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A TWO PHASE CLOSED THERMOSYPHON

by

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of the University of Ottawa
in partial fulfilment of the requirements for the
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A TWO PHASE CLOSED THERMOSYPHON

✓ Advisor

Candidate

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SUMMARY

A two phase closed thermosyphon is a totally enclosed heat transfer system based on the principle that heat is transferred from one part of the system to the other part by employing the heat transfer characteristics of boiling and condensation of the working fluid.

The present system is a sealed tube partially filled with a working fluid but, unlike a heat pipe, has no internal wick and works under a body force field such as gravity, centrifugal force etc. The system is capable of transmitting fairly large rates of heat transfer particularly in one direction and therefore among its many applications in engineering, it can be used as a means of preserving permafrost. For its simplicity in geometry and structure, it appears to be cheaper and more reliable than other methods known for the purpose. However, little attempt was made previously to study the system with change of phase varying the various parameters systematically.

It is the purpose of the present study, therefore, to make an analytical and experimental investigation into the performance of a two phase closed thermosyphon. Particular attention is given to the low pressure operation.

An experimental set up including a refrigeration circuit was constructed. The test tube made of 1 in. SPS copper pipe, 54 in. long, was electrically heated at the lower end and cooled at the upper section

by a number of separate cooling jackets, which help maintaining a constant temperature and also facilitates changing the heated length-cooled length ratio. The test tube had a provision to remove air and other non-condensable gases, and was installed with various pressure, temperature, and heat flux measuring probes.

In the present investigation, the following parameters were studied.

- a) amount of the working fluid, b) heated length-cooled length ratio,
- c) mean operating pressure, d) heat flux, and e) working fluid

The heat transfer coefficient which is defined as tube heat flux divided by the mean wall temperature difference between heated and cooled sections, was influenced little by the quantity of the working fluid above a certain quantity. At very small quantities, the heat transfer coefficient increases with increasing amount of the working fluid. An increase in heated length-cooled length ratio resulted in a decrease in the heat transfer coefficient but it showed a tendency of becoming independent of heat flux at high pressures. The heat transfer coefficient, as might be expected, was very sensitive to the operating pressure. Water was found superior to Freon-11 as the working fluid.

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NOMENCLATURE

C_p	specific heat at constant pressure of the working fluid in the liquid phase, B.T.U./lb ^o F
D	inside diameter of the thermosyphon tube, ft
h	heat transfer coefficient for the two phase closed thermosyphon tube, $q/(T_h - T_c)$, B.T.U./hr. ft ² . ^o F
h_m	maximum heat transfer coefficient for the two phase closed thermosyphon tube, $q_m/(T_h - T_c)$, B.T.U./hr. ft ² . ^o F
h_{fg}	latent heat of evaporation of the working fluid, B.T.U./lb
G	mass flow rate of the condensate, lb/hr
g	acceleration due to gravity, ft/sec ²
g_c	gravitational constant, ft/sec ²
k	thermal conductivity of the working fluid in the liquid phase, B.T.U./hr. ft. ^o F
L_c	cooled length of the thermosyphon tube, ft
L_h	heated length of the thermosyphon tube, ft
ΔP_g	force due to the gravity head
ΔP_l	force due to the liquid viscous drag
ΔP_m	force due to the pressure drop caused by the momentum changes

ΔP_v	force due to the vapor pressure drop
Q	total heat input in the evaporator section of the thermosyphon tube, B.T.U./hr
q	uniform heat flux in the evaporator section of the thermosyphon tube, B.T.U./hr. ft ²
q_m	maximum heat transfer rate of the thermosyphon tube, B.T.U./hr. ft ²
R_i	inside radius of thermosyphon tube, ft
R	heated length-cooled length ratio,
r_i	radius of the vapor flow passage in the condensing section of the thermosyphon tube, ft
r	radius of an infinitesimal element of the liquid film in the condensing section, ft
T_c	average temperature in the condensing section of the thermosyphon tube, °F
T_h	average temperature in the heated section of the thermosyphon tube, °F
T_s	saturation temperature corresponding to the mean operating pressure p , inside the thermosyphon tube, °F
T_v	vapor core temperature in the thermosyphon tube, °F

- u velocity of the condensate in the x-direction, ft/sec.
- V_1 volume of the initial charge of the liquid in thermosyphon, cu ft
- V_t total volume of the thermosyphon tube, cu ft
- V^+ dimensionless quantity of the working fluid in the thermosyphon tube, V_1/V_t
- y_1 dimensionless liquid film thickness at any z , $(1 - r_1/R)$
- y_1' dimensionless liquid film thickness at the end of the condensing section at $z = L_c$, $(1 - \delta/R)$
- z downward vertical axis of the thermosyphon tube from the top of the condensing end.
- Nu_d Nusselt number, $(h D/k)$
- Pr Prandtl number, $(C_p \mu/k)$
- Gr Grashof number, $(\rho^2 g \theta \beta D^3 / \mu^2)$
- Gr_d modified Grashof number $(\theta \beta D^4 \rho^2 C_p / \mu k L_h)$

Greek letter symbols

- β coefficient of thermal expansion, $^{\circ}\text{F}$
- δ liquid film thickness at the condensing end, ft
- θ temperature difference between the average temperature in the heated section and the average temperature in the condensing section, $(T_h - T_c)$, $^{\circ}\text{F}$
- μ absolute viscosity of the working fluid in the liquid phase, $\text{lb}_f \cdot \text{hr} / \text{ft}^2$
- ρ density of the working fluid in the liquid phase, lb/ft^3

1. INTRODUCTION

The transmission of heat plays an increasingly important role in many areas of modern technology. For example, a nuclear power station produces energy in the form of heat which should be taken out of the core of the reactor so that it can be converted into other useful forms of energies. Other important instances where heat removal is required are heat exchangers of various kinds, gas turbine rotor blades, and as a special case, permafrost when used as a structural foundation.

Efficient and economical means of moving heat from one place to another have been the subject of a great deal of research, as a result of which systems known as "thermosyphons" and "heat pipes" came into existence. Out of several systems of heat transfer, a thermosyphon seems to be able to perform the task of transporting heat from one place to another satisfactorily.

A syphon is defined as a closed conduit, which lifts the liquid to a level higher than its free surface and then discharges it at a lower level, without the use of any mechanical work. In a syphon potential energy is continuously converted into the kinetic energy.

A thermosyphon, on the other hand, is a system where the circulation of the working fluid is caused by an external force field such as gravity, centrifugal forces etc. These forces act on the fluid of variable density. The density variation is a result of a temperature gradient in the fluid. Under steady state conditions, the

potential and kinetic energies within a thermosyphon remain constant.

A thermosyphon may thus be defined as a system containing a fluid which circulates because of density variations caused by the temperature gradients but without the use of mechanical work.

Contrary to the general understanding, a thermosyphon greatly differs from a heat pipe. A heat pipe uses an internal wick to bring the condensate back to the evaporator which relies on the capillary action of the wick and the working fluid. A thermosyphon employs an external force field like gravity, centrifugal forces etc. for the task of bringing the condensate back to the evaporator.

1.1 The classification of the thermosyphons:

The broad definition of a thermosyphon covers a variety of natural phenomena and man made systems. They may be identified as a particular class by examining the following characteristics:

1. The nature of the boundaries
2. The regime of heat transfer
3. The number of phases present
4. The nature of the body force

1.1.1 The nature of the boundaries:

Thermosyphons may be grouped in one of the following two categories depending upon their nature of the boundaries.

- a. closed thermosyphon
- b. open thermosyphon

A closed thermosyphon is a system completely enclosing a finite volume of the working fluid and where no mass transfer of the fluid across the boundaries take place. It is clear that heat should be both received and rejected within such a system to obtain steady state conditions and circulation of the fluid.

An open thermosyphon, on the other hand, is a system where mass transfer of the working fluid may also take place. In such systems steady state conditions and circulation of the working fluid can be obtained even when heat transfer takes place only in one direction within the system. This is made possible by mass transfer across the open boundaries.

A typical open thermosyphon is a tube sealed at the lower end which opens up at the top into a large reservoir of fluid maintained at a constant temperature. The tube is externally heated over its entire length. If the tube is also sealed at the top and heat is supplied over a part of its length and rejected over another part, it would represent a closed thermosyphon.

1.1.2 The regime of heat transfer:

Depending upon the regime of heat transfer, thermosyphons may be further classified. Two regimes of convective heat transfer are possible.

- a. Free or natural convection
- b. Mixed (free and forced combined) convection

Free convection occurs when the circulation of the fluid is obtained by differences in the density under the action of a force field. The density variations are caused by heat conducted to, or from, the fluid adjacent to the heat transfer surface.

Forced convection occurs when bulk motion is created by pressure differences. From the definition of a thermosyphon, an external source of mechanical work can not be responsible for this pressure difference but they may be created when internal boundaries separate the flow. Free and forced convection when present together are referred to as mixed convection.

Tube geometry of a thermosyphon would decide as to whether free or mixed convection would result.

1.1.3 The number of phases present:

Thermosyphons may be further categorized in one of the following two groups depending upon the number of phases present.

- a. Single phase
- b. Two phase

A single phase thermosyphon is a system in which the working fluid is either an entirely liquid or an entirely gaseous phase throughout its operation. On the other hand, a system which has both liquid and vapor

phases of the same working fluid is termed as a two phase thermosyphon.

Two phase thermosyphons have a great advantage over those employing single phase because boiling and condensation are efficient means of heat transfer. Further, under conditions other than critical point, a change in phase results in a large change in density. In a two phase closed thermosyphon, mixed convection may result if the descending liquid and ascending vapor have separate paths in either the boiling section or the condensing section, as in Foster's thermosyphon shown in Fig. 1.1(c).

1.1.4 The nature of the body force:

Finally the nature of the body force causing fluid motion in a thermosyphon would decide its one of the following two classes.

- a. Stationary thermosyphon
- b. Rotating thermosyphon

A stationary thermosyphon is a system where gravity is the only body force field. If a thermosyphon is rotated, centrifugal forces also exist which could be as much as 10^4 greater than the forces due to gravity. A rotating thermosyphon is one where, in addition to gravity, centrifugal forces are also present. In a rotating thermosyphon, the force due to gravity is negligible compared to centrifugal forces. The development of rotating thermosyphon for the cooling of gas turbine rotor blades was pioneered by Schmidt^{(1)*}, this device is shown in Fig.1.1(a). Also shown in Figures 1.1(b) and 1.1(c) are the rotating thermosyphons as

* Number in parentheses designate References at the end of the thesis.

proposed by Bayley and Cohen⁽¹⁶⁾ and Foster⁽²⁾ respectively, all for the cooling of gas turbine rotor blades.

In addition, there could also exist electrical and magnetic body force fields.

1.2 General characteristics of thermosyphons

Though thermosyphons may be designed to transfer heat or mass, or both, and belong to any class, use any working fluid and have any geometrical configuration, the following characteristics are common to them all.

