in phase with very large amplitude flow oscillations. This behaviour was not observed when three heated channels were operated in parallel with a large by-pass so as to maintain a constant pressure drop across the heater channels (13).

Koschelev et al (14) reported the tested results of a non-recirculating bank of three heated tubes in parallel. The phase shifts of flow oscillations were quite different for various tubes. Sometimes the flow oscillations in two tubes were in phase while the flow oscillation in the third tube was in a phase shift of 120 to 180 degrees. The amplitudes of the in-phase oscillations were different, i.e. high in one and negligible in the other. Sometimes phase shift between individual tubes took place without apparent reason, but there were always tubes in which the flow oscillations were 120 degrees or 180 degrees out of phase.

B.3.2 NATURAL CIRCULATION LOOP OSCILLATIONS

Flow oscillation in a natural circulation loop is characterised by small flow rates and by the elevation pressure drop being the dominant fraction of the channel pressure drop.

In a natural circulation loop, the driving pressure drop is directly related to the void in the boiling channel. Thus, a temporary reduction of the inlet flow leads to an increase in the local void fraction, tending to accelerate the exit flow. The increase in pressure drop may offset the increase in available driving elevation head due to the void increase and
a further reduction of flow rate occurs with a consequent further increase in voids. This temporary diverging instability is finally terminated by the sudden expulsion of a large part of the remaining liquid phase in the channel. The pressure drop before this may have increased sufficiently to stop or even reverse the flow temporarily. In the wake of the following sudden pressure release, liquid rushes into the channel, evaporation rapidly takes place there, and the cycle is repeated. Quandt (15) analysed the phenomenon, using the Laplace transformation to solve the linearized hydrodynamic and thermodynamic equations obtained from perturbation technique. Jones (16) used a similar approach to obtain a function which indicates the degree of stability on a Nyquist locus. Wallis (17) used the Lagrangian concept to predict the stability of a natural circulation loop. He also experimentally proved the beneficial effects of the inlet and the ill effects of outlet restrictions on the stability of a natural circulation loop with constant temperature boundary conditions. This has been the only reference that is directly applicable to the thermosiphon loop experiments considered in this thesis work.

B.4 EFFECTS OF VARIOUS PARAMETERS ON FLOW INSTABILITY

A summary of effects of various parameters on flow instability, which have been observed in various studies mentioned (11, 18) are listed below:

1. High heat input aggravates flow instability because of the high rate of evaporation.
2. Increased inlet subcooling, within a critical value, stabilizes the flow. Beyond that value, a further increase in subcooling may reverse the trend and de-stabilize the flow, because there is a maximum rate of change in the pressure drop at the critical inlet subcooling.

3. Increased resistance at the exit strongly decreases the stability because it increases the exit pressure drop when a large void is generated in the channel.

4. Increased resistance at the inlet strongly increases the stability. When the flow is slowed by a disturbance, a large pressure head will be retained at the inlet and will be available for forcing the fluid through the channel, thereby stabilizing the flow.

5. Increased system pressure increases the stability because the difference in specific volumes of vapour and liquid then becomes smaller.

6. Increased static head in a long vertical channel increases the stability of upward boiling flow and decreases the stability of downward boiling flow. This occurs because the buoyancy of the vapour aids upward flow and opposes downward flow. With the latter, there is a tendency for the vapour to agglomerate and remain in the channel.
PROGRAM GDE
DIMENSION TSUM(12), TMP(40), TMS(12)

WRITE(2,1)
1 FORMAT("INPUT MONTH, DAY, TEST NO."/"FORMAT(A4,3,1,4)")
READ(1,3) MONT, IDATE, ITEST
3 FORMAT(A2,A2,13,14)
WRITE(2,60)
60 FORMAT("INPUT DET, DCT, EL, CL"/
       "FORMAT(F7.5)")
READ(1,603) DET, DCT, EL, CL
603 FORMAT(F7.5,F7.5,F7.5,F7.5)

