Int. J. Heat Mass Transfer. Vol. 15, pp. 1695–1707. Pergamon Press 1972. Printed in Great Britain

A TWO-PHASE CLOSED THERMOSYPHON

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(*Received* 3 *December* 1971)

Abstract—An experimental study on the heat transfer performance of a two-phase closed thermosyphon together with a simple theoretical analysis for its maximum heat transfer capacity has been made. Water and Freon-11 were used as the working fluids. Out of many possible controlling variables, the effects of (a) the amount of working fluid in the tube, (b) the ratio of heated-length to cooled length, (c) the operating pressure, (d) the heat flux and (e) the working fluid, were investigated.

NOMENCLATURE

B, C, D,	defined quantities;
c,	specific heat at constant pressure of the working fluid at the liquid phase;
h,	heat transfer coefficient;
G,	mass flow rate of condensate;
g,	acceleration due to gravity;
k,	thermal conductivity of the working fluid in the liquid phase;
L,	length;
L^+,	dimensionless length, L_h/L_c;
ΔP,	force difference;
q,	heat flux in the evaporator section of the thermosyphon tube;
R,	inside radius of thermosyphon tube, $D/2$;
r,	radius;
T,	temperature;
u,	velocity of the condensate in z-direction;
V,	volume;
V^+,	dimensionless volume of the working fluid in the thermosyphon, V_l/V_t;
y,	dimensionless thickness, r/R;
z,	downward vertical axis of the thermosyphon tube from the top of the condensing end;
Nu,	Nusselt number, (hD/k);
Pr,	Prandtl number, $(c\mu/k)$;
Gr,	Grashof number, $(\rho^2 g\theta\beta D^3/\mu^2)$;
Ra,	$GrPr$.

Greek letter symbols

β,	coefficient of thermal expansion;
δ,	liquid film thickness at $z = L_c$;
θ,	temperature difference between the average temperature in the heated section and the average temperature in the condensing section, $(T_h - T_c)$;
μ,	absolute viscosity of the working fluid in the liquid phase;
ρ,	density of the working fluid in the liquid phase.

Subscripts

b,	boiling;
c,	cooled length or condenser;
fg,	latent heat of evaporation;
g,	gravity;
h,	heated length or evaporator;
i,	condition defined in Fig. 14;
l,	liquid;
m,	maximum, momentum or mean;
s,	saturation;
t,	total;
v,	vapor or vapor core.

INTRODUCTION

WHILE the concept of the two-phase thermosyphon has been known for a long time, its use has been neglected—especially for low temperature operation—although large quantities of heat can be transferred easily from one place to another with small temperature differences. Serious interests in the thermosyphon first arose from the application of an open thermosyphon to the cooling of gas turbine rotor blades by Schmidt [1] during World War II.

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

Because the condensate is returned in the direction of the force, the system has the advantage of large rates of heat transfer particularly in one direction and therefore acts as a thermal rectifier. The system has also a special advantage because it does not require compressors nor pumps.

There have been very few investigations on the two-phase closed thermosyphon other than those reported in the open literature [2–6], and accordingly the design information and operating technique of the system are neither understood nor available. On the other hand, a thorough review on "heat pipes" which were developed mainly for use in zero gravity is given, for example by Feldman [7].

Cohen and Bayley [2] who briefly reviewed the possible methods of cooling gas-turbine rotor blades and suggested the thermosyphon system as the most attractive carried out a number of experiments using both static and rotating apparatuses. They reported that over a wide range, the heat transferred from the heated to the cooled ends of the thermosyphon is independent of the quantity of coolant enclosed. They also determined a criterion for the maximum rate of heat flow from the static-rig tests.

Since a closed two-phase thermosyphon can be used as a thermal rectifier, Long [3] used the system to keep permafrost frozen. The boiling section of the thermosyphon is placed in the ground adjacent to or is made part of the building foundation. The condensing section is above the ground surface level and heat is transferred to the atmosphere. In this way, the heat flow out of the ground throughout the whole year could be greatly increased without a corresponding increase in heat flow into the ground. Long [3], however, did not make any systematic study of the various parameters governing the fluid flow and heat transfer characteristics of the system.

The general problem of condensation in a variable acceleration field was investigated analytically by Chato [4] and the special case of the linear variation which occurs in a constant cross-section, rotating thermosyphon was studied and he concluded that the effects of the temperature differential and the Prandtl number are similar to those in other condensation problems.

The possible application of heat transfer by evaporative cooling of a liquid metal under high centrifugal acceleration to turbine blade cooling was discussed by Genot and Le Grives [5]. They stated that liquid alkali metal in the blade cavity could be used for the cooling of turbine blades but definite conclusions on the favorable aspects and practical difficulties of its use cannot be drawn until they complete the experimental study they were performing.

Very recently, Larkin [6] reported on an experimental study on a closed thermosyphon very similar to that used in the present study but his temperature range was much higher. He concluded that his test program did not lead to a precise relationship for the prediction of boiling and condensing heat transfer coefficients in a two-phase thermosyphon, but the results showed that when water was used as a working fluid, the coefficients were typical of boiling and condensing.