1. It has no mechanical parts and will operate indefinitely without maintenance provided there is no deterioration of the fluid or the tube material.
2. The surfaces at which heat is received or rejected may be remote from one another.
3. The apparent conductivity increases as the over-all temperature difference increases, unlike the conductivity of the solids.
4. The operation of the thermosyphon tube is almost noiseless over most of its range.
5. It may act as a heat transformer.
6. Unlike heat pipes, heat flows mainly in one direction, especially in case of a two phase closed thermosyphon and therefore behaves like a diode.

1.3 Objective of the present thesis

The object of the present thesis is therefore, to make an experimental investigation into the performance of a stationary two phase closed thermosyphon tube. The investigation was carried out with a view to using the thermosyphon tube as a means of preserving permafrost. The effect of the following parameters on the tube performance as a heat transfer system was studied.

1. Heat flux
2. Amount of the working fluid
3. Heated length-cooled length ratio
4. Mean operating pressure (or corresponding saturation temperature)
5. Working fluid

Particular attention has been given to the low pressure operation. A qualitative analysis of the system was also made.

2. PREVIOUS STUDIES

The interest in the thermosyphon first arose from the proposed application of an open thermosyphon for the cooling of gas turbine rotor blades by Schmidt⁽¹⁾ during 1939-45 war. There after, the thermosyphon system without change of phase has been the subject of a great deal of research.

The earlier investigations were concerned with the open thermosyphon in which the heated lower region opens up at the top into a large reservoir of fluid from which heat is removed either by allowing evaporation to occur⁽¹⁾ or by means of an immersed cooling coil^(3 to 7). On the other hand, closed thermosyphons, which certainly have many advantages over open thermosyphons, have received little attention, probably because of the complexities of flow and boiling heat transfer.

Hahnemann⁽⁸⁾ built a number of closed thermosyphons to operate without phase change but, according to Reference (9), appears to have made little attempt to vary the various parameters systematically. Bayley and Lock⁽⁹⁾ described a theoretical and experimental investigation into the performance of a single phase closed thermosyphon. They observed that decreasing the heated length-cooled length ratio leads to a consistently increasing over all heat transfer performance. They also reported that a single phase closed thermosyphon has an optimum performance in the Prandtl number range of 10 and 300. The effect of the ratio of the tube diameter to heated length was noted to be significant for

values of above 7.5.

Though extensive literature is available on systems known as heat pipes,^(10 to 15) which greatly differ from two phase closed thermosyphons as explained in the section 1.1, little is known on the system studied in the present thesis, other than the work reported in References (16) and (17).

Cohen and Bayley⁽¹⁶⁾ have reported a number of experiments on both rotating and stationary two phase closed thermosyphon tubes. They studied them, however, for the purpose of using them for the cooling of gas turbine rotor blades. For rotating tests, they used $6\frac{1}{8}$ in. long circular steel tubes of three different bore sizes $\frac{1}{16}$ in., $\frac{1}{8}$ in., and $\frac{1}{4}$ in., but of the same wall thickness $\frac{1}{16}$ in. Water was used as the working fluid and heated length-cooled length ratio was fixed approximately at one. They noted that the tube heat transfer coefficient increases with increase in tube diameter, saturation pressure, and rotating speed. They also reported that the effect of the quantity of the working fluid was un-noticable above a very small quantity, this quantity being less for tubes of larger diameters. For stationary tests, a $2\frac{1}{2}$ ft long copper tube of $\frac{3}{4}$ in. I.D. was used. The heated length-cooled length ratio was fixed at 0.25. They noticed that as in rotating tests, the tube heat transfer coefficient was not sensitive to the amount of the working fluid above a small quantity and increased with increase in the tube diameter and mean operating pressure. Water was found superior compared to Butyl alcohol and Chlorobenzene as the working fluid. They observed a dry out at the bottom of the tube at

a heat input of 2.5 KW when the quantity of water in the stationary tube was 4 cu cm. The effect of the heated length-cooled length ratio on the heat transfer capability of the thermosyphon tube was not studied by Cohen and Bayley.

Larkin⁽¹⁷⁾, in his unpublished work, noted that the depth of the working fluid may be reduced to one third of the heated length without significantly affecting the performance of the thermosyphon tube.

3. QUALITATIVE ANALYSIS OF THE TWO PHASE CLOSED THERMOSYPHON

Two phase closed thermosyphons are capable of transmitting heat transfer rates greatly in excess of that which could be obtained by the use of a homogenous piece of any known material. This property is achieved within the tube by the evaporation of a liquid, transport of the vapor to another part of the tube, condensation of the vapor and return of the condensate to the evaporator through gravity or any other external body force action.

The two phase closed thermosyphon principle is applicable over a very wide range of sizes, shapes, temperatures, and materials. Unlike solid heat conductors, two phase closed thermosyphons can not be characterized by a single property (an equivalent thermal conductivity) because their behavior and limitations are largely integral properties of the device as a whole. Further, even if the size, shape, temperature, and material of a two phase closed thermosyphon is fixed, the mass, vapor volume and maximum heat flux are under the control of the designer. The particular application will determine which allowed combination of these properties is most suitable.

A purely theoretical study of the problem is not possible at present due to lack of information and complexities of the boiling process, particularly in a closed cavity. However, with the aid of the experimental observations, the effect of the various parameters may be examined on the performance of a thermosyphon tube.

In view of such complexities and uncertainties associated with the

boiling process in a thermosyphon tube, its operation and mechanism of heat transfer would first be studied.

3.1 Operating principle and mechanism of heat transfer:

The operation of a two phase closed stationary thermosyphon combines two familiar principles given below.

1. Two phase heat transfer in a thermosyphon
2. Flow under gravity

Two phase heat transfer in a thermosyphon is responsible for transferring heat from the evaporator section to the condensing section. This process in a thermosyphon is different than that occurring in a conventional two phase boiling and condensing equipment. A thermosyphon transfers heat with very small temperature differences, which is achieved by utilizing the fluid's latent heat as the primary mode of heat transfer. Conventional boilers, however, employ primarily convection for transferring heat and the contribution of latent heat is comparatively small toward heat transfer.

The function of the working fluid within the thermosyphon is to absorb the heat energy at the evaporator section, transport it through the pipe, and release it at the condensing end. The vaporization of the working fluid in the thermosyphon results in the absorption of a large quantity of heat from the evaporator and an increased pressure at the evaporator end. The heat is absorbed because energy is required to separate the molecules which are in contact in the liquid state. The increase in pressure is produced by the thermal excitation of the molecules comprising the newly created vapor. The

pressure difference set up by the vapor pressure causes the vapor, and therefore, the heat energy, to move toward the condenser section. The vapor comes in contact with a lower temperature in the condensing section and condenses. In so doing, it releases the thermal energy and the vapor pressure decreases. This pressure decrease gives a pressure difference which maintains vapor and heat flow.

The vapor stores heat energy in the form of enthalpy at a temperature at which it was formed and it retains it at that temperature till it meets a colder surface. This means that the temperature along the entire length of the thermosyphon tube theoretically remains constant.

The condensate returns back to the evaporator under the influence of gravity alone in case of a stationary thermosyphon. The use of gravity requires that the evaporator be always located below the condenser.

3.2 Choice of a working fluid

The choice of a working fluid in a thermosyphon is dependent on several physical properties of the fluid. The most significant of them are vapor pressure, latent heat of vaporization, and surface tension. Other important properties are thermal conductivity, viscosity, and density of vapor and liquid.

The vapor pressure curve of the working fluid dictates the temperature range of its applicability. Generally, a fluid should be used in a steeply sloped region of its vapor pressure curve so that the temperature in the thermosyphon tube does not vary appreciably even though the pressure does.

Fig. 3.1 and 3.2 show , for example, the vapor pressure curves for water and Freon-11, the two working fluids that were used in the present investigation. In addition, the vapor pressure should also be reasonably high because a low vapor pressure would result in low vapor densities and relatively large pressure drops in the moving vapor.

A high latent heat of vaporization is desirable because the mass flow rate required to transfer a given amount of heat is inversely proportional to the latent heat. A high conductivity in the liquid phase and low surface tension are other desirable physical properties.

3.3 An analysis : maximum heat transfer for an ideal case:

We shall now analyse the factors governing the maximum heat transfer capability of a two phase closed thermosyphon circular tube. Among the considerations by which the rate of heat transfer through a two phase closed thermosyphon may be limited are the following.

1. The maximum heat fluxes that may be tolerated in the evaporator and condenser sections.
2. The ability of the gravity force to keep the evaporating-condensing fluid in circulation.

The work reported here is concerned with two phase closed thermosyphons limited by the last of these two considerations..

The mathematical model is based upon the idealized two phase closed thermosyphon configuration shown in the Fig. 3.3 The heat transfer capability of a two phase closed thermosyphon depends on the following basic flow parameters.

1. force due to the gravity head (ΔE_g)
2. force due to the vapor pressure drop (ΔP_v)
3. force due to the liquid viscous drag (ΔP_f)
4. force due to the pressure drop caused by the momentum changes (ΔP_m)

From a pressure balance criterion alone, the equation for thermosyphon operation is given by :

$$\Delta P_q \geq \Delta P_v + \Delta P_l + \Delta P_m \quad \text{-----}(3.1)$$

If the thermosyphon is operated above the limit set by Eq.(3.1), the evaporator does not receive enough working fluid to absorb the incoming heat energy. Consequently, the temperature of the evaporator section rises rapidly and undesirably.

The various pressure losses in Eq. (3.1) depends upon the properties of the working fluid such as heat of vaporization, density, surface tension, etc.

The vapor after condensation may return to the evaporator either as a film flowing down the tube along its inside circumference or as a series of closely spaced droplets of liquid, depending upon the complex situation and the liquid. In the present analysis we have assumed the ideal situation where condensate returns as a continuous film, flowing down the tube along its circumference.

We will now develop expressions for the pressure drop terms appearing in the Eq. (3.1) for a two phase closed thermosyphon circular tube with the following assumptions.

1. The system may be treated as one dimensional.
2. The liquid flow in the boundary layer is laminar.
3. The temperature in the condensing section is constant along its length.

4. Heat is supplied uniformly to the evaporator section.
5. All the heat added at the evaporator section is removed at the condensing end.
6. The condensate returns to the evaporator as a continuous film flowing down the tube along its circumference.

The coordinates adopted are shown in the Fig. (3.3). Considering a small element of the condensate in the condensing section, as shown in the Fig. 3.3, we have,

$$\Delta P_g = \pi (r^2 - r_i^2) \frac{\rho g dz}{g_c} \quad \text{-----}(3.2)$$

$$\Delta P_t = -2\pi r \mu \frac{du}{dr} dz \quad \text{-----}(3.3)$$

The pressure drops caused by vapor flow and that due to momentum changes are considered negligible in a thermosyphon tube. Also effects of gravity on the vapor flow are neglected since the density of the water vapor is negligible compared to that of the liquid water.