WRITE(2,42)
42 FORMAT("INPUT EVANG, CNANG, ELEV, HFLOW, CFLOW, CHRG"/
        "/"FORMAT(I3,13,F7.4,F6.4,F6.4,F5.1)")
IRUN=0
READ(1,4) EVANG, CNANG, ELEV, HFLOW, CFLO, CHRG
4 FORMAT(I3,13,F7.4,F6.4,F6.4,F5.1)

WRITE(2,43)
READ(1,44) DSETS, MM
IDUN=11B
CALL 'L66(2, IDUN)
CALL STATS(IDUN)
IRUN=IRUN+1
WRITE(IDUN,301)
301 FORMAT("02C0")
CALL STATS(IDUN)
44 FORMAT(F5.0,I1)
WRITE(IDUN,302)
302 FORMAT("02I07C0")
CALL STATS(IDUN)
43 FORMAT("INPUT DATA SETS, MM"/
        "FORMAT(F5,0, I1)")
READ(IDUN,303) ISHR, ISMIN, ISSEC, ISMSC
303 FORMAT(I4,I2,I2,I2)
CALL STATS(IDUN)
VCUN=0.0
VAUN=0.0
VEUN=0.0
DO 100 I=1,12
100 TSUM(I)=0.0
ZD=1.0
210 TEST=0.0
TCS=0.0
CP=4186.8
K=1
DO 200 JJ=1,40
GOTO(400,401,402,403,220)K
400 WRITE(IDUN,404)
404 FORMAT("9210300")
GOTO 66
401 WRITE(IDUN,405)
405 FORMAT("9210400")
GOTO 66
402 WRITE(IDUN,406)
406 FORMAT("9210500")
GO TO 66
403 WRITE(1DVN, 407)
407 FORMAT("#2I600")
66 CALL STATS(IDVN)
WRITE(1DVN, 408)
408 FORMAT("#1E2I100")
CALL STATS(IDVN)
READ(IDVN, 306) DATA
306 FORMAT(2X, F10.0)
CALL STATS(IDVN)
TMP(JJ) = (46638.749761 DATA - 1321251.435 * 
8 DATA * 2 + 59189465.8 DATA * 3) * 5.0 / 9.0
GOTO(410, 411, 412, 413, 200) K
410 WRITE(1DVN, 420)
420 FORMAT("#2I3D0")
GOTO 77
411 WRITE(1DVN, 414)
414 FORMAT("#2I4D0")
GOTO 77
412 WRITE(1DVN, 415)
415 FORMAT("#2I5D0")
GOTO 77
413 WRITE(1DVN, 416)
416 FORMAT("#2I6D0")
77 CALL STATS(IDVN)
200 K = (JJ / 10) + 1
471 IF(ZN=DSET5) 225, 220
220 WRITE(1DVN, 312)
312 FORMAT("#2I700")
CALL STATS(IDVN)
READ(IDVN, 313) IFHR, IFMIN, IFSEC, IFMSC
313 FORMAT(14, 12, 12, 12)
CALL STATS(IDVN)
WRITE(2, 450) (TMP(M), M = 33, 38)
DO 451 N = 1, 5
451 WRITE(2.5) TMP(N), TMP(N + 5), TMP(N + 10), TMP(N + 15), TMP(N + 20), TMP(N + 25)
5 FORMAT(6(5X, F6.2))
450 FORMAT(6(5X, F6.2)/)
WRITE(2, 452) TMP(31), TMP(32)
452 FORMAT(14X, F6.2, 27X, F6.2)/
225 DC 50 I = 1, 15
TCS = TCS + TMP(I + 15)
50 TES = TES + TMP(I)
TMP(29) = TES / 15.0
TMP(30) = TCS / 15.0
DO 300 I = 1, 12
TMP(I) = TMP(I + 29)
TSM(I) = TSM(I) + TMP(I)
300 TMS(I) = TSM(I) / ZN:
IF(ZN < 0) GOTO 321
800 CD = ( (TMP(5) + TMP(9) + TMP(12)) / 3.0 ) - TMP(4)
HD = TMP(3) - ( (TMP(5) + TMP(6) + TMP(7)) / 3.0 )
GOTO 503

