

Theoretical Steady-state Forced Flow Boiling Evaporator for Double-tube Thermosyphon (FFDTT)

Loula A. Shouman, M. Abd El-Aziz, Adel Alyan

Atomic Energy Authority, Reactors Department, 13759 Cairo, Egypt
bidosalem2@yahoo.com

Abstract: A mathematical model has been developed for the new type of the thermosyphon, in order to describe thermal behavior of the forced flow boiling double-tube thermosyphon in steady state operation. The present work of theoretical forced flow boiling inside the evaporator of the double-tube thermosyphon is described in a mathematical model in one dimensional flow. The model is describing both thermal and phase flow of the forced flow double tube thermosyphon (FFDTT). A computer program is designed to solve the presented model to estimate the performance of the thermosyphon. The current model can predict the behavior of a forced flow boiling double-tube closed two-phase thermosyphon, at different operating conditions. Analysis of the thermosyphon response to various operating conditions has been performed.

Index Terms— Forced Flow Boiling; Double-tube closed-thermosyphon; steady-state; Theoretical investigation; Model; Analytical Solution.

1. INTRODUCTION

The conventional thermosyphon is typically made by drawing a vacuum on an air-tight vessel, then a controlled amount of working fluid, charged into the vessel [1]. When heat added at the bottom of thermosyphon, the liquid vaporizes and rises to the top of the vessel due to buoyancy effect and liquid runs down the walls to the bottom of the thermosyphon, due to gravity [2]. While the double tube thermosyphon (DTT) is a conventional closed two-phase thermosyphon with an inserted inner tube in the evaporator section. In the double tube thermosyphon, the inner tube divides the liquid pool in the evaporator into two regions: an inner cylinder (down comer) and an annular channel. When the heat added to the outer tube of the evaporator, liquid in the annular gap is heated to saturation and ultimately boils. Vapor-liquid flow is formed in the annular gap flows upwards along the gap to the top of the inner-tube. Since there is two-phase in the annular gap and one phase liquid is in the inner tube, down comer, forced circulation by the pump causes the liquid in inner tube to flow from the bottom of the tube to the annular gap. This process produces an inner forced annular circulation of the working fluid inside the evaporator section. Since the latent heat of evaporation is high, the wickless heat pipe can transport considerable quantities of heat over considerable distance with a very small temperature drop [3]. The flow boiling in double tube thermosyphon is due to the natural circulation of fluid. The natural circulation is started when the heat added in the riser and a density difference is occurred between the riser and down comer. At this investigation a new type of forced flow boiling inside the evaporator of thermosyphon (FFDTT) has been developed by using a pump to circulate the fluid. The theoretical investigations were performed to study the response of a water thermosyphon to abrupt changes in input power,

different forced flow rates at evaporator length.

2- The model

A mathematical model is in one dimensional flow simulate the steady-state performance and the operational characteristics of the gravity-assisted forced flow two-phase closed thermosyphon with double-tube evaporator in the vertical orientation FFDTT. The model is considerably simplified to reduce the processing computer time as much as possible, which consider the lumped analytical and numerical models, which require long computational time. Therefore, the model is primarily based on the relation of heat transfer by convection and conduction in each section of double tube thermosyphon. Then, the temperature profile for the wall and working fluid is accurately determined by the proposed model. The current model can predict the behavior of the forced flow double tube- closed two-phase thermosyphon, at different operating conditions.

2-1 Model description and assumptions

Fig. (1) illustrates the geometry and dimensions of the thermosyphon main tube made of copper with 32 mm inner diameter, 1.5mm thickness and 1000 mm long. The evaporator and condenser section lengths are 600 mm and 250 mm respectively, while the adiabatic section is 150 mm long. Inner tube installed inside the evaporator. The inner tube has 600 mm lengths. This structure forms a hot channel, where the forced flowing liquid circulated between inner and outer tubes [4,5,6,7,8]. The evaporator section was uniformly heated and the heat source is simulated directed to the outer wall of the evaporator section. While the condenser section was convectively cooled, using cooling water at ambient temperature and constant pressure at 0.05 MPa, the forced flow boiling two-phase closed thermosyphon with double-

tube evaporator (FFDTT) is basically divided axially into three basic regions: evaporator (heating), adiabatic (thermally shielded) and condenser (cooling) sections. Individually, each region is mathematically and thermally treated due to variation of the heat transfer processes. In addition, the evaporator region is sub-divided into two main regions: a liquid pool and a liquid film region. The heat transfer regimes are completely different for the two regions and consequently, the pertinent heat transfer coefficients. The model is analyzed in one-dimension, where the axial coordinate x is mainly measured from the evaporator bottom. To facilitate the current analysis, the present numerical steady-state model has considered the following assumptions:

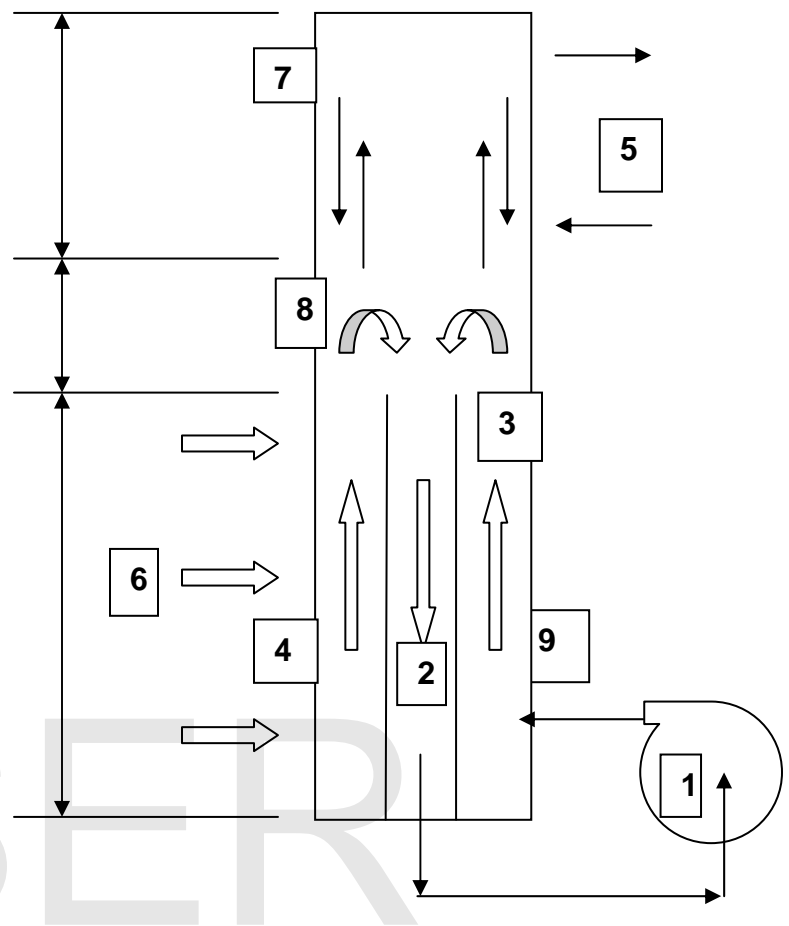
1. One-dimensional flow for both vapor and liquid in the axial direction.
2. Isoflux heating, i.e uniform radial heat flux of heating and cooling processes in the evaporator and condenser sections.
3. The vapor superheat is very small; therefore the vapor is taken at the saturated conditions.
4. The condensate calculation is considered at the saturation temperature, thus sub-cooling is neglected.
5. The pressure drop in the liquid film and vapor core is considered negligible.
6. Axial conduction through the adiabatic wall is negligible.
7. Conduction and convection heat transfer at the liquid-vapor interface are considered very small.
8. Saturation vapor temperature in the adiabatic section equals that of liquid-wall interface at the exit of the evaporator.

2.2 Mathematical Modeling

A one-dimensional, steady state flow model is presented for the water/copper forced flow double-tube thermosyphon. This model is proposed to predict the local liquid and thermosyphon wall temperature distribution. For the present model, the most controlling parameters considered are the sub cooling flow rate at the bottom end of the evaporator, and the heat transfer rate (thermal load). Once, the values of these parameters are predetermined, and then the local external wall, liquid and vapor temperature can be readily calculated. The heat transfer through the working fluid and thermosyphon wall is treated as a semi-conductor and is characterized by a thermal resistance, defined in the following relation.

Therefore, the local temperature of any semi-conductor is primarily dependent on the local thermal resistance. But, the thermal resistance is inversely proportional to the thermal conductivity of wall material, or the convective heat transfer coefficient through the working fluid. Alternatively, the heat transfer coefficient is commonly related to the thermohydraulic characteristics of flow. Thus, each section of the thermosyphon is separately treated, as the heat transfer regime differ from one section to the other.

The general model description is as follows:



- | | |
|------------------------------|--------------------------------|
| 1. Pump. | 8. Adiabatic Section, 15 cm. |
| 2. Inner Tube. | 9. Evaporator Section , 60 cm. |
| 3. Annular Gap. | |
| 4. Thermosyphon Tube. | |
| 5. Cooling Water. | |
| 6. Input Power. | |
| 7. Condenser Section, 25 cm. | |

Figure.1. A schematic of the model Forced Flow Boiling Double-Tube Thermosyphon.

2.2.1 Heat transfer analysis in heat sink

Exit temperature of cooling water $t_{cw,o}$ can be given as function of cooling flow rate and inlet temperature $t_{cw,i}$:

$$t_{cw,o} = \frac{Q}{m_{cw} * c_{p,cw}} + t_{cw,i} \tag{1}$$

Where, the temperature difference across the heat calorimeter is:

$$\Delta t_{cw} = t_{cw,o} - t_{cw,i} \tag{2}$$

$$T_{cw} = \frac{q_c}{h_{cw}} + t_{wc,o} \tag{3}$$

The average heat transfer coefficient for forced convection, between the condenser surface and cooling water can be calculated in the case of laminar and turbulent flow regimes [9] respectively as:

$$h_{cw} = 1.86(Re Pr)_{cw}^{0.33} (d_h/L_c)^{0.33} (\mu_{cw}/\mu)^{0.14} (k_{cw}/d_h) \text{ for laminar} \tag{4}$$

$$h_{cw} = 0.023(Re_{cw})^{0.8} (Pr_{cw})^{0.33} (k_{cw}/d_h) \text{ for turbulent} \tag{5}$$

The hydraulic diameter d_h for the annulus area of cooling jacket The flow is assumed as turbulent at $Re_{cw} \geq 3000$, or laminar at $Re_{cw} < 3000$, also all physical properties are evaluated at a mean temperature equal to the average between the mean wall temperature and the bulk cooling temperature,

$$Re_{cw} = 4m_{cw} / \pi * d_h * \mu_{cw} \tag{6}$$

As a good approximation, all properties are evaluated at the exit temperature of cooling water $t_{cw,o}$.

2.2.2 Heat transfer in condenser section (wall, falling film & vapor)

As the convective heat transfer coefficient h_{cw} was determined, the outer condenser wall temperature $t_{wc,o}$ is evaluated as:

Furthermore, the local inner condenser wall temperature $t_{wc,i}$ is determined by mean of the heat transfer by conduction through the wall. Then, the vapor temperature in the condenser can be calculated as:

The falling condensate film flow inside thermosyphon surface is assumed as laminar flow