Substituting the Eq. (3.2) and Eq. (3.3) in the Eq. (3.1) and simplifying, we have,

$$G = \rho g \frac{B}{g_c} \left[\frac{1}{4} - y_i^2 + \frac{3}{4} y_i^4 - y_i^4 \ln y_i \right] \quad \text{-----}(3.4)$$

where,

$$y_i = \frac{r_i}{R_i}$$

and

$$B = \left(\rho \pi R_i^4 \right) / 2\mu$$

Also from assumption (5), heat addition at the evaporator section equals heat removal at the condensing section, which gives,

$$q \ 2 \pi R L_h = \int_0^{L_c} \frac{2 \pi K (T_s - T_c)}{\ln(1/y_i)} dz \quad (3.5)$$

From Eq.(3.4) and Eq. (3.5), we have,

$$q R L_h = \frac{L_c K (T_s - T_c) \left[-y_i'^2 + \frac{3}{8} y_i'^4 - \frac{y_i'^4}{2} \ln y_i' + \frac{1}{8} \right]}{\left[y_i'^4 \ln y_i' \left(\frac{\ln y_i'}{2} - \frac{1}{2} \right) + \frac{y_i'^4}{8} + y_i'^2 \left(\frac{\ln y_i'}{2} - \frac{1}{4} \right) + \frac{1}{8} \right]} \quad (3.6)^*$$

Eq. (3.6) gives the maximum heat transfer rate for a two phase closed thermosyphon circular tube of given heated and cooled lengths at a given operating pressure for the idealized model used in this analysis.

The maximum heat transfer rate as a function of saturation temperature, as obtained by Eq. (3.6), for a two phase closed thermosyphon circular tube of I.D. 1.062 in. is shown in the Figures 3.4 , 3.5 , and 3.6 for heated length-cooled length ratios of 2.0, 1.25, and 0.8 respectively, using water as the working fluid and Fig. 3.7 , using Freon-11 as the working fluid.

* The derivation of Eq. (3.6) is given in Appendix 1

3.4 Dimensional analysis

As explained earlier, it is extremely difficult to obtain a vigorous theoretical analysis for a two phase closed thermosyphon. However, a qualitative picture may be obtained by dimensional analysis.

The tube heat transfer coefficient, h , may depend upon the following factors.

$$h = f(\rho, \theta, D, g, \kappa, c_p, \mu, \beta, L_h, L_c, v_l)$$

Using Buckingham π theorem, it may be shown that,

$$Nu = f\left(v^+, \frac{L_h}{D}, R, Pr, Gr\right) \text{ ----- (3.7) }^*$$

where,

$$Nu = \frac{h D}{\kappa}$$

$$Pr = \frac{c_p \mu}{\kappa}$$

$$Gr = \frac{\rho^2 g D^3 \theta \beta}{\mu^2}$$

$$R = \frac{L_h}{L_c}$$

and

$$\theta = (T_h - T_c)$$

The dimensional analysis shows that the tube heat transfer coefficient is dependent upon the quantity of the working fluid, heated length-

* The derivation of Eq. (3.7) is given in Appendix 2.

cooled length ratio, heated length to tube diameter ratio, working fluid, and the properties of the working fluid.

4. EXPERIMENTAL APPARATUS

An experimental apparatus was designed to investigate the heat transfer characteristics of the proposed two phase closed thermosyphon under the controlled laboratory conditions.

4.1 Main heat transfer loop

A schematic diagram of the experimental set up is shown in the Fig. 4.1 and a detailed diagram of the test section in the Fig. 4.2 . A pictorial view is given in the Fig. 4.3 . Main heat transfer loop consisted of a thermosyphon tube, a resistance heater assembly to supply the input thermal energy, a guard heater, number of separate jackets for cooling the upper part of the thermosyphon tube, and a cooling system.

Water and Freon-11 were used as the working fluids. Also water was used as the coolant.

A refrigeration loop as a cooling system was also constructed (the details are given in Appendix 3) to study the performance of the thermosyphon tube at extremely low temperatures. Due to lack of time the study using the refrigeration loop was not carried out in the present investigation.

The thermosyphon used in the present study was made of a 1 in. standard pipe size copper tube. The tube was 54 in. long and 1.062 in. I.D. with a wall thickness of 0.127 in. Closure of the thermosyphon tube was made by screwing a machined end threaded cap into each end.

The lower cap carried a $\frac{1}{4}$ in. copper tube to height of one inch inside the thermosyphon tube serving as a thermocouple well. This well was provided with an Iron-constantan thermocouple.

A $\frac{3}{8}$ in. copper tube was soldered to the upper cap which provided connections to the vacuum pump, charging line of the working fluid, and pressure measuring instruments. All the connections were done either using Swagelok type brass compression fittings or employing silver soldering.

Twelve Iron-constantan thermocouples insulated with magnesia and sheathed in Stainless Steel tube of 0.032 in. O.D. were embedded in the grooves machined on the out side wall of the test tube. The thermocouples were uniformly spaced on its circumference along the length, their positions of locations being shown in the Fig. 4.2 . Provision was made to measure the temperature in the vapor core by means of a traversing Iron-constantan thermocouple probe.

The main requirement in the heat input section, called evaporator, was to have a heat input which could be measured accurately. The system chosen for this purpose was to use two electrical heaters.

1. the main heater
2. the guard heater

The main heater was used to supply the total heat input to the evaporator. The guard heater was employed to eliminate the outward radial heat losses from the main heater. Both the heaters were made

of 15 AWG Chromel-A Nichrome wire with ceramic beads slid on it. The resistance of the heating wire was 0.2001 ohms per linear foot, according to the specifications. Each ceramic bead (Leed and Northrup make, part no. STD-715-APK insulators) was 0.14 in. long and 0.17 in. O.D. with a hole diameter of 0.068 in. The small bead size was chosen so that the beaded heating wire can easily be wound around the thermosyphon tube of 1.316 in. O.D.

The main heater was kept in position with the help of a $\frac{1}{2}$ in. wide fibre glass tape and $\frac{1}{2}$ in. diameter 'Atlas' asbestos rope insulations. Between this rope insulation and the guard heater, a $1\frac{1}{2}$ in. thick asbestos pipe insulation was placed. A 2 in. thick fibre glass insulation covers the guard heater and the entire assembly.

Acromag model 370 relay was used to maintain a constant temperature in the insulation between the two heaters, the relay being operational on the two calibrated Iron-constantan thermocouples embedded in the insulation between the two heaters as shown in the Fig. 4.2 . The main heater was supplied electrical power from a variable transformer rated at 13.4 KVA, 240 V, 56 A and power to the guard heater was controlled by an Acromag relay which operated on 115 V, 60 cycles.

The heat in the test section is removed at the upper part of the thermosyphon tube by a number of separate cooling jackets, each 6 in. long and 3.63 in. O.D. with a wall thickness of 0.1 in. Each jacket has an 'O' rubber ring on either end so that it can easily slide over the

test tube and also provide leak proof ends. This arrangement facilitates changing the heated length-cooled length ratio. Also, each jacket has a separately controllable supply of the coolant so that a uniform temperature may be maintained in the condensing section.

4.2 Instrumentation

Temperatures throughout the system were measured by Iron-constantan thermocouples. All the thermocouples were calibrated up to 600⁰F. The calibration of the thermocouples was carried out against a standard thermometer by measuring the temperature of an oil bath which was maintained at a constant temperature. The cold junction of the thermocouple was kept at the melting point of ice. The output of these thermocouples, during the early stages of the investigation, were recorded on a double pen Hewlett and Packard model 7100 B strip chart recorder and also read, upon reaching the steady state, by a Leeds and Northrup Cat. No. 8687 potentiometer. In the later experiments, the output of the thermocouples was read from a self calibrated Helwitt and Packard DVM model 240 IC, with a resolution of one microvolt. Common cold junction for all the thermocouples was maintained at the melting point of ice. A 16 point Leeds and Northrup rotary thermocouple switch was used for the ease of recording the temperatures from the different thermocouples.

Power to the test section was measured by means of a 1000/5 General Electric Cat.No. 750 x 99 G2 current transformer, calibrated ammeter,

and a Hewlett and Packard model 427 A voltmeter. To provide more sensitivity, an ammeter was directly put in the line for small power inputs.

The flow of the coolant in each jacket could be individually measured by means of an orifice meter provided for each jacket. The orifice meters were precalibrated. The calibration was done with water under a constant head supplied by a centrifugal pump. To remove air and other non condensible gases, a Leybold model S1TEL 147283 mechanical vacuum pump with a capacity of 10^{-1} torr was used. Provision was made to reach an ultimate vacuum of 5×10^{-5} torr by using a Leybold design no. 930827 diffuser pump in conjunction with the mechanical pump. The pressure inside the thermosyphon tube was measured by a vacuum Bourden pressure gauge. A U-tube mercury manometer was also used to read the pressure inside the thermosyphon tube accurately.

4.3 Procedure

Before starting the actual experiments, the thermosyphon tube was thoroughly cleaned by repeated washings with acetone followed by distilled water. It was then vacuum dried, tested for vacuums and leaks checked and corrected. After the system has been evacuated, a known amount of the working fluid was charged. The amount to be charged was kept in a charging tank which was connected to the charging valve by a polythene tube. The charging valve was then opened and the working fluid gets into the tube due to pressure difference. Care was taken to see that no condensible

gases or air remained inside the thermosyphon tube. The power was then supplied to both the main and guard heaters. Power to the main heater was increased in steps upto a desired heat flux. At each power level, the power to the guard heater was automatically controlled by the relay and when a steady state was reached, (within 30-50 minutes) temperature, pressure, and power readings were recorded. While increasing the power supply to the heating coil, great care was necessary because a maximum heat flux, depending upon the quantity of the working fluid in the tube, would be reached above which the wall temperature would begin to rise suddenly and rapidly. An immediate power reduction is then necessary to prevent the tube from burning out.

4.4. Data reduction

The tube heat transfer coefficient, h , was calculated from the definition,

$$h = \frac{q}{(T_h - T_c)}$$

where,

q = local heat input per unit area

T_h = average temperature in the heated section

T_c = average temperature in the cooled section

The heat flux per unit area, q , was calculated from the power measurements which was done from the voltmeter and ammeter coupled into the supply circuit of the main heater. The heat losses through the insulation were calculated from the temperature gradient obtained by a

pair of thermocouples which actuate the relay. The temperature in the insulation between the guard heater and the main heater was maintained constant within 1 to 2⁰F and this gave rise to the heat losses through the insulation which were always found to be less than 0.5 percent and therefore negligible.

The temperature in the heated section and condensing section were measured by calibrated thermocouples. The errors involved in the calculation of the tube heat transfer coefficient was generally due to the accuracy of the temperature and the power measurements. Since the readings were recorded after the steady state has been reached, the error due to delay in reading the measurements was considered negligible. At high heat fluxes, (about 4500 B.T.U. per hr. ft²) the temperatures, particularly in the evaporator section, were fluctuating by $\pm 2^0$ F. In such cases a mean value was taken.

5. RESULTS AND DISCUSSIONS

The range of the variables investigated in the present study are shown in Table 5.1 and the experimental results are described in Tables 5.2 to 5.22. These experimental results enable us to evaluate the various characteristics of heat transfer coefficient for a two phase closed thermosyphon tube.