The present study originated from the work one of the authors [8] was doing in connection with the preservation of permafrost as a structural foundation in cold regions such as the Canadian Arctic. Permafrost is a suitable foundation for building but its thawing will cause movement and damage to the structure. As discussed already Long has tried the thermosyphon for this purpose, but, the performance characteristics which are essential to apply the system in this way are not yet known.

The object of the present paper is, therefore, to make an experimental study into the performance of a stationary two-phase closed thermosyphon along with a theoretical analysis of the system for an ideal case. In particular, the effects of the following parameters on the performance of the thermosyphon as a heat transfer system were investigated:

a. amount of working fluid,
b. heated length-cooled length ratio,
c. mean operating pressure (or corresponding saturation temperature),
d. heat flux, and
e. working fluid.

EXPERIMENTAL APPARATUS

A schematic diagram of the experimental set up is shown in Fig. 1. The main heat transfer loop consists of a closed thermosyphon test tube, a main electrical resistance heater assembly, a guard heater, a number of separate cooling jackets, a cooling system and associated instrumentation.

The thermosyphon test tube used in the present study was made of a 1 in. standard 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 threaded cap into each end.

A 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. The lower cap also carried a copper tube to height of about one inch inside the thermosyphon test tube serving as a thermocouple well. All the connections were made using either Swagelock brass fittings or 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 outside wall of the test tube. The thermocouples were uniformly spaced on its circumference along the length. Provision was also made to measure the temperature in the vapor core by means of a traversing iron–constantan thermocouple probe, and another thermocouple in the T/C well of the lower cap.

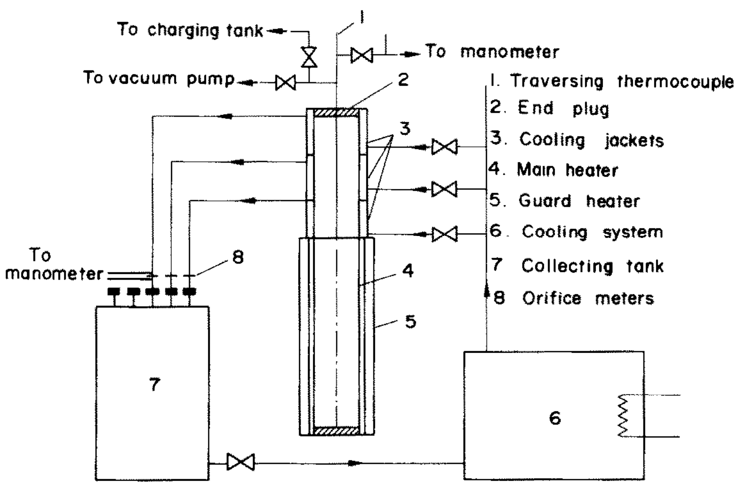

FIG. 1. Schematic diagram of the experimental apparatus.

The main requirement in the heat input section, here called the evaporator, was to have a heat input which could be measured accurately. The system chosen for this purpose was two electrical heaters, (a) the main heater and (b) the guard heater.

The main heater was used to supply the total 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 strung on it. The small wire and bead sizes were chosen so that the beaded heating wire could easily be wound around the thermosyphon tube of 1·316 in. o.d.

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

An Acromag model 370 relay was used to maintain a constant temperature in the insulation between the two heaters, the relay being activated by the two calibrated iron–constantan thermocouples embedded in the insulation between the two heaters. The main heater was supplied with electrical power from a variable transformer rated at 13·4 KVA, 240 V, and power to the guard heater was controlled by the relay which operated on 115 V, 60 cycles.

The heat from the test section is removed at the upper part of the thermosyphon test tube by a number of separate cooling jackets which have an 'O' ring on either end so that they can easily slide over the test tube and 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.

All the thermocouples were calibrated up to 600°F. The cold junction of the thermocouple was kept at the melting point of ice. The outputs of these thermocouples, during the early stages of the investigation, were recorded on a Hewlett and Packard model 7100 B strip chart recorder and also read, upon reaching the steady state, by a Leeds and Northrup Model 8687 potentiometer. In the later experiments, the output of the thermocouples was read from a self calibrated Hewlett and Packard DVM model 240 IC, with a resolution of one microvolt. Power to the test section was measured by an ammeter with a 1000/5 current transformer and a voltmeter.

The flow of the coolant in each jacket could be individually measured by means of precalibrated orifice meters provided for each jacket. To remove air and other non condensible gases, a Leybold mechanical vacuum pump with a rating of 10^{-1} torr was used. Provision was made to reach an ultimate vacuum of 5×10^{-5} torr by using a Leybold diffusion pump in conjunction with the mechanical pump. The pressure inside the thermosyphon tube was measured by a vacuum pressure gauge.

EXPERIMENTAL 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 vacuum and leaks checked and corrected. After the system has been evacuated, a known amount of the working fluid was charged into the tube. The amount to be charged was kept in a charging tank which was connected to a charging valve. The charging valve was then opened and the working fluid flowed into the tube due to the pressure difference. Care was taken to see that no condensible gases or air leaked into the thermosyphon tube. The power was then supplied to both the main and guard heaters. Power to the main heater was increased in steps up to the desired heat flux. At each power level, the power to the guard heater was controlled by the relay and when a steady state was reached, (within 30–50 min) 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 at a given operating condition, would be reached above which the wall temperature would begin to rise suddenly and rapidly, in which case the power input was cut off.