301 CD=TMP(12)
    HD=TMP(11)
303 QC=CFL0*CP*CD
    DM=TMP(1)-TMP(2)
    GE=FLO*CP*HD
    UC=QC/DM/3./3.14/DCT/CL
    UA=(GE+QC)/DM/3./3.14/(DET+DCT)/CL
    UE=QC/DM/3./3.14/DET/EL
    IF(MM-11804,505)
304 COS=((TMS(8)+TSC9)+TMS(12))/3.0-TMS(4),
    HDS=TMS(3)-((TMS(5)+TMS(6)+TMS(7))/3.0)
GOTO 506

505 CDS=TMS(12)
    HDS=TMS(11)
    DMS=TMS(1)-TMS(2)
506 QES=FLO*CP*HDS
    CCS=CFL0*CP*CD
    UEE=QES/3./3.14/DET/EL/DMS
    UCA=QES/3./3.14/(DET+DCT)/CL/DMS
    UCS=CSS/3./3.14/DET/CL
    UAC=QES/3./3.14/DET/EL/DMS
    UAS=QES/3./3.14/(DET+DCT)/CL
    VCUN=(VCUN+(ZN-1.0))/(UC-UCS)**2/ZN
    VEUN=(VEUN+(ZN-1.0))/(UE-UEE)**2/ZN
    UEA=QES/3./3.14/(DET+DCT)/EL/DMS
    ZN=ZN-DSET5700.701
    IF(ZN<0,C)
GOTO 210

701 VCUN=SCRT(VCUN)-100.0/UAS
WRITE(2,110)HC,CD
705 CAN=SCRT(VCAN)-100.0/UAS
VEUN=SCRT(VUN)-100.0/UUEE

310 FORMAT('INST. HW AND CV TEMP DIFF.(C)=",F6.2,5X,F6.2,2/
WRITE(2,120)I TEST, I RUN, MONTH, DATE
10 FORMAT('TEST NO."",IX,13,IX,"RUN NO."",IX,12,"A2",A2,IX,12,IX,"7")
WRITE(2,12)ISHR,ISMIN,ISSEC,ISHRC,IFMIN,IFSEC,IFMSC
12 FORMAT('START TIME"",IX,12,"12","",12,"","",12,"FINISH TIME","",12,"")
WRITE(2,14)EVANG,CNANG,ELEV,CIRG
14 FORMAT('EV-ANG.(DEG)=",IX,12,"CN-ANG.(DEG)=",IX,12,/"
0"CN-ELEV(M)=",IX,F5.3,"EVA-CHARGE(Z)=",IX,F5.3,"/
WRITE(2,16)DSETS
16 FORMAT('12X"AVG-RESULTS OF",IX,F4.0,1X,"SETS OF DATA","
WRITE(2,18)HFL0,CFL0
18 FORMAT('HOT WATER FLOW RATE(KG/SEC)=",IX,F7.5,"COLD WATER FLOW RA
STECK/SEC="",1X,F7.5)
WRITE(2,20)TMS(3),TMS(5),TMS(6),TMS(7)
20 FORMAT('TEMP OF HOT WATER IN(C)=",IX,F6.2,"TEMP OF COLD WATER IN(C)=",IX,F6.2")
WRITE(2,22)TMS(4),TMS(8),TMS(9),TMS(10)
0PS(C)="", 1X,F6.2, 1X,F6.2, 1X,F6.2
WRITE(2,24)TMS(1),TMS(2)

24 FORMAT("AVG-EVAP-WALL TEMP(C)="", 1X,F6.2,"/"AVG-COND-WALL TEMP(C)="", 1X,F6.2)
WRITE(2,32)CES, CCS

32 FORMAT("GE(WATT)="", 1X,F9.1,"/"GC(WATT)="", 1X,F9.1)
WRITE(2,38)VUNZ, VUNZ, VUNZ

38 FORMAT(3X,"EVAP", 7X,"COND", 7X,"AVG", "/"COEF. OF VARIATION
OF U(%)="", 2X,F7.1, 5X,F7.1, 5X,F7.1, 3/
WRITE(2,41)UEE, UEA, UAE, UAS, UAC, UCA, UCS
WRITE(2,40)