As noted from the functional relationship obtained by the dimensional analysis in the section 3.4, the heat transfer coefficient depends upon the quantity of the working fluid, V^+ ; the heated length cooled length ratio, R ; heated length to tube diameter ratio, L_h/D ; the saturation temperature (corresponding to the mean operating pressure) T_s ; the heat flux, q ; and the working fluid. Here we will study the effect of each of these parameters, except that of heated length to tube diameter ratio, L_h/D , on the heat transfer coefficient.

To clearly explain the terms, we may define the maximum heat transfer rate and the corresponding maximum heat transfer coefficient used in the present discussion.

For a given operating pressure and a given wall temperature in the condensing section in a thermosyphon tube, there should be a limiting value of heat flux above which the operating pressure inside the thermosyphon tube can not be maintained. This limiting heat flux is termed as the maximum heat transfer rate and the heat transfer

coefficient of the tube at the limiting heat flux as the maximum heat transfer coefficient.

The steady state wall temperature distribution of the thermosyphon tube as well as the vapor core temperatures are shown in the Figs. 5.1 to 5.5 as examples for different values of V^+ , R , T_s , q and Pr . They provide useful information for the purpose of designing or choosing a thermosyphon tube for a particular application. Another useful design requirement may be the temperature drop between the heated section and the cooled section of a thermosyphon tube for a given maximum heat transfer coefficient. They are shown in Figs. 5.6, 5.7 and 5.8 for the ratio R of 2, 1.25, and 0.8 respectively using water as the working fluid and in Fig. 5.9 using Freon-11 as the working fluid.

5.1 Effect of the quantity of the working fluid, V^+ .

The maximum heat transfer coefficient as a function of saturation temperature, corresponding to the operating pressure inside the thermosyphon tube, are shown in Figs. 5.10, 5.11 and 5.12 for the ratio R of 2, 1.25, and 0.8 respectively, and for different quantities of water. From these figures, the variation of the maximum heat transfer coefficient with the quantity of the working fluid is obtained as given in Fig. 5.13. Similarly, the behavior of maximum heat transfer rate with increase in the quantity of water, obtained from Figs. 5.14, 5.15, is shown in Figs. 5.16 and 5.17 for the ratio R of 2 and 1.25 respectively.

As may be noted from Figs. 5.13, 5.16, and 5.17, both the maximum heat transfer coefficient and the maximum heat flux increase with an increase in the amount of the working fluid upto a certain quantity of the working fluid and then become independent of V^+ , within the range of V^+ studied. The quantity of the working fluid at which the maximum heat flux becomes independent of the increase in V^+ is termed as the minimum quantity of the working fluid. Cohen and Bayley⁽¹⁶⁾ also reported the similar trends of V^+ on the heat transfer rate.

This phenomena may be explained by the fact that so long as the quantity of the working fluid in the thermosyphon tube is sufficient to fill the space with saturated vapor and wet the evaporating and condensing surfaces, the heat flow circuit will operate satisfactorily. The quantity of the working fluid should be sufficient so that the heat at the evaporator section may all be utilized in its evaporation. Condensation of the vapor in the cooled section and evaporation of the condensate in the heated section makes it a closed cycle. It follows that as the rate of heat flow increases so does the quantity of the working fluid in the circulation. If the rate of heat flow exceeds that required to maintain the heat flow circuit, the heat input is in excess of that which can be utilized in evaporating the given quantity of the working fluid. Then the temperature of the heated section begins to rise rapidly and undesirably. This situation may be termed as the "dry out". An immediate reduction of the heat input is then necessary to prevent the burning out. A dry out occurred at a heat input of 2.25 KW

for the ratio R of 1.25 when the percentage quantity of water (V^+) was 0.0103.

The minimum quantity of the working fluid decreases with decrease in R , as observed in Fig. 5.13. A decrease in R means that the cooled length increases and the heated length decreases. The increasing cooled length gives an efficient condenser which, in turn, should have reduced the minimum quantity of the working fluid for the same heat input.

5.2 Effect of the heated length-cooled length ratio, R

The maximum heat transfer coefficient increases with decrease in the heated length-cooled length ratio, R , as shown in Figs. 5.18, and 5.19. Bayley and Lock⁽⁹⁾ reported similar effects in their study on a single phase stationary closed thermosyphon tube. Unfortunately no previous experimental or theoretical results are available with which the effect of this parameter may be compared. The increase in the maximum heat transfer coefficient for decreasing values of R would be accounted by the same reasoning as given in the last paragraph of the section 5.1.

5.3 Effect of mean operating pressure, p (or the corresponding saturation temperature, T_s)

The heat transfer coefficient increases appreciably with the increase in the mean operating pressure inside the thermosyphon tube as shown in Fig. 5.20 and this was expected as the boiling heat transfer coefficient increases with increasing pressure.

The temperature drop within the working fluid along the length of the thermosyphon tube depends upon the pressure drop along the tube and on the gradient of the saturation pressure-temperature relationship of the working fluid. As the pressure increases, the vapor specific volume decreases resulting in a decrease in the pressure drop, because the mass flow rate of vapor is essentially constant for transferring a given heat input. The slope of the saturation pressure-temperature relationship, both for water and Freon-11 as shown in Figs. 3.1 and 3.2 respectively, increase rapidly with increase in pressure and therefore, give small temperature difference at high pressure. Both of these factors give rise to a decreasing temperature drop within the working fluid which, in turn, increases the heat transfer coefficient. Also the reduction in the pressure drop, as the pressure inside the thermosyphon tube increases, allows condensate to move fast from the condensing section, which increases the heat transfer coefficient.

In the present investigation, a heat flux of $22,400 \text{ B.T.U./hr.ft}^2$ (70.6 KW/meter^2) was transferred for the thermosyphon tube of 1.062 in. I.D. at a pressure of 3.97 psia. without reaching the burning out situation. Cohen and Bayley⁽¹⁶⁾ observed a critical heat flux (heat flux at which the burning out of the tube occurs is referred to as the critical heat flux) of $40,000 \text{ B.T.U./hr.ft}^2$ at atmospheric pressure for a rotating thermosyphon tube of $\frac{1}{4}$ in. I.D., and of $73,500 \text{ B.T.U./hr.ft}^2$ at a pressure of 190 psig. However, the critical heat flux can not be

increased beyond a certain limit even by increasing the operating pressure. This limit is imposed due to the restriction set on the vapor bubble movement by the narrow confines of the tube, so that at very high rates of heat flow, the liquid at certain sections is completely displaced by the vapor, at which points the cooling fails and the wall temperature rises rapidly.

5.4 Effect of heat flux, q

The maximum heat transfer coefficient for water increases nearly linearly with increase in the maximum heat flux, within the range investigated. Their variations are given in Figs. 5.21, 5.22, and 5.23 for the ratio R of 2, 1.25 and 0.8 respectively. The heat transfer coefficient at comparatively low pressures increases appreciably with heat flux as noted from Figs. 5.21, 5.22, and 5.23. However, at high pressures, the heat transfer coefficient is less sensitive to heat flux as shown in Fig. 5.24. In Fig. 5.25 is shown the plot of maximum heat transfer coefficient with maximum heat flux when Freon-11 is used as the working fluid. The maximum heat transfer coefficient for Freon-11 becomes constant at a heat flux of about $2000 \text{ B.T.U./hr.ft}^2$ and higher. From these results, it is noted that the heat transfer coefficient has a tendency of becoming less dependent of heat flux at high pressures.

Figs. 5.26, 5.27, and 5.28 show the variation of both theoretical and experimental maximum heat transfer rate with the saturation temperature, corresponding to the mean operating pressure in the

thermosyphon tube. Although the theory predicts the trend very closely, it over-estimates the actual results for water by a factor of 1.6 to 2.0 depending upon the value of R . However, it under-estimates the maximum heat flux for a given mean operating pressure when Freon-11 is used as the working fluid, as shown in the Fig. 5.29. This discrepancy may be due to the idealized assumptions made in the analysis discussed in the section 3.3.

The steady state temperature distribution of the thermosyphon tube for different values of q , as given in Fig. 5.4, shows increasing temperatures in the evaporator section and decreasing temperatures in the condensing section for a given mean operating pressure.

Since no other experimental results are available with which the present findings could be compared, it may not only be of interest to compare the present results with that of a single phase stationary closed thermosyphon tube reported by Bayley and Lock⁽⁹⁾, but also it gives a measure of the heat transfer capability of a two phase closed stationary thermosyphon tube taking a single phase thermosyphon as a reference point. The present results are also compared with that of an open thermosyphon, predicted theoretically by Lighthill⁽¹⁹⁾. These comparisons are shown in Fig. 5.30.

It may easily be concluded that the performance of a two phase closed thermosyphon is far greater than that of either an open thermosyphon or a single phase closed thermosyphon. The high performance

of a two phase closed thermosyphon is due to the fact that it uses heat transfer mechanism of phase change, namely boiling and condensation, which are efficient means of heat transfer. A single phase closed thermosyphon, on the other hand, entirely depends on free convection alone.

5.5 Effect of working fluid, Pr

In addition to water, Freon-11 was also used as a working fluid in the thermosyphon tube to study the effects of its physical properties on the heat transfer capability of the thermosyphon tube. A comparison is made between the two working fluids as shown in Fig. 5.31.

As expected, the maximum heat transfer coefficient with Freon-11 was small as compared to that of water, within the range investigated. This may have been due to the low latent heat of evaporation and low thermal conductivity of Freon-11 compared to that of water. Water, in addition to having a high latent heat of evaporation and high thermal conductivity, is non toxic, chemically inert if no impurity is present, and above all, available in abundance.

However, Freon-11 may be a useful working fluid for extremely low temperature operations where water can not be used. Investigation of Freon-11 as the working fluid in the low temperature range is, therefore, recommended as a further study.

6. CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER STUDY

In the present work, an experimental investigation of the heat transfer performance of a two phase closed thermosyphon together with a simple theoretical analysis for its maximum heat transfer capacity, has been described. Water and Freon-11 were used as the working fluids. Varying the different parameters while maintaining, as far as possible, a uniform temperature along the cooled wall and a uniform heat flux along the heated wall, results have been obtained from which the following conclusions may be drawn:

1. The heat transfer coefficient is not sensitive to the quantity of the working fluid above a certain amount of the working fluid. The minimum quantity of the working fluid decreases with decrease in R , within the range studied.
2. Decreasing the ratio R led to a consistently increasing heat transfer coefficient.
3. The heat transfer coefficient is very sensitive to the operating pressure and rapidly increases with increasing pressures.
4. Water was found superior to Freon-11 as the working fluid under a given condition.
5. A theoretical analysis for the maximum heat transfer rate predicts the trends closely but over-estimates the actual results for water and under-estimates those for Freon-11.