DISCUSSION OF RESULTS

A purely theoretical study of the closed two-phase thermosyphon is not possible at present due to lack of information and the complexity of the condensation and boiling processes in a closed cavity. However, guided by the functional relationship among the variables obtained by dimensional analysis, the effects of various parameters on the performance of a closed two-phase thermosyphon were evaluated experimentally and they were compared with the results of an analysis based on a simple model for an ideal case.

In the present study, the tube heat transfer coefficient, h, was defined as

$$h = q/(T_h - T_c). \tag{1}$$

This coefficient, h, is used throughout the present report in preference to the boiling coefficient, h_b or the condensing coefficient, h_c. However, the tube coefficient, h, can be related to h_b and h_c of the tube through the following expression,

$$\frac{1}{h} = \frac{1}{h_b} + \frac{L^+}{h_c} \tag{2}$$

where L^+ is the ratio of heated-length to cooled-length of the thermosyphon.

It can be shown by dimensional analysis that this tube heat transfer coefficient depends upon the quantity of the working fluid in the tube, V^+, the ratio of heated-length to cooled-length, L^+, the ratio of heated-length to tube-diameter, L_h/D, the liquid saturation temperature (corresponding to the mean operating pressure) T_s, the heat flux q, and the working fluid. Here we will examine the effect of each of these parameters, except that of L_h/D, on the heat transfer coefficient, particularly at very low operating pressure.

To clearly explain the terms, we may define here the maximum heat tranfer rate, q_m, and the corresponding maximum heat transfer coefficient, h_m.

For a given operating pressure and a given wall temperature in the evaporating section in a thermosyphon tube, there should be a limiting value of heat flux above which the operating pressure inside the thermosyphon tube cannot be maintained. This limiting heat flux is termed the maximum heat transfer flux, q_m and the heat transfer coefficient of the tube at the limiting heat flux is the maximum heat transfer coefficient, h_m.

The maximum heat transfer for an ideal case was studied analytically (see Appendix) and the expression obtained is given as

$$q = \frac{k(T_s - T_c)}{RL^+} \cdot \frac{\left(\frac{1}{8} - \frac{y_i'^2}{2} + \frac{3}{8}y_i'^4 - \frac{y_i'^4}{2}\ln y_i'\right)}{\left[y_i'^4 \ln y_i'\left(\frac{\ln y_i'^2}{2} - \frac{1}{2}\right) + \frac{y_i'^4}{8} + y_i'^2\left(\frac{\ln y_i'}{2} - \frac{1}{4}\right) + \frac{1}{8}\right]}. \tag{3}$$

The maximum heat flux as a function of saturation temperature as given by equation (3), for a two-phase closed thermosyphon circular tube of i.d. 1·062 in. is shown in the Fig. 2 for L^+ values of 2·0, 1·25 and 0·8 respectively, using water and Freon-11 as the working fluids. These results will be compared with those obtained from experiment in the following section.

Some typical equilibrium temperature distributions of the test tube as well as vapor temperatures are shown in Fig. 3.

(a) *Effect of the quantity of the working fluid* V^+

The maximum heat transfer coefficients as a function of saturation temperature (correspond-

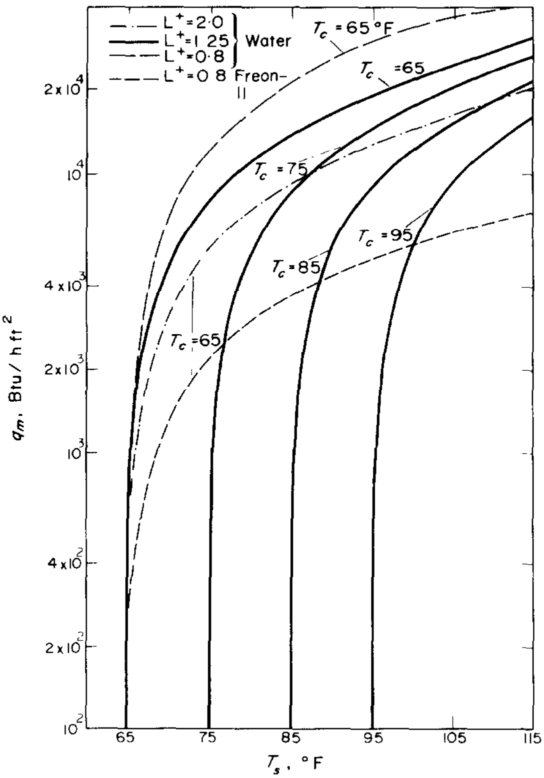

FIG. 2. Theoretical maximum heat flux.

ing to the operating pressure) inside the thermosyphon tube, are shown in Figs. 4 and 5 for the ratio L^+ of 1·25 and 2, respectively, and for different quantities of water, V^+. From these figures, the variation of the maximum heat transfer coefficient with the quantity of working fluid is obtained and shown in Fig. 6. Similarly, the behavior of maximum heat transfer flux with increase in the quantity of water, is shown in Figs 7 and 8 for the ratio of L^+ of 1·25 and 2, respectively. This trend was also repeated in all of the cases where Freon-11 was used as the working fluid.