40 FORMAT(5/"TO RESTART, PUSH RUN-TO REPEAT, TYPE 4. TO CONTINUE WITH
0 NEW"/"INPUT, TYPE '3.' TO STOP; TYPE '2.'", 3/
PAUSE 55
READ(1,2)SKIP

2 FORMAT(F4.0)
IF(3,-SKIP)205,204,203

203 STOP
END

SUBROUTINE STATS(IDVN)

350 CALL L66(1,IDVN,ISTAT,ITRLG)
IF(ISTAT)350,351
351 IF(IAND(ISTAT,37B)-1)350,352,353
353 PAUSE 77
352 RETURN
END

**END-OF-TAPE
*/
**END-OF-TAPE
*/
**END-OF-TAPE
### Sample Output of Experiment

<p>| | | | | | |</p>
<table>
<thead>
<tr>
<th></th>
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<td>34.60</td>
<td>25.70</td>
<td>25.71</td>
<td>25.71</td>
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<tr>
<td>34.02</td>
<td>34.01</td>
<td>33.99</td>
<td>26.79</td>
<td>26.81</td>
<td>26.83</td>
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<tr>
<td>33.98</td>
<td>33.96</td>
<td>33.94</td>
<td>26.43</td>
<td>26.44</td>
<td>26.46</td>
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<tr>
<td>33.95</td>
<td>33.94</td>
<td>33.92</td>
<td>26.15</td>
<td>26.15</td>
<td>26.17</td>
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<tr>
<td>33.90</td>
<td>33.89</td>
<td>33.87</td>
<td>25.89</td>
<td>25.91</td>
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<td>33.83</td>
<td>33.82</td>
<td>33.80</td>
<td>25.67</td>
<td>25.69</td>
<td>25.70</td>
</tr>
</tbody>
</table>

#### Test No. 1 Run No. 3

- **JUNE**: 12 77
- **START TIME**: 0:19:14: 3
- **FINISH TIME**: 0:19:31: 6
- **EV. ANG. (DEG)** = 0
- **CN. ANG. (DEG)** = 0
- **CN. ELEV(M)** = .900
- **EVAP. CHARGE(%)** = 70.00

#### Avg. Results of 10. Sets of Data

- **HOT WATER FLOW RATE (KG/SEC)** = .08320
- **COLD WATER FLOW RATE (KG/SEC)** = .08630
- **TEMP. OF HOT WATER IN(°C)** = 35.20
- **HOT WATER OUTLET TEMPS(°C)** = 34.61 34.60 34.60
- **TEMP. OF COLD WATER IN(°C)** = 25.18
- **COLD WATER OUTLET TEMPS(°C)** = 25.70 25.71 25.71
- **AVG. EVAP. WALL TEMP(°C)** = 33.92
- **AVG. COND. WALL TEMP(°C)** = 26.22
- **QE(WATT)** = 207.6
- **QC(WATT)** = 192.4
- **COEF. OF VARIATION OF U(%)** = .2

#### Watts/MRE

<table>
<thead>
<tr>
<th>TFE</th>
<th>UEA</th>
<th>UAE</th>
<th>UAA</th>
<th>UC</th>
<th>UGA</th>
<th>UCC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1027.1</td>
<td>762.6</td>
<td>934.6</td>
<td>731.0</td>
<td>581.3</td>
<td>699.4</td>
<td>556.2</td>
</tr>
</tbody>
</table>
APPENDIX D

This Appendix describes the sequence of tests carried out to provide data to test the simulation program as severely as possible. Table D1 shows the detailed description of parameter variation.