6. A dry out was observed at a heat input of 2.25 KW for the ratio R of 1.25 and $V^+ = 0.0103$.

The results obtained in the present investigation are encouraging, as the simple sealed tube form of thermosyphon has a high heat transfer capacity for many engineering applications. However, there is still lack of information on many other aspects of the two phase closed thermosyphon tube at the moment. Because of lack of time, the author could not investigate the effect of the heated length to tube diameter ratio and performance of the thermosyphon tube in the low temperature range. It is, therefore, recommended that a further study be made of this system, particularly in the following areas:

1. The effect of the heated length to tube diameter ratio on the various heat transfer characteristics .

2. The performance of the system in very low temperature ranges (nearly 30°F). For this investigation, it is suggested that the refrigeration system constructed in the present study should be used with the modifications as described in the appendix 3.

APPENDIX 1 ANALYSIS FOR MAXIMUM HEAT TRANSFER RATE FOR AN IDEAL CASE

In this appendix, an expression for the maximum heat transfer rate for a two phase closed thermosyphon circular tube is developed. The various assumptions made in its derivation are stated in the section 3.3.

The basic flow parameters, upon which depends the heat transfer rate of a two phase closed thermosyphon, are as follows.

1. force due to the gravity head (ΔP_g)
2. force due to the vapor pressure drop (ΔP_v)
3. force due to the liquid viscous drag (ΔP_l)
4. force due to the pressure drop caused by the momentum changes (ΔP_m)

For the satisfactory performance of the thermosyphon tube, the available gravity head must be sufficient to overcome the pressure losses caused by the vapor transport, by the viscous drag of the returning liquid, and by the momentum changes, so that the evaporating-condensing fluid can be kept in circulation. This criterion gives the following equation for the operation of a two phase closed thermosyphon.

$$\Delta P_g \geq \Delta P_v + \Delta P_l + \Delta P_m \quad \text{--- (A.1.1)}$$

The mathematical model based upon the idealized continuous film condensation is shown in the Fig. (3.3). Considering the forces acting on an infinitesimal element of the liquid film in the condensing

section as shown in the Fig. 3.3 , we have,

$$\Delta P_f = \pi (r^2 - r_i^2) f dz \quad \text{-----(A.1.2)}$$

$$\Delta P_i = 2 \pi r \mu \left(\frac{du}{dr} \right) dz \quad \text{-----(A.1.3)}$$

Since the pressure drops due to vapor flow and the momentum changes are considered negligible, the Eq. (A.1.2) and the Eq. (A.1.3) when substituted in the Eq. (A.1.1) and integrated gives:

$$G = \frac{\rho}{g} B \left[\frac{1}{4} - y_i^2 + \frac{3}{4} y_i^4 - y_i^4 \ln y_i \right] \text{---(A.1.4)}$$

where,

$$y_i = \frac{r_i}{R}$$

and

$$B = \frac{\rho \pi R^4}{g 2 \mu}$$

Eq. (A.1.4) is identical to the equation for the mass flow rate of a condensing film running down the inside circumference of a vertical tube at a constant wall temperature as given in the Reference (18).

Also the heat removed at the wall over the length dz , should equal the increamental mass flow over the same length dz times the

section as shown in the Fig. 3.3, we have,

$$\Delta P_g = \pi (r^2 - r_i^2) \rho \frac{q}{g_c} dz \quad \text{---(A.1.2)}$$

$$\Delta P_l = -2\pi r \mu \left(\frac{du}{dr} \right) dz \quad \text{---(A.1.3)}$$

Since the forces due to vapor pressure drop and that due to pressure drop caused by momentum changes are considered negligible, Eq. (A.1.2) and Eq. (A.1.3) when substituted in the Eq. (A.1.1) gives:

$$\pi (r^2 - r_i^2) \rho \frac{q}{g_c} dz = -2\pi r \mu \left(\frac{du}{dr} \right) dz$$

$$du = + \frac{\rho q}{2\mu g_c} \int_r^{R_i} \frac{(r^2 - r_i^2)}{r} dr$$

$$u = - \frac{\rho q}{2\mu g_c} \left[\frac{r^2 R_i^2}{2} - r_i^2 \ln \frac{r}{R_i} \right]$$

The average velocity is then given by u_m as:

$$u_m = \frac{\int_{r_i}^{R_i} u \cdot 2\pi r dr}{\pi (R_i^2 - r_i^2)}$$

$$u_m = - \frac{\rho q}{\mu (R_i^2 - r_i^2) g_c} \int_{r_i}^{R_i} \left[\frac{r^2 R_i^2}{2} - r_i^2 \ln \frac{r}{R_i} \right] r dr \quad \text{---(A.1.3a)}$$

Introducing the variables, $y = \frac{r}{R_i}$ and $y_i = \frac{r_i}{R_i}$; the Eq. (A.1.3a) may be re-arranged as follows:

$$U_m = \frac{\rho \cdot \pi R_i^4 q}{\pi R_i^2 (1 - y_i^2) 2 \mu g_c} \int_{y_i}^1 [1 - y^2 + y_i^2 \ln y^2] y dy$$

Integrating, we get,

$$U_m = \frac{\rho B}{\pi R_i^2 (1 - y_i^2) g_c} \left[\frac{1}{4} - y_i^2 + \frac{3}{4} y_i^4 - y_i^4 \ln y_i \right]$$

where,

$$B = \frac{\rho \pi R_i^4}{2 \mu}$$

The mass flow rate of the condensate is now given by G as follows:

$$G = \rho \pi (R_i^2 - r_i^2) U_m$$

$$G = \rho q \frac{B}{g_c} \left[\frac{1}{4} - y_i^2 + \frac{3}{4} y_i^4 - y_i^4 \ln y_i \right] \text{----(A.1.4)}$$

Eq.(A.1.4) is identical to the equation for the mass flow rate of a condensing film running down the inside circumference of a vertical tube at a constant wall temperature as given in the Reference (18).

Also the heat removed at the wall over the length dz , should equal the incremental mass flow over the same length dz times the latent heat

of condensation of the vapor. Thus,

$$dG \cdot h_{fg} = \frac{2 \pi K (T_s - T_c)}{\ln\left(\frac{R_i}{r_i}\right)} dz$$

$$\text{or, } h_{fg} dG = \frac{2 \pi K (T_s - T_c)}{\ln\left(\frac{1}{y_i}\right)} dz \quad \text{---(A.1.5)}$$

Substituting the value of dG from the Eq. (A.1.4) in the Eq. (A.1.5) and integrating, we have,

$$\frac{2 \pi K \mu (T_s - T_c) g_c}{p^2 R_i^4 h_{fg} \gamma} z = y_i^4 \ln y_i \left(\frac{\ln y_i}{2} - \frac{1}{2} \right) + \frac{y_i^4}{8} + y_i^2 \left(\frac{\ln y_i}{2} - \frac{1}{4} \right) + \frac{1}{8} \quad \text{---(A.1.6)}$$

Also heat supplied to the heated section equals the heat removal at the condensing section, that is;

$$q \cdot 2 \pi R_i L_h = \int_0^{L_c} \frac{2 \pi K (T_s - T_c)}{\ln\left(\frac{1}{y_i}\right)} dz \quad \text{---(A.1.7)}$$

Differentiating the Eq. (A.1.6) w.r.t. z , we have,

$$\frac{dy_i}{dz} = \frac{2 K \mu (T_s - T_c) g_c}{q \cdot p^2 R_i^4 h_{fg} \ln y_i (y_i - y_i^3 + 2 y_i^3 \ln y_i)} \quad \text{---(A.1.8)}$$

From the Eq. (A.1.6) and the Eq. (A.1.8), we have,

$$\frac{dy_i}{dz} = \frac{\left[y_i^4 \ln y_i \left(\frac{\ln y_i}{2} - \frac{1}{2} \right) + \frac{y_i^4}{8} + y_i^2 \left(\frac{\ln y_i}{2} - \frac{1}{4} \right) + \frac{1}{8} \right]}{L_c \ln y_i (y_i - y_i^3 + 2 y_i^3 \ln y_i)} \quad \text{---(A.1.9)}$$

where, $y'_i = 1 - \frac{\delta}{R}$

and δ is the liquid film thickness at $z = L_c$

To integrate the right hand side of the Eq.(A.1.7), we use the Eq.(A.1.9) as follows:

$$\int_0^{L_c} \frac{2\pi K(\tau_s - \tau_c)}{\ln(y_i)} dz = \frac{2\pi K(\tau_s - \tau_c) L_c}{\left[y_i'^4 \ln y_i' \left(\frac{\ln y_i'}{2} - \frac{1}{2} \right) + \frac{y_i'^4}{8} + y_i'^2 \left(\frac{\ln y_i'}{2} - \frac{1}{4} \right) + \frac{1}{8} \right]} \int_1^{y_i'} \frac{\ln y_i (y_i - y_i'^3 + 2y_i'^3 \ln y_i) dy_i}{\ln(y_i)} \quad \text{---(A.1.10)}$$

From the Eq.(A.1.7) and the Eq.(A.1.10), we have,

$$q R L_h = \frac{L_c K(\tau_s - \tau_c) \left[-\frac{y_i'^2}{2} + \frac{3}{8} y_i'^4 - \frac{y_i'^4}{2} \ln y_i' + \frac{1}{8} \right]}{\left[y_i'^4 \ln y_i' \left(\frac{\ln y_i'}{2} - \frac{1}{2} \right) + \frac{y_i'^4}{8} + y_i'^2 \left(\frac{\ln y_i'}{2} - \frac{1}{4} \right) + \frac{1}{8} \right]} \quad \text{---(A.1.11)}$$

Also from the Eq.(A.1.11) and the Eq.(A.1.6), we have,

$$\frac{2\mu q R L_h}{\rho^2 R^4 h_{fg}} = \left[-\frac{y_i'^2}{2} + \frac{3}{8} y_i'^4 - \frac{y_i'^4}{2} \ln y_i' + \frac{1}{8} \right] \quad \text{---(A.1.12)}$$

The Eq.(A.1.11) is the same as the Eq.(3.6) in the section 3.3.

where, $y_i' = 1 - \frac{\delta}{R_i}$

and δ is the liquid film thickness at $z = L_c$

To integrate the right hand side of the Eq.(A.1.7), we use the Eq.(A.1.9) as follows:

$$\int_0^{L_c} \frac{2\pi k(T_s - T_c)}{\ln(y_i)} dz = \frac{2\pi k(T_s - T_c) L_c}{\left[y_i'^4 \ln y_i' \left(\frac{\ln y_i'}{2} - \frac{1}{2} \right) + \frac{y_i'^4}{8} + y_i'^2 \left(\frac{\ln y_i'}{2} - \frac{1}{4} \right) + \frac{1}{8} \right]} \int_1^{y_i'} \frac{\ln y_i (y_i - y_i^3 + 2y_i^3 \ln y_i) dy_i}{\ln(y_i)} \quad \text{---(A.1.10)}$$

From the Eq.(A.1.7) and the Eq.(A.1.10), we have,

$$q R_i L_h = \frac{L_c k (T_s - T_c) \left[-\frac{y_i'^2}{2} + \frac{3}{8} y_i'^4 - \frac{y_i'^4}{2} \ln y_i' + \frac{1}{8} \right]}{\left[y_i'^4 \ln y_i' \left(\frac{\ln y_i'}{2} - \frac{1}{2} \right) + \frac{y_i'^4}{8} + y_i'^2 \left(\frac{\ln y_i'}{2} - \frac{1}{4} \right) + \frac{1}{8} \right]} \quad \text{---(A.1.11)}$$

Also from the Eq.(A.1.11) and the Eq.(A.1.6), we have,

$$\frac{2\mu q R_i L_h g_c}{\rho^2 R_i^4 h_{fg} g} = \left[-\frac{y_i'^2}{2} + \frac{3}{8} y_i'^4 - \frac{y_i'^4}{2} \ln y_i' + \frac{1}{8} \right] \quad \text{---(A.1.12)}$$

The Eq.(A.1.11) is the same as the Eq.(3.6) in the section 3.3.