As may be noted from Figs. 7–9, within the range of V^+ studied, both the maximum heat transfer coefficient and the maximum heat flux increase with an increase in the amount of the working fluid up to a certain quantity of the working fluid and then become independent of V^+. The quantity of the working fluid at which the maximum heat flux becomes independent of the amount of V^+ is termed the minimum quantity of the working fluid. Cohen and Bayley [2] also reported similar trends of V^+ on the heat transfer rate but Larkin's results [6] indicate otherwise.

This phenomena may be explained by the fact that so long as the quantity of the working fluid

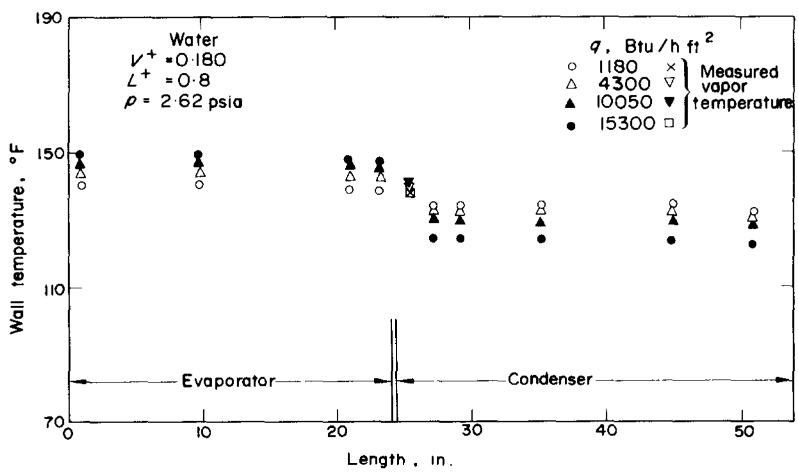

FIG. 3. Typical stabilized wall temperature distribution for various heat fluxes.

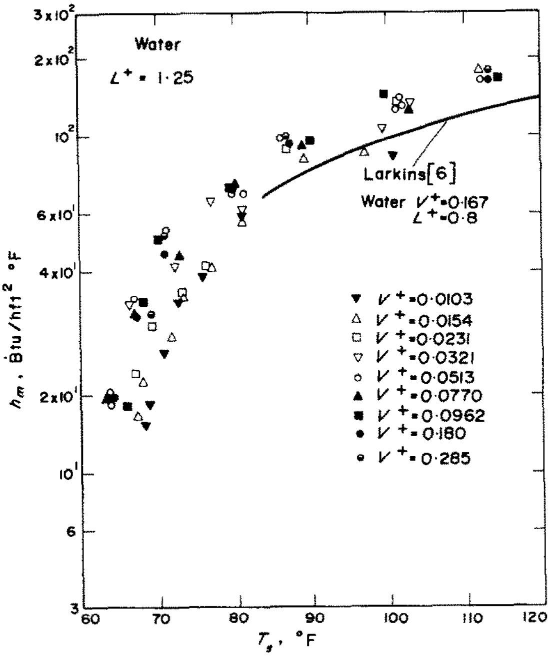

FIG. 4. Variation of maximum heat transfer coefficient with T_s.

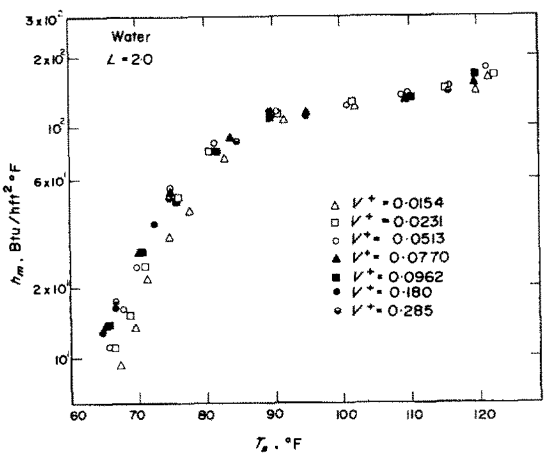

FIG. 5. Variation of maximum heat transfer coefficient with T_s.

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 being 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 quantity of the working fluid. Then the temperature of the heated section begins to rise rapidly and undesirably. An immediate reduction of the heat input is then necessary to prevent burn-out. This occurred for an example at a heat input of 2·25 kW for the ratio of L^+ of 1·25 when the quantity of water, V^+ was 0·013. However, the results from the experiments where burn-out occurred were not presented in the paper as they were not reproducible.