The experimentally measured value of $U_{ee}$ is used to compare with the corresponding simulated value. In any experimental sequence (I, II, III, etc.), if it is possible, it is preferable to set the angle of inclination of either heat exchanger first and then vary other parameters. Any other method of variation during experiments will be more time consuming.

The following list discusses the purpose of each test sequence. The simulation plots of results shown in Chapter 4 show the experimental results as points on the plots.

Test Sequence I: Effect of evaporator static charge and condenser inclination for a unidirectional loop. R-11 as working fluid.

Test Sequence II: Effect of temperature difference for a unidirectional loop. R-11 as working fluid.

Test Sequence III: Effect of evaporator static charge and larger evaporator tube diameter (smaller condenser tube diameter). R-11 as working fluid.

Test Sequence IV: Effect of evaporator static charge and loop inclination for a bidirectional loop. Larger tube evaporator. R-11 as working fluid.
Test Sequence V: Effect of evaporator static charge and loop inclination for a bidirectional loop. Smaller tube evaporator. R-11 as working fluid.

Test Sequence VI: Effect of evaporator static charge using a different working fluid (R-113) for a unidirectional loop. Larger tube evaporator.

Test Sequence VII: Effect of evaporator static charge and loop inclination for a bidirectional loop using R-113 as working fluid and smaller tube evaporator.

Test Sequence VIII: Effect of condenser inclination and evaporator static charge for a unidirectional loop with R-113 as working fluid. Smaller tube evaporator.

Test Sequence IX: Effect of temperature difference and charge for a unidirectional loop with condenser inclined at 45°. Smaller tube evaporator and R-11 as working fluid.

Test Sequence X: Effect of evaporator inclination with larger tube evaporator for a unidirectional loop. R-113 as working fluid.

Test Sequence XI: Effect of temperature difference and evaporator static charge for a bidirectional loop. Smaller tube evaporator and R-11 as working fluid.
<table>
<thead>
<tr>
<th>Test sequence</th>
<th>Working fluid</th>
<th>Diameter evap/cond (mm)</th>
<th>Cond. elev. (m)</th>
<th>Levap. (degrees)</th>
<th>Lcond. (degrees)</th>
<th>%Evap static charge</th>
<th>$\Delta T$ evap cond (°C)</th>
<th>$T_{mean}$ evap cond (°C)</th>
<th>Fig. No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>R-11</td>
<td>4.83/7.90</td>
<td>0.9</td>
<td>0</td>
<td>0,15,30, 45 &amp; 75</td>
<td>20 to 70</td>
<td>10</td>
<td>30</td>
<td>21</td>
</tr>
<tr>
<td>II</td>
<td>R-11</td>
<td>4.83/7.90</td>
<td>0.9</td>
<td>0</td>
<td>0</td>
<td>20 to 70</td>
<td>6 &amp; 20</td>
<td>30</td>
<td>23</td>
</tr>
<tr>
<td>III</td>
<td>R-11</td>
<td>4.83/7.90</td>
<td>0.9</td>
<td>0</td>
<td>75</td>
<td>15 to 70</td>
<td>10</td>
<td>35</td>
<td>21</td>
</tr>
<tr>
<td>IV</td>
<td>R-11</td>
<td>4.83/7.90</td>
<td>0.0</td>
<td>0</td>
<td>0,15,45, 6 &amp; 75</td>
<td>25 to 40</td>
<td>10</td>
<td>45</td>
<td>26</td>
</tr>
<tr>
<td>V</td>
<td>R-11</td>
<td>4.83/7.90</td>
<td>0.0</td>
<td>0</td>
<td>0,15,45, 6 &amp; 75</td>
<td>40 to 50</td>
<td>10</td>
<td>45</td>
<td>25</td>
</tr>
<tr>
<td>VI</td>
<td>R-113</td>
<td>4.83/7.90</td>
<td>0.9</td>
<td>0</td>
<td>0</td>
<td>25 to 60</td>
<td>10</td>
<td>55</td>
<td>22</td>
</tr>
<tr>
<td>VII</td>
<td>R-113</td>
<td>4.83/7.90</td>
<td>0.0</td>
<td>0,15,45, 6 &amp; 75</td>
<td>0,15,45, 6 &amp; 75</td>
<td>40 to 50</td>
<td>10</td>
<td>55</td>
<td>27</td>
</tr>
<tr>
<td>VIII</td>
<td>R-113</td>
<td>4.83/7.90</td>
<td>0.9</td>
<td>0</td>
<td>0,45 &amp; 75</td>
<td>35 to 65</td>
<td>10</td>
<td>55</td>
<td>22</td>
</tr>
<tr>
<td>IX</td>
<td>R-11</td>
<td>4.83/7.90</td>
<td>0.9</td>
<td>0</td>
<td>45</td>
<td>20 to 60</td>
<td>5,10,20,30</td>
<td>45 \</td>
<td>24</td>
</tr>
<tr>
<td>X</td>
<td>R-113</td>
<td>4.83/7.90</td>
<td>0.9</td>
<td>0</td>
<td>45</td>
<td>40</td>
<td>10</td>
<td>55</td>
<td>22</td>
</tr>
<tr>
<td>XI</td>
<td>R-11</td>
<td>4.83/7.90</td>
<td>0.0</td>
<td>75</td>
<td>75</td>
<td>35 to 55</td>
<td>6,10 &amp; 20</td>
<td>45</td>
<td>28</td>
</tr>
</tbody>
</table>