To calculate q for a given T_s , Eqs.(A.1.11) and (A.1.12) are used. First, from Eq. (A.1.12) heat flux is calculated for an assumed value of y_i' , taking the values of k , μ , ρ , and h_{fg} at a given constant condensing wall temperature. T_s is then calculated from Eq. (A.1.11). k , μ , ρ , and h_{fg} are again calculated at the film temperature $(T_s + T_c)/2$

With these modified values of k, μ, ρ , and h_{fg} ; heat flux q and saturation temperature T_s are again calculated using the same Eqs. (A.1.11) and (A.1.12). The procedure is repeated till the same values of T_s , within a desired accuracy, are obtained.

APPENDIX 2 DIMENSIONAL ANALYSIS APPLIED TO A TWO PHASE THERMOSYPHON

In this appendix, the derivation of the Eq.(3.7) is described, using Buckingham's π theorem.

The tube heat transfer coefficient, h , which has been defined as

$$h = \frac{q}{(T_h - T_c)}$$

may be a function of the following quantities.

$$h = f(\rho, \theta, D, q, K, \mu, \beta, L_h, L_c, V_e, V_t) \quad \text{--- (A.2.1)}$$

The dimensions of the quantities appearing in the Eq.(A.2.1) are given in the following table.

<u>Quantity</u>	<u>Symbol</u>	<u>Dimensions</u>
Diameter	D	L^1
Density	ρ	$M^1 L^{-3}$
Acceleration due to gravity	g	$L^1 T^{-2}$
Temperature difference	θ	Θ^1
Heated Length	L_h	L^1
Cooled length	L_c	L^1
Viscosity	μ	$M^1 L^{-1} T^{-1}$
Coefficient of the thermal expansion	β	Θ^{-1}
Thermal conductivity	K	$M^1 L^1 T^{-3} \Theta^{-1}$
Specific heat	c_p	$L^2 T^{-2} \Theta^{-1}$
Tube heat transfer coefficient	h	$M^1 L^0 T^{-3} \Theta^{-1}$

According to the Buckingham's π theorem, the Eq. (A.2.1) may be written as follows,

$$0 = f(\rho, \theta, D, q, k, c_p, h, \mu, \beta, L_h, L_c, V_l, V_t) \quad \text{-- (A.2.2)}$$

Since the Eq. (A.2.2) has four dimensions (M, L, T, and Θ), four fundamental quantities will be chosen which will then be used as the repeated variables.

As may clearly be seen, the three π parameters are given as follows.

$$\pi_1 = V_l / V_t = V^+$$

$$\pi_2 = \frac{L_h}{L_c}$$

$$\pi_3 = \frac{L_h}{D}$$

Choosing ρ, θ, D , and q as the fundamental quantities and applying Buckingham's π theorem, we have,

$$\pi_4 = \rho^{a_1} \theta^{b_1} D^{c_1} q^{d_1} h$$

$$\pi_5 = \rho^{a_2} \theta^{b_2} D^{c_2} q^{d_2} k$$

$$\pi_6 = \rho^{a_3} \theta^{b_3} D^{c_3} q^{d_3} c_p$$

$$\pi_7 = \rho^{a_4} \theta^{b_4} D^{c_4} q^{d_4} \mu$$

$$\pi_8 = \rho^{a_5} \theta^{b_5} D^{c_5} q^{d_5} \beta$$

By substituting in the dimensions for π_4 parameter,
we have,

$$\pi_4 = (M L^{-3})^{a_1} (\Theta)^{b_1} (L)^{c_1} (L T^{-2})^{d_1} M T^{-3} \Theta^{-1} = M^0 L^0 T^0 \Theta^0 \quad \text{--- (A.2.3)}$$

Equating the exponents of each dimension which must be the same
on both sides of the Eq. (A.2.3), we have,

$$\text{For } M, \quad a_1 + 1 = 0$$

$$\text{For } L, \quad -3a_1 + c_1 + d_1 = 0$$

$$\text{For } T, \quad -2d_1 - 3 = 0$$

$$\text{For } \Theta, \quad b_1 - 1 = 0$$

Solving the above equations, we get,

$$a_1 = -1$$

$$b_1 = 1$$

$$c_1 = -\frac{3}{2}$$

$$d_1 = -\frac{3}{2}$$

Thus, we get,

$$\pi_4 = \frac{\theta h}{\rho (Dq)^{3/2}}$$

Similarly, we obtain,

$$\pi_5 = \frac{\theta K}{\rho D^{5/2} q^{3/2}}$$

$$\pi_6 = \frac{\theta C_p}{D q}$$

$$\pi_7 = \frac{\mu}{D^{3/2} q^{1/2} \rho}$$

$$\pi_8 = \theta \beta$$

The functional relationship is now given as,

$$\frac{\theta h}{\rho (D q)^{3/2}} = f \left[v^+, \frac{L_h}{L_c}, \frac{L_h}{D}, \frac{\theta K}{\rho D^{5/2} q^{3/2}}, \right.$$

$$\left. \frac{\theta C_p}{q D}, \frac{\mu}{\rho q^{1/2} D^{3/2}}, \theta \beta \right]$$

Since any of the π parameters may be inverted or raised to any power without affecting their dimensionless status, we

re-arrange the π parameters as follows,

$$\frac{\theta h}{\rho (Dq)^{3/2}} \frac{\theta k}{\rho D^{5/2} q^{3/2}} = \frac{h D}{k} = Nu$$

$$\frac{\theta c_p}{q D} \frac{\mu}{q^{1/2} \rho D^{3/2}} \frac{\theta k}{\rho D^{5/2} q^{3/2}} = \frac{c_p \mu}{k} = Pr$$

$$\frac{\theta \beta}{\frac{\mu^2}{q \rho^2 D^3}} = \frac{q \rho^2 D^3 \theta \beta}{\mu^2} = Gr$$

Hence we may re-write the non-dimensional correlation as follows.

$$Nu = f \left(v^+, R, \frac{L_h}{D}, Pr, Gr \right) \text{ ---- (A.2.4)}$$

The Eq. (A.2.4) is the same as the Eq. (3.7) in the section 3.4.

APPENDIX 3 REFRIGERATION CIRCUIT AS A COOLING SYSTEM

In this appendix, the details of the refrigeration circuit, constructed for the investigation on the performance of a two phase closed thermosyphon tube in very low temperature range (nearly 30°F), are given. Fig. A.3.1 gives the various components of the refrigeration system and the details of each part.

The refrigeration loop, which was to be used to transfer heat from the coolant, was designed upto a capacity of approximately 10 KW. It is suggested to make a by pass for the coolant so that the refrigeration system may be used for any heat transfer rate while any desired coolant temperature may still be maintained. A temperature controller may also be installed in the coolant tank for precisely controlling the coolant temperature.

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TABLE 5.1

Range of the variables studied in the present investigation

WATER									
R=2	v^+	0.0154	0.0231	0.0513	0.0770	0.0962	0.180	0.285	
	q	upto 11000	upto 11000	upto 11000	upto 10600	upto 10800	upto 11000	upto 11000	
	p	upto 1.82	upto 1.83	upto 1.83	upto 1.71	upto 1.72	upto 1.83	upto 1.82	
R=1.25	v^+	0.0103	0.0154	0.0231	0.0321	0.0513	0.077	0.0962	0.180 0.285
	q	upto 7300	upto 11000	upto 11000	upto 12700	upto 12300	upto 13200	upto 12700	upto 12700 13000
	p	upto 0.97	upto 1.2	upto 1.22	upto 1.41	upto 1.37	upto 1.35	upto 1.43	upto 1.41 1.41
R=0.8	v^+	0.0513	0.180	0.180					
	q	upto 16500	upto 15300	upto 22400					
	p	upto 1.40	2.62	3.97					
FREON-11									
R=0.8	v^+	0.068	0.109						
	q	upto 5500	upto 5500						
	p	15 to 30	15 to 30						

TABLE 5.2

Working fluid Water

Quantity of the working fluid (V^+) = 0.0154Heated length-cooled length ratio (R) = 2.0

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	89.6	72.9	63.1	72.3	67.4	9.8	5.5	9.16	2.34	17180.53
2	180.3	76.8	63.2	74.4	69.7	13.6	7.1	13.23	3.35	27474.82
3	339.2	78.8	63.3	76.1	71.5	15.6	7.3	21.78	5.50	33635.02
4	654.2	83.0	63.3	79.0	75.0	19.7	8.0	33.27	8.34	48564.15
5	1006.0	87.0	63.3	82.4	78.4	23.8	8.7	42.32	10.54	66638.53
6	2061.3	92.2	63.4	87.2	82.7	28.8	9.5	71.52	17.67	94061.81
7	4089.0	102.8	63.4	95.9	92.3	39.4	10.5	103.90	25.28	172306.44
8	6086.1	114.1	63.6	104.8	102.4	50.5	11.7	120.63	28.92	295037.20
9	9200.3	128.7	64.1	120.0	115.9	64.6	12.8	142.43	33.58	529550.44
10	10997.9	135.7	67.1	126.1	122.3	68.6	13.5	160.42	37.56	650891.09

TABLE 5.3

Working fluid Water

Quantity of the working fluid (V^+) = 0.0231Heated length-cooled length ratio (R) = 2.0

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	89.6	71.0	63.1	69.5	66.9	7.9	4.2	11.31	2.90	12984.07
2	180.3	74.8	63.2	72.6	69.2	11.7	5.7	15.42	3.92	22018.30
3	339.2	77.3	63.2	75.4	71.1	14.1	6.2	24.01	6.08	28955.86
4	1006.0	84.0	63.2	80.0	76.2	20.8	7.8	48.43	12.13	52939.71
5	2061.3	90.0	63.3	84.1	81.0	26.6	8.9	77.41	19.20	81447.29
6	4089.0	101.1	63.4	91.4	91.0	37.8	10.1	108.24	26.40	158346.27
7	6086.1	112.5	63.6	102.4	102.1	48.9	10.4	124.52	29.91	274965.73
8	9200.3	129.4	64.2	120.0	116.3	65.2	13.1	141.10	33.24	542993.02
9	10997.9	135.6	68.0	125.4	122.6	67.6	13.0	162.71	38.10	639885.08