(b) *Effect of the ratio of heated-length to cooled-length,* L$^+$

The maximum heat transfer coefficient increases with decrease in the heated length–cooled length ratio, L^+, as shown in Fig. 9. Bailey and Lock [9] reported similar effects in their study on a single phase stationary closed thermosyphon tube. This trend has been predicted in Fig. 2 which was obtained from the analysis given by equation (3). A decrease in L^+ means that the area of condensing region increased and that of the evaporator decreased. This can be related by the following equation which is from an energy balance for the whole tube,

$$q_m = q_c/L^+ \tag{4}$$

where q_c is the condensation heat flux at a given operating condition. The increased condenser area seems to provide a more efficient system as a whole. However, there must be a limiting L^+ below which the trend will be opposite.

(c) *Effect of mean operating pressure,* p *(or the corresponding temperature,* T$_s$)

The heat transfer coefficient increases

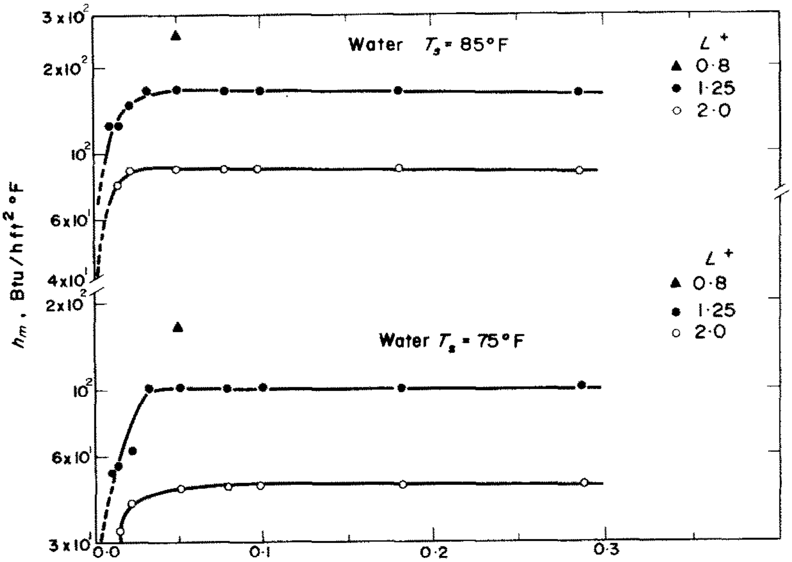

FIG. 6. Maximum heat transfer coefficient vs. V.

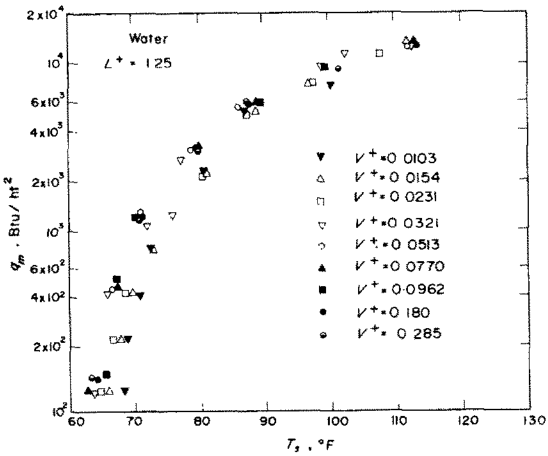

FIG. 7. Variation of maximum heat flux with T_s.

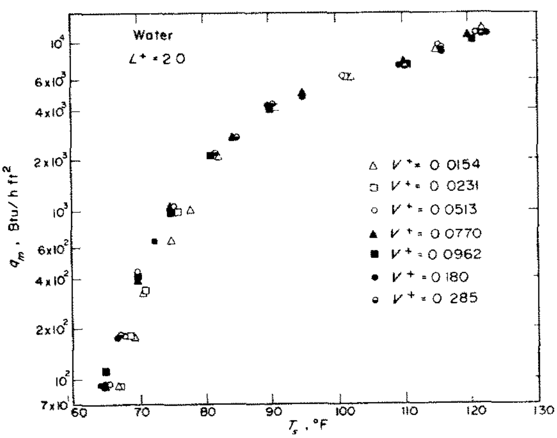

FIG. 8. Variation of maximum heat flux with T_s.

appreciably with an increase in the mean operating pressure inside the thermosyphon tube of a given L^+, V^+, and fluid as shown in Figs. 7–9, 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 slopes of the saturation pressure–temperature relationship for water and Freon-11 increase rapidly with increase in pressure and therefore, give small temperature

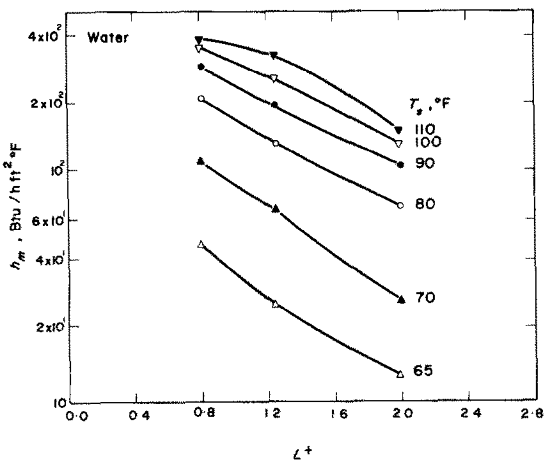

FIG. 9. Effect of L^+ on maximum heat transfer coefficient.