**TABLE D 1:** DESCRIPTION OF PARAMETER VARIATION
APPENDIX E

LIST OF EQUIPMENT AND INSTRUMENTATION

1. **Armstrong Armflex Pipe Insulation**, 1/2" I.D., 1/2" wall thickness, thermal conductivity: 0.258 BTU/ft²-HR-°F at 90°F. Armstrong Cork Co., Canada

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, Clock timer control card. One scanner card for up to 10 data-channels. Hewlett Packard, U.S.A.

4. **Dial Pressure Gauge**, 200 psig to 30 inches Hg vacuum, for R-12, R-22 and R-502. Marsh Instrument Co. Skokie, 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**, cylinder, DuPont of Canada

8. **Freon-11**, cylinder, DuPont of Canada

9. **Flowmeter (Rotameter)**, type 8-1110 for water flow rate, graduations 0 - 250 in increments of 10. Hatfield, Penn., U.S.A.

10. **Integrating Digital Voltmeter**, Model 2402A, 0.001mv reading accuracy after 60 minutes warm up. 43 measurements per second (triggered externally). One measurement every 10 seconds to 10 measurements per second (triggered internally). Hewlett Packard, U.S.A.
11. **Lauda Constant Temperature Bath**, Type K-40, 115V, 800W, Messerate-Werk Lauda, W. Germany


13. **Stir-Cool Unit for ice bath**. Lab-Line Instruments Inc. Chicago, Ill., U.S.A.

14. **Foam Insulation Sheets**, 1-1/2" thickness x 18" x 4 feet Thermal conductivity: 0.258 BTU/ft² hr. °F at 90°F. Armstrong Cork Co. Canada.

15. **Silicone Seal**, adhesive. Canadian General Electric, Canada

16. **Teleprinter**, Model 2752A. Typing speed: 100 words per minute. Punching and reading speed: 10 words per second Hewlett Packard, U.S.A.

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


19. **"Skan-Flo" Purge Rotameter Type 175**, 1/8" NPT, 0-170 graduations in increments of 1, choice of different floats like sapphire, stainless steel, etc. Supplied by Brian Engineering Ltd., Kitchener, Ontario, Canada.
## APPENDIX F