TABLE 5.4

Working fluid Water

Quantity of the working fluid ($\dot{V}^+$) = 0.0513

Heated length-cooled length ratio (R) = 2.0

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	89.6	71.7	63.1	70.9	65.7	8.6	6.1	10.38	2.66	14528.78
2	180.3	74.4	63.1	73.0	68.0	11.3	6.5	15.96	4.06	20985.58
3	339.2	77.4	63.2	77.2	70.2	14.2	7.2	23.92	6.06	29169.64
4	1006.0	82.7	63.3	79.0	75.4	19.5	7.3	51.71	12.97	47661.24
5	2061.3	90.4	63.4	85.5	81.7	27.0	8.7	76.27	18.90	83747.96
6	4089.0	101.1	63.5	96.2	90.6	37.6	10.6	108.75	26.52	157610.89
7	6086.1	113.9	63.8	105.2	101.9	50.1	12.1	121.38	29.11	292154.00
8	9200.2	128.8	64.9	119.7	116.1	63.9	12.7	143.98	33.94	525474.82
9	10997.9	135.4	68.5	126.4	122.5	66.8	12.8	164.61	38.55	630051.77

TABLE 5.5

Working fluid Water

Quantity of the working fluid (V^+) = 0.077Heated length-cooled length ratio (R) = 2.0

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	87.8	69.8	63.1	69.5	65.4	6.8	4.5	12.95	3.32	10638.27
2	385.4	77.1	63.1	76.1	70.0	14.0	7.1	27.61	7.00	28418.89
3	1022.3	84.0	63.3	80.0	75.1	20.7	8.9	49.32	12.35	52881.27
4	2682.8	94.2	63.3	88.3	84.0	30.9	10.1	86.95	21.42	106703.29
5	4769.4	106.8	63.6	101.7	95.2	43.2	11.6	110.38	26.71	210323.06
6	7563.1	121.9	64.0	113.8	109.9	57.9	12.0	130.5	31.00	408390.00
7	10599.2	134.6	67.3	123.4	120.4	67.3	14.2	157.4	36.90	624987.56

TABLE 5.6

Working fluid Water
Quantity of the working fluid (V^+) = 0.0962
Heated length-cooled length ratio (R) = 2.0

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Cr_d
1	111.3	71.3	63.1	70.5	65.9	8.2	5.4	13.61	3.49	13540.52
2	408.3	77.5	63.2	76.5	70.3	14.3	7.2	28.56	7.23	29516.43
3	981.8	84.1	63.3	80.3	75.4	20.9	8.8	47.03	11.77	53505.56
4	2061.3	90.4	63.4	85.2	81.6	27.0	8.8	76.27	18.90	83747.96
5	4089.0	102.0	63.6	94.5	90.4	38.4	11.6	106.44	25.93	164686.93
6	7563.1	122.1	64.4	113.5	110.3	57.7	11.8	131.1	31.14	408304.61
7	10731.3	135.0	68.0	123.1	120.6	67.1	14.4	160.03	37.49	628052.37

TABLE 5.7

Working fluid Water

Quantity of the working fluid (V^t) = 0.180

Heated length-cooled length ratio (R) = 2.0

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_D	Gr_D
1	90.8	70.1	63.1	69.5	64.5	7.0	5.6	12.92	3.32	11137.69
2	171.7	73.9	63.1	72.3	66.8	10.3	7.1	15.94	4.06	19616.39
3	654.3	81.0	63.2	78.0	72.8	17.8	8.2	36.70	9.24	41308.34
4	2061.3	86.0	63.3	82.8	76.2	22.7	9.8	90.82	22.67	61560.71
5	4089.0	100.6	63.6	95.9	90.0	37.1	10.7	110.37	26.94	153249.00
6	7563.1	121.9	64.4	113.5	109.7	57.4	12.2	131.7	31.29	404272.01
7	10997.9	134.9	67.8	123.7	122.6	67.1	12.4	163.95	38.42	627008.94

TABLE 5.8

Working fluid Water

Quantity of the working fluid (V^+) = 0.285

Heated length-cooled length ratio (R) = 2.0

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	90.8	70.2	63.1	69.5	64.8	7.1	5.4	12.83	3.29	11238.56
2	180.3	73.8	63.2	73.0	67.1	10.7	6.8	16.87	4.30	19426.38
3	1022.3	84.1	63.2	80.0	74.7	20.9	9.4	48.95	12.25	53446.66
4	2682.8	94.8	63.3	89.3	85.1	31.5	9.8	85.15	20.96	110957.63
5	4769.4	106.8	63.6	102.1	95.2	43.2	11.7	110.36	26.71	210379.99
6	7563.1	121.8	63.9	113.5	110.0	57.9	11.8	130.66	31.04	407030.00
7	9200.3	129.5	64.8	119.7	115.9	64.8	13.5	142.10	33.45	539851.87
8	10997.0	135.7	68.1	125.8	122.4	67.6	13.3	162.64	38.08	642025.45

TABLE 5.9

Working fluid Water

Quantity of the working fluid (V^+) = 0.0103

Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	107.5	74.0	63.1	70.2	68.4	10.9	5.6	9.86	2.52	2389.02
2	216.4	75.2	63.2	73.3	68.8	12.1	6.4	17.95	4.56	27607.08
3	407.0	77.9	63.2	74.7	70.9	14.7	7.1	27.70	7.00	36984.21
4	785.1	80.6	63.3	77.9	72.6	17.2	8.0	45.60	11.47	47278.19
5	1207.2	84.8	63.6	79.7	76.0	21.3	8.8	56.74	14.19	66860.85
6	2473.5	90.4	64.8	87.2	80.8	25.6	9.5	96.71	23.97	95136.27
7	4906.8	97.4	67.5	91.7	87.3	29.9	10.0	164.11	40.24	135643.49
8	7303.3	115.2	70.6	100.3	100.6	44.6	14.6	163.90	39.24	321219.11

TABLE 5.10

Working fluid Water

Quantity of the working fluid (V^+) = 0.0154

Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu _d	Gr _d
1	107.5	69.8	63.1	70.1	66.3	6.7	3.5	16.02	4.11	12619.73
2	216.4	72.8	63.2	73.0	67.7	9.6	5.1	22.50	5.74	20233.38
3	441.6	76.7	63.3	76.5	70.6	13.4	6.0	33.03	8.37	32204.75
4	785.1	80.4	63.4	79.7	73.3	17.0	7.1	46.05	11.60	46586.60
5	1226.7	83.6	63.7	81.7	76.7	20.0	6.9	61.50	15.40	60385.73
6	2404.4	90.2	64.5	87.9	81.3	25.7	8.9	93.61	23.20	95055.19
7	4906.8	98.4	67.7	92.1	88.9	30.7	9.5	160.03	39.18	143136.29
8	7184.1	112.7	70.7	100.7	96.9	42.0	15.8	170.99	41.07	284778.93
9	11040.4	124.6	77.7	108.9	107.3	46.9	17.3	235.30	55.72	421721.39

TABLE 5.11

Working fluid Water

Quantity of the working fluid (V^+) = 0.0231

Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	107.5	68.8	63.1	68.4	64.6	5.7	4.2	18.96	4.88	10242.19
2	216.4	72.1	63.2	71.0	66.9	8.9	5.2	24.21	6.19	18310.31
3	441.6	75.6	63.3	75.8	69.2	12.3	6.4	36.10	9.16	28388.18
4	785.1	79.3	63.4	79.0	72.8	16.0	6.5	49.41	12.47	41800.46
5	1226.7	83.2	63.7	81.7	76.2	19.5	6.9	63.07	15.81	58013.31
6	2404.4	88.8	64.8	85.5	81.0	24.0	7.8	100.20	24.89	85091.03
7	4906.8	95.9	68.2	89.3	78.5	27.7	8.4	177.40	43.60	120356.81
8	7303.3	110.7	70.7	99.3	97.9	40.0	12.8	182.92	44.04	257438.33
9	11040.4	123.9	77.7	106.6	108.2	46.2	15.7	239.23	56.70	407947.67

TABLE 5.12

Working fluid Water

Quantity of the working fluid (V^+) = 0.0321Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	102.5	68.5	63.1	67.1	63.7	5.4	4.8	18.99	4.89	9639.73
2	412.7	72.7	63.2	69.5	66.0	9.6	6.7	43.11	11.00	20067.47
3	1038.3	80.3	63.2	80.3	72.2	17.1	8.1	60.71	15.26	46471.55
4	2543.7	86.2	63.4	84.1	77.3	22.9	8.9	111.29	27.76	74953.26
5	5617.8	102.2	65.0	93.1	91.7	37.3	10.6	150.80	36.72	193071.99
6	9075.7	111.8	68.4	102.8	99.5	43.5	12.3	208.84	50.21	288488.99
7	11040.4	116.7	74.5	106.9	102.7	42.2	14.1	261.57	62.51	315835.19
8	12640.0	132.9	84.2	114.8	113.2	48.7	19.8	259.55	60.93	524598.53

TABLE 5.13

Working fluid Water

Quantity of the working fluid (V^+) = 0.0513Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	106.8	68.9	63.1	67.4	64.3	5.8	4.6	18.57	4.77	10418.25
2	465.5	73.0	63.1	69.5	66.6	9.9	6.3	47.22	12.05	20841.63
3	1226.7	77.6	63.2	74.7	70.9	14.4	6.6	85.10	21.53	35831.73
4	3219.4	89.1	63.2	85.2	79.6	25.9	9.5	124.51	30.92	92483.15
5	5409.8	94.9	67.2	90.3	86.2	27.7	8.8	195.07	48.00	117581.75
6	8942.7	112.8	77.9	98.6	102.2	34.9	10.5	256.50	61.59	236808.71
7	12249.4	122.4	85.1	109.0	112.2	37.4	10.2	327.77	77.81	319894.37

TABLE 5.14

Working fluid Water

Quantity of the working fluid (V^+) = 0.077

Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	105.3	68.6	63.1	67.1	62.9	5.5	5.8	19.12	4.92	9885.99
2	462.5	74.4	63.2	70.9	67.5	11.3	6.9	41.00	10.44	25126.20
3	1216.9	81.0	63.2	77.9	73.3	17.8	7.7	68.4	17.20	49582.94
4	3299.4	89.5	63.3	85.5	80.1	26.2	9.4	126.19	31.32	94639.47
5	5723.3	99.0	67.5	92.8	89.2	31.5	9.9	181.51	44.40	149769.77
6	9075.7	114.7	78.8	99.3	103.2	35.9	11.5	252.62	60.52	256204.64
7	13197.4	123.5	86.1	111.0	111.8	37.4	11.7	352.74	83.64	328272.36