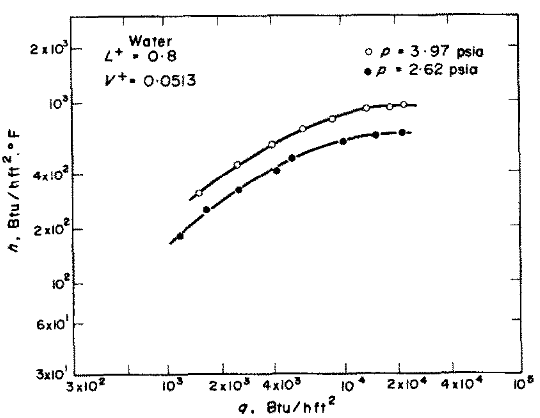

FIG. 10. Effect of heat flux on heat transfer coefficient.

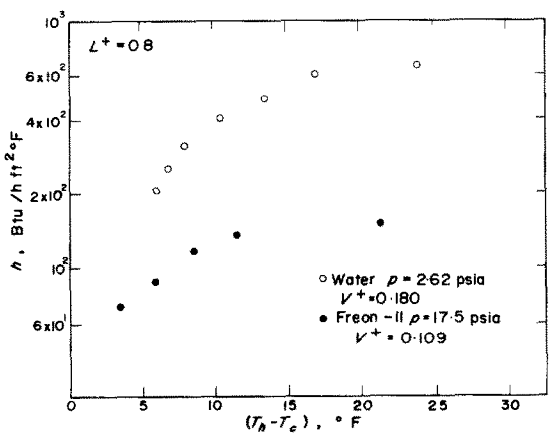

FIG. 11. Effect of working fluid on heat transfer coefficient.

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 faster from the condensing section, which increases the heat transfer coefficient.

In the present investigation, a heat flux of 22400 Btu/hft^2 (70·6 kW/m^2) was transferred for the thermosyphon tube of 1·062 in. i.d. at a pressure of 3·97 psia without reaching the burn-out situation. Cohen and Bayley [2] observed a critical heat flux of 40000 Btu/hft^2 at atmospheric pressure for a rotating thermosyphon tube of $\frac{1}{4}$in. i.d., and of 73500 Btu/hft^2 at a pressure of 190 psia.

(d) *Effect of heat flux*

At low operating pressure the effect of heat flux on the heat transfer coefficient, h, seems to be significant as seen from Fig. 10 which illustrates the effect with typical results. This trend was also observed by Larkin [6] who reported that the lower heat flux results showed some dependence on heat flux especially at the lower operating temperature ($T_s < 200$°F). From these results, it is noted that the heat transfer coefficient has a tendency to become less dependent on heat flux at higher operating pressure (or high operating temperature).

(e) *Effect of working fluid*

In addition to water, Freon-11 was also used as a working fluid 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 in Fig. 11. The operating conditions were not identical but a qualitative comparison can be easily made from this figure.

As expected, the maximum heat transfer coefficient with Freon-11 was small 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 cannot be used. Investigation of Freon-11 as the working fluid in the low temperature range (below freezing point of water at 1 atm) is being continued.

(f) *Comparison of the experimental results with the simple analysis*

Figure 12 shows the typical variation of both theoretical and experimental maximum heat transfer rates with the saturation temperature corresponding to the mean operating pressure

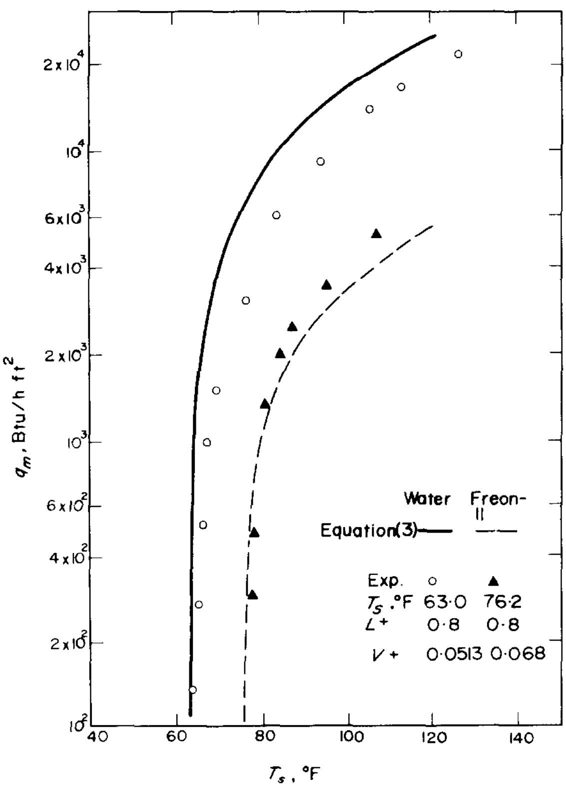

FIG. 12. Comparison of the experimental results with the analytical prediction.

in the thermosyphon tube. Although equation (3) predicts the trend, it overestimates the actual results for water by a factor of 1·6–2·0 depending upon the value of L^+. However, it underestimates the maximum heat flux for a given mean operating pressure when Freon-11 is used as the working fluid, as shown in the figure. This discrepancy may be due to the idealized assumptions made in the analysis discussed in the Appendix.