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

<table>
<thead>
<tr>
<th>Property</th>
<th>R-11</th>
<th>R-113</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical Formula</td>
<td>$CCl_3F$</td>
<td>$CCl_2F$</td>
</tr>
<tr>
<td>Molecular weight</td>
<td>137.38</td>
<td>187.39</td>
</tr>
<tr>
<td>Boiling temperature at 14.7 psia $^\circ F$</td>
<td>74.9</td>
<td>117.6</td>
</tr>
<tr>
<td>Freezing temperature at 14.7 psia $^\circ F$</td>
<td>-168</td>
<td>-31</td>
</tr>
<tr>
<td>Critical temperature, $^\circ F$</td>
<td>388.4</td>
<td>417.4</td>
</tr>
<tr>
<td>Critical pressure, psia</td>
<td>640</td>
<td>498.9</td>
</tr>
<tr>
<td>Critical density, lb/cu ft</td>
<td>34.6</td>
<td>36.0</td>
</tr>
<tr>
<td>Density of liquid, $86^\circ F$, lb/cu ft</td>
<td>91.39</td>
<td>96.96</td>
</tr>
<tr>
<td>Sp vol of sat vapour $5^\circ F$, cu ft/lb</td>
<td>12.205</td>
<td>27.04</td>
</tr>
<tr>
<td>Sp heat of liquid $86^\circ F$, BTU/lb $^\circ F$</td>
<td>0.21</td>
<td>0.218</td>
</tr>
<tr>
<td>Sp heat ratio ($C_p/C_v$):</td>
<td></td>
<td></td>
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<tr>
<td>vapour at $86^\circ F$ and 14.7 psia</td>
<td>1.13</td>
<td>1.12</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sat liquid, $5^\circ F$</td>
<td>0.058</td>
<td>0.044</td>
</tr>
<tr>
<td>Sat liquid, $86^\circ F$</td>
<td>0.049</td>
<td>0.037</td>
</tr>
<tr>
<td>Vapour at sat pres, $5^\circ F$</td>
<td>0.0034</td>
<td>0.0035</td>
</tr>
<tr>
<td>Vapour at 14.7 psia, $86^\circ F$</td>
<td>0.0045</td>
<td>0.0045</td>
</tr>
<tr>
<td>Viscosity, centipoises</td>
<td></td>
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</tr>
<tr>
<td>Sat liquid, $5^\circ F$</td>
<td>0.630</td>
<td>1.28</td>
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<tr>
<td>Sat liquid, $86^\circ F$</td>
<td>0.404</td>
<td>0.638</td>
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<tr>
<td>Vapour at sat pres, $5^\circ F$</td>
<td>0.0087</td>
<td>0.0079</td>
</tr>
<tr>
<td>Vapour at 14.7 psia, $86^\circ F$</td>
<td>0.0108</td>
<td>0.0096</td>
</tr>
</tbody>
</table>
APPENDIX G
ERROR ANALYSIS

The uncertainty in the calculation of loop conductance is presented here.

The probable error in a calculated value due to uncertainties in the quantities used in the calculation were estimated by using the following equation:

$$W_z = \left[ \left( \frac{dz}{dx_1} \cdot W_1 \right)^2 + \left( \frac{dz}{dx_2} \cdot W_2 \right)^2 + \ldots + \left( \frac{dz}{dx_n} \cdot W_n \right)^2 \right]^{1/2}$$  \hspace{1cm} \text{(AG-1)}

where $z$, the calculated quantity, 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.

G.1 ERROR IN LOOP CONDUCTANCE

The error in the loop conductance 'U' due to inaccuracies in measurement was calculated the following way:

$$U_{e,e} = \frac{\left( m_{hw} \cdot C_{hw} \cdot \Delta T_{hw} \right)}{3 \cdot \Delta T_{ec} \cdot \pi \cdot D_e \cdot L_e}$$  \hspace{1cm} \text{(AG-2)}

where $m_{hw}$ stands for hot water, $\Delta T_{ec}$ stands for temperature difference between the average inner tube surface temperatures of evaporator and condenser tubes and $e$ stands for evaporator tube.

$$\Delta T_{hw} = T_{hw_i} - T_{hw_o}$$  \hspace{1cm} \text{(AG-3)}

where $T_{hw_i}$ and $T_{hw_o}$ are temperature readings at the inlet and outlet of the evaporator water respectively.
\[ \Delta T_{ec} = T_e - T_c = \frac{(T_1 + T_2 + \ldots + T_{15}) - (T_{16} + T_{17} + \ldots + T_{30})}{15} \]  

where \( T_1, T_2, \ldots, T_{15} \) are the inside surface temperatures of the evaporator and \( T_{16}, \ldots, T_{30} \) are the inside surface temperatures of the condenser respectively. (Figure 15)