TABLE 5.15

Working fluid Water

Quantity of the working fluid (V^+) = 0.0962

Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_D	Gr_D
1	133.6	70.4	63.1	69.8	66.0	7.3	4.3	18.36	4.71	13960.63
2	490.0	74.0	63.2	70.9	68.3	10.8	5.6	45.33	11.55	23769.63
3	1178.1	78.2	63.2	74.7	70.5	14.9	7.7	78.83	19.93	37872.33
4	3219.4	89.4	63.4	85.9	80.6	26.1	8.9	123.45	30.64	94238.78
5	5723.3	99.3	69.2	94.1	89.6	30.1	9.7	190.45	46.57	143637.63
6	9075.7	110.8	79.3	102.1	99.5	31.5	11.3	287.98	69.33	203878.13
7	12719.0	126.2	87.7	116.2	113.7	38.4	12.4	330.90	78.23	357789.92

TABLE 5.16

Working fluid Water

Quantity of the working fluid (V^+) = 0.180Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	125.6	69.7	63.1	69.1	65.0	6.5	4.7	19.22	4.94	12207.50
2	412.7	73.4	63.2	70.5	67.3	10.2	6.1	40.35	10.29	21948.31
3	1178.1	79.8	63.2	75.1	71.4	16.6	8.4	71.13	17.94	44315.08
4	3219.4	89.5	63.4	85.5	79.9	26.1	9.6	123.20	30.57	94607.72
5	5617.8	99.1	68.4	91.0	87.7	30.7	11.4	183.23	44.82	145855.58
6	8942.7	115.9	79.7	103.8	101.5	36.2	14.4	247.31	59.16	265153.61
7	12719.0	126.6	88.3	112.8	113.4	38.3	13.2	331.91	78.43	360196.19

TABLE 5.17

Working fluid Water

Quantity of the working fluid (V^+) = 0.285Heated length-cooled length ratio (R) = 1.25

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	125.6	69.4	63.1	68.8	64.5	6.3	4.9	20.00	5.14	11624.57
2	441.6	74.4	63.2	71.0	69.1	11.2	5.3	39.40	10.03	24917.69
3	1178.1	77.8	63.2	75.4	71.0	14.6	6.8	80.90	20.46	36485.81
4	3219.4	89.8	63.4	85.9	81.0	26.3	8.8	122.24	30.32	96145.70
5	5723.3	97.0	68.7	91.7	87.4	28.3	9.6	202.50	49.68	126821.57
6	9075.7	113.0	80.6	105.5	101.3	32.4	11.7	280.20	67.27	221205.13
7	12877.5	126.5	89.7	111.7	113.3	36.9	13.3	349.20	82.53	346152.73

TABLE 5.18

Working fluid Water

Quantity of the working fluid (V^*) = 0.0513

Heated length-cooled length ratio (R) = 0.8

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu _d	Gr _d
1	134.4	67.9	63.1	67.4	63.6	4.8	4.3	28.15	7.25	10405.66
2	270.5	69.7	63.2	68.8	65.0	6.6	4.7	41.07	10.54	15429.97
3	508.7	72.1	63.2	70.5	65.9	8.9	6.2	57.00	14.57	22856.44
4	981.4	74.7	63.2	73.0	67.3	11.4	7.3	85.77	21.82	32110.94
5	1509.0	77.7	63.3	74.4	69.6	14.4	8.1	104.84	26.53	44900.97
6	3091.9	84.9	63.6	80.0	76.6	21.4	8.4	144.79	36.20	84109.15
7	6133.5	94.5	69.4	88.3	83.9	25.1	10.6	244.36	60.18	131307.10
8	9129.2	105.9	76.6	98.6	94.1	29.3	11.7	311.43	75.46	208905.28
9	13800.5	118.7	81.8	108.6	105.3	36.8	13.4	374.58	89.30	361062.43
10	16496.8	127.6	85.4	117.2	112.8	42.3	14.8	390.47	92.17	507740.49
11	21448.4	144.9	94.1	131.2	125.9	50.8	19.0	422.03	97.98	860994.17

TABLE 5.19

Working fluid Water

Quantity of the working fluid (V^+) = 0.180

Heated length-cooled length ratio (R)=0.8

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu _d	Gr _d
1	1187.5	139.5	133.5	138.0	135.9	6.0	3.6	197.95	46.18	92037.42
2	1690.6	140.2	133.4	138.3	135.9	6.8	4.2	248.32	57.90	105801.45
3	2591.4	141.2	133.1	138.3	135.9	8.1	5.2	320.64	74.69	127988.76
4	4225.4	142.5	132.0	139.0	135.9	10.5	6.6	402.42	93.63	170424.54
5	6381.3	144.0	130.6	139.3	135.9	13.4	8.1	477.79	111.02	222726.82
6	10049.2	145.9	128.9	139.7	135.9	17.0	10.0	589.90	136.84	293795.81
7	15311.8	147.8	123.6	139.7	135.9	24.2	11.9	631.67	146.30	431851.23

TABLE 5.20

Working fluid Water

Quantity of the working fluid (V^+) = 0.180Heated length-cooled length ratio (R) = 0.8

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	2591.4	156.3	150.4	154.3	152.4	5.9	3.8	441.78	101.66	118773.93
2	4024.2	157.5	150.3	154.3	152.4	7.2	5.1	560.46	128.85	147865.82
3	6133.5	158.8	150.1	154.3	152.4	8.8	6.4	698.74	160.50	183927.60
4	8832.3	160.7	149.7	154.3	152.4	10.9	8.2	807.25	185.19	234518.35
5	13617.1	162.2	146.7	154.7	152.4	15.4	9.7	882.00	202.13	336795.30
6	18569.9	165.1	144.7	155.0	152.4	20.4	12.6	912.12	208.65	458382.41
7	22375.8	167.0	143.1	155.0	152.4	23.9	14.6	937.09	214.11	547848.75

TABLE 5.21

Working fluid Freon-11

Quantity of the working fluid (V^+) = 0.068

Heated length-cooled length ratio (R) = 0.8

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu _d	Gr _d
1	294.2	80.3	76.2	77.5	77	4.11	3.3	71.6	126.68	1516542.83
2	500.0	82.4	76.3	77.9	78.0	6.12	4.36	81.74	144.68	2199969.43
3	1369.3	88.1	77.1	83.1	80.6	11.1	7.5	123.91	219.32	3717325.24
4	2034.9	93.6	79.2	85.5	83.9	14.5	9.7	140.68	249.00	4579121.23
5	2492.7	96.7	79.7	87.6	85.8	17.0	10.9	146.52	259.35	5212572.76
6	3542.7	107.2	83.2	93.5	94.3	24.0	12.9	147.72	261.46	6633077.76
7	5506.4	131.7	95.1	115.5	114.9	36.6	16.8	150.5	266.36	8234550.65

TABLE 5.22

Working fluid Freon-11

Quantity of the working fluid (V^*) = 0.109

Heated length-cooled length ratio (R)=0.8

Run No.	q	T_h	T_c	T_v	T_s	$T_h - T_c$	$T_h - T_s$	h	Nu_d	Gr_d
1	245.3	79.8	76.2	77.2	76.7	3.6	2.8	68.28	120.87	1334283.87
2	508.7	82.4	76.5	78.3	78.0	5.9	4.4	86.12	152.43	2124752.20
3	981.4	85.2	76.6	81.4	79.0	8.6	6.2	113.97	201.72	2996292.75
4	1521.1	89.2	77.6	83.5	81.0	11.6	8.2	131.02	231.90	3856406.42
5	3091.9	102.1	80.9	92.8	90.1	21.2	11.9	146.03	258.50	6147953.43
6	3925.5	112.1	85.2	101.4	98.1	26.9	14.0	145.95	258.33	7112795.67
7	5535.5	131.7	95.2	115.2	114.4	36.5	17.3	151.85	268.78	8203510.62

Fig. 3.1 Saturation temperature-pressure relationship

Fig. 3.2 Saturation temperature-pressure relationship

Fig. 3.3 Idealized two phase closed thermosyphon configuration.

Fig. 3.4 Theoretical maximum heat transfer rate for different cooled wall temps. and for $R = 2$

Fig. 3.5 Theoretical maximum heat transfer rate for different cooled wall temps. and for $R = 1.25$

Fig. 3.6 Theoretical maximum heat transfer rate for different cooled wall temps. and for $R = 0.8$

Fig. 3.7 Theoretical maximum heat transfer rate for different cooled wall temps. and for $R=0.8$

Fig. 4.1 Schematic set up of the Experimental Apparatus

Fig. 4.2 Details of the test section

Fig. 4.3 Pictorial view of the experimental apparatus

Fig. 5.1 Effect of V^+ on wall temperature distribution

Fig. 5.2 Effect of R on the wall temperature distribution

FIG. 5.3 Effect of p on wall temperature distribution

Fig. 5.4 Effect of q on wall temperature distribution

Fig. 5.5 Effect of working fluid on wall temperature distribution

Fig. 5.6 Variation of maximum heat transfer coefficient with θ for different values of V^+ and $R=2$

Fig. 5.7 Variation of maximum heat transfer coefficient with Θ for different values of V^+ and $R=1.25$

Fig. 5.8 Variation of maximum heat transfer coefficient with Θ
 for $v^+ = 0.0513$ and $R = 0.8$

Fig. 5.9 Variation of maximum heat transfer coefficient with Θ for different values of v^+ and $R=0.8$

Fig. 5.10 Variation of maximum heat transfer coefficient with T_s for different values of v^+ and $R=2$

Fig. 5.11 Variation of maximum heat transfer coefficient with T_s for different values of v^+ and $R=1.25$

FIG. 5.12 Variation of maximum heat transfer coefficient with T_s for $V^+ = 0.0513$ and $R = 0.8$

FIG. 5.13 Maximum heat transfer coefficient as a function of V

Fig. 5.14 Variation of maximum heat flux with T_s for different V^+ and $R=2$.

Fig. 5.15 Variation of maximum heat flux with T_s for different V^+ and $R = 1.25$.

Fig. 5.16 Effect of V^+ on maximum heat flux for $R=2$

Fig. 5.17 Effect of V^+ on maximum heat flux for $R = 1.25$

Fig. 5.18 Effect of R on maximum heat transfer coefficient

Fig. 3.19 Effect of R on maximum heat transfer coeff. for different values of T_s

Fig. 5.20 Effect of operating pressure on heat transfer coefficient

Fig. 5.21 Maximum heat transfer coeff. as a function of maximum heat flux for $R=2$

Fig. 5.22 Maximum heat transfer coeff. as a function of max. heat flux for $R=1.25$

Fig. 5.23 Maximum heat transfer coeff. as a function of max. heat flux for $R=0.8$

Fig. 5.24 Heat transfer coeff. as a function of heat flux at high pressures

Fig. 5.25 Variation of max. heat transfer coeff. with max. heat flux

Fig. 5.26 Comparison of maximum heat transfer rate with theory for $R=2$

Fig. 5.27 Comparison of max. heat transfer rate with theory for $R=1.25$

Fig. 5.28 Comparison of maximum heat transfer rate with theory for $R=0.8$ for water

Fig. 5.29 Comparison of maximum heat transfer rate with theory for $R=0.8$ for Freon-11

Fig. 5.30 Comparison with single phase closed and open thermosyphons.

Fig. 5.31. Effect of the working fluid

Fig. A.3.1 Details of the refrigeration system