It is very doubtful that the inclusion of such effects as sub-cooling of the condensate, nonlinearities in the film temperature distribution, acceleration, convection and shear stress at the liquid–vapor interface in the analysis would improve the results. These effects were neglected in the present analysis. However, the analyses of Koh, Sparrow and Hartnett [10] and Chen [11], which are representatives of recent works on the condensation heat transfer, demonstrate that the results of the analyses including the terms stated above did not substantially differ from the classic Nusselt's results for values of $c\Delta T/h_{fg}$ between 0 and 0·2. The present analysis predicted the general characteristics of a two-phase thermosyphon such as the effects of L^+, T_s and the kind of working fluid used as has been shown in Fig. 2.

Since no other experimental results than those of Larkin's [6] are available with which the present findings could be compared, it may be of interest to compare the present results with those of a single-phase stationary closed thermosyphon tube reported by Bayley and Lock [9] and this comparison will give 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 [12]. These comparisons are shown in Fig. 13.

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

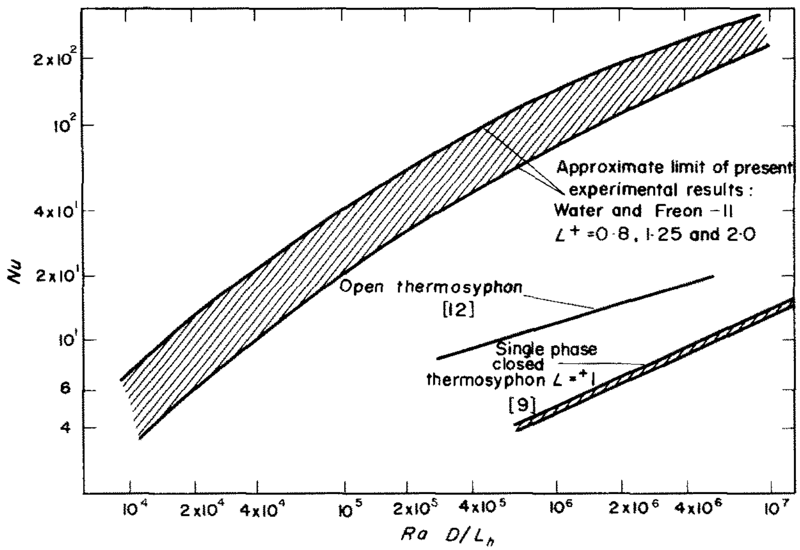

FIG. 13. Comparison of present experimental results with single phase closed and open thermosyphons.

phase closed thermosyphon is due to the fact that it utilizes heat transfer characteristics of phase change which are more efficient means of heat transfer than that of single-phase.

CONCLUSION

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, have 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.
2. Decreasing the ratio L^+ led to a consistently increasing heat transfer coefficient, within the range of L^+ studied.
3. The heat transfer coefficient is very sensitive to the operating pressure and rapidly increases with increasing pressures.
4. A theoretical analysis for the maximum heat transfer rate predicts the trends closely and the agreement with the experimental results for water and Freon-11 is good.

The results obtained in the present investigation are encouraging, as the simple sealed tube from of thermosyphon has a high heat transfer capacity suitable for many engineering applications. However, there is still a lack of information on other aspects of the two-phase closed thermosyphon than those so far studied, and further work is being continued to reach definitive conclusions.

ACKNOWLEDGEMENTS

The partial financial support of the Defence Research Board of Canada under Grant No. 951196 is gratefully acknowledged and the authors would like to thank V. J. Lunardini for his useful discussion.

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APPENDIX

Maximum Heat Transfer Rate

The basic flow equation upon which depends the heat transfer of a two-phase closed thermosyphon is obtained from a force balance as:

$$\Delta P_g \geqslant \Delta P_r + \Delta P_l + \Delta P_m. \qquad \text{(A.1)}$$

The following assumptions are made for the analysis.

1. The condensate returns to the evaporator as a continuous film flowing down the tube along its circumference.
2. The liquid flow in the boundary layer is laminar and the sensible heat associated with sub-cooling or superheating is negligible compared to the latent heat.
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.

Since the forces due to vapor pressure drop and that due to pressure drop caused by momentum changes are considered negligible, the forces acting on an infinitesimal

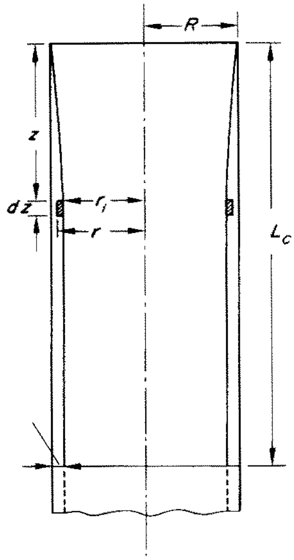

FIG. 14. Idealized two phase thermosyphon.

element of the liquid film in the condensing section of the idealized two-phase thermosyphon, shown in Fig. 14, are

$$\pi(r^2 - r_i^2)\,\rho \frac{g}{g_c}\,\mathrm{d}z = -\,2\pi r \mu \left(\frac{\mathrm{d}u}{\mathrm{d}r}\right)\mathrm{d}z. \qquad \text{(A.2)}$$

From equation (A.2), we now obtain

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

and

$$u_m = \frac{gB}{\pi R^2(1 - y_i^2)\, g_c}\left[\tfrac{1}{4} - y_i^2 + \tfrac{3}{4}y_i^4 - y_i^4 \ln y_i\right]$$

where

$$B = \frac{\rho \pi R^4}{2\mu} \quad \text{and} \quad y_i = \frac{r_i}{R}.$$

Then the mass flow rate of the condensate is given by

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

Equation (A.3) 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 [13].

From an energy balance, we also obtain

$$h_{fg}\,\mathrm{d}G = \frac{2\pi k(T_s - T_c)}{\ln(1/y_i)}\,\mathrm{d}z. \tag{A.4}$$

Substituting the value of dG from equation (A.3) into equation (A.4) and integrating, we have

$$\frac{2\pi k\mu(T_s - T_c)}{\rho^2 R^4 h_{fg}}\frac{g_c}{g}z = y_i^4 \ln y_i\left(\frac{\ln y_i}{2} - \frac{1}{2}\right) + \frac{y_i^4}{8} + y_i^2\left(\frac{\ln y_i}{2} - \frac{1}{4}\right) + \frac{1}{8}. \tag{A.5}$$

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

$$q2\pi RL_h = \int_0^{L_c} \frac{2\pi k(T_s - T_c)}{\ln(1/y_i)}\,\mathrm{d}z. \tag{A.6}$$

Differentiating equation (A.5), rearranging, and substituting into equation (A.6), we now have

$$q = \frac{k(T_s - T_c)}{RL^+}\frac{C}{D} \tag{A.7}$$

where

$$C = \frac{1}{8} - \frac{y_i'^2}{2} + \tfrac{3}{8}y_i'^4 - \frac{y_i'^4}{2}\ln y_i'$$

$$D = y_i'^4 \ln y_i'\left(\frac{\ln y_i'^2}{2} - \frac{1}{2}\right) + \frac{y_i'^4}{8} + y_i'^2\left(\frac{\ln y_i'}{2} - \frac{1}{4}\right) + \frac{1}{8}$$

and

$$y_i' = 1 - \frac{\delta}{R}.$$

Also from equations (A.5) and (A.7),

$$q = \frac{\rho^2 R^3 h_{fg} g}{2\mu L_h g_c}C \tag{A.8}$$

or

$$Nu = \frac{\rho^2 R^4 h_{fg} g}{\pi k\mu L_h(T_h - T_c)\,g_c}C = \frac{h}{h_c}\frac{C}{D}. \tag{A.9}$$

To calculate q for a given T_s, equations (A.7) and (A.8) are used. First, from equation (A.8) 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 equation (A.7). The properties k, μ, ρ and h_{fg} are next calculated at the film temperature $(T_s + T_c)/2$. With these modified values of k, μ, ρ and h_{fg} the heat flux q and saturation temperature T_s are again calculated using equation (A.7) and (A.8). The procedure is repeated until the same value of T_s, within a desired accuracy, is obtained.

UN THERMOSYPHON FERME BIPHASIQUE

Résumé—On fait une étude expérimentale sur les performances thermiques d'un thermosyphon fermé biphasique en même temps qu'une étude théorique simple sur sa capacité de transfert thermique maximale. On a pris comme fluides caloporteurs de l'eau et du fréon 11. Parmi plusieurs variables actives possibles on a considéré les effets de (a) la quantité de fluide en mouvement dans le tube, (b) le rapport de la longueur chauffée à la longueur refroidie, (c) la pression, (d) le flux thermique et (e) le fluide en circulation.

EIN GESCHLOSSENER THERMOSYPHON FÜR ZWEI PHASEN

Zusammenfassung—Das Wärmeübergangsverhalten eines geschlossenen Thermosyphons für zwei Phasen wurde experimentell untersucht und durch eine einfache theoretische Betrachtung über den maximalen Wärmetransport ergänzt. Als Arbeitsmedien wurden Wasser und Frigen 11 verwendet.

Von der Vielzahl der Einflussgrössen wurden untersucht (a) die Füllmenge im Rohr, (b) das Verhältnis; beheizte Länge zu gekühlter Länge, (c) Arbeitsdruck, (d) Wärmestromdichte, (e) Arbeitsmedium.

ДВУХФАЗНЫЙ ЗАМКНУТЫЙ ТЕРМОСИФОН

Аннотация—Проведено экспериментальное исследование характеристики теплообмена двухфазного замкнутого термосифона и простой теоретический анализ его максимальной мощности теплообмена. Рабочими жидкостями служили вода и фреон — 11. Из множества возможных определяющих переменных изучалось влияние а/количества рабочей жидкости в трубе, б/отношения длины нагреваемого участка к длине охлаждаемого, в/рабочего давления, г/теплового потока и д/рабочей жидкости.