The manufacturing tolerances of the inside diameter of the evaporator tube and the fabrication tolerance on the length of the evaporator were \( 0.190" \pm 0.003" \), and 2 feet \( \pm 0.0026' \) respectively. The error in temperature measurement is \( \pm 0.1^\circ C \) using thermocouple manufacturer's data.

The mass flow rate of hot water was measured to the accuracy of \( \pm 2 \) lbs/hr. based on the sampling time and the least count of the weighing scale. The variation of specific heat with temperature was neglected as the temperature difference across the evaporator was hardly ever \( 2^\circ C \).

Then according to equation (AG-1), the probable error in loop conductance 'U', based on evaporator area and evaporator heat transfer may be written as:

\[
W_{Ue,e} = \left[ \left( \frac{dU}{dW_{hw}} \right) \left( \frac{dW_{hw}}{dW_e} \right)^2 + \left( \frac{dU}{d\Delta T_{hw}} \right) \left( \frac{\Delta T_{hw}}{W_e} \right)^2 + \left( \frac{dU}{d\Delta T_{ec}} \right) \left( \frac{\Delta T_{ec}}{W_e} \right)^2 \right]^{1/2}
\]

\[
+ \left( \frac{dU}{dD_e} \right)^2 \left( \frac{dD_e}{W_e} \right)^2 \right]^{1/2}
\]

AG-5a

where

\[
\Delta T_{hw} = \left[ \left( \frac{d\Delta T_{hw}}{d\Delta T_{hwi}} \right) \left( \frac{\Delta T_{hwi}}{W_{hwo}} \right)^2 + \left( \frac{d\Delta T_{hw}}{d\Delta T_{hwo}} \right) \left( \frac{\Delta T_{hwo}}{W_{hwo}} \right)^2 \right]^{1/2}
\]

\[
\Delta T_{ec} = \left[ \left( \frac{d\Delta T_{ec}}{dT_1} \right) \left( \frac{\Delta T_1}{W_{T_1}} \right)^2 + \left( \frac{d\Delta T_{ec}}{dT_2} \right) \left( \frac{\Delta T_2}{W_{T_2}} \right)^2 + \ldots + \left( \frac{d\Delta T_{ec}}{dT_{15}} \right) \left( \frac{\Delta T_{15}}{W_{T_{15}}} \right)^2 + \left( \frac{d\Delta T_{ec}}{dT_{16}} \right) \left( \frac{\Delta T_{16}}{W_{T_{16}}} \right)^2 \right]^{1/2}
\]
\[ \ldots \ldots + \left( \frac{dT_{ec}}{dT_{30}} \frac{wT_{30}}{wT_{30}} \right)^2 \] 1/2

or dividing the equation (AG-5a) by 'U', the error can be obtained as:

\[ \frac{WU_{e,e}}{U_{e,e}} = \left[ \left( \frac{\Delta T_{hw} - \Delta m_{hw}}{\Delta T_{hw}} \right)^2 + \left( \frac{m_{hw} - \Delta T_{hw}}{\Delta T_{hw}} \right)^2 \right]^{1/2} \]

AG-5b

We have considered 'U_{e,e}' here because this value of loop conductance was used to verify the simulation program result.

A sample calculation for the probable error for a typical experimental run would be as follows:

From equation AG-5b,

\[ \frac{WU_{e,e}}{U_{e,e}} = \pm 0.05 \text{ or } 5\% \]

If we assume approximately an equal amount of uncertainty including the effect of condenser also, the net uncertainty in the calculation of 'U_{a,a}' will be \( \pm 5\% \) where 'U_{a,a}' is the average loop conductance based on average Q and average area of condenser and evaporator surface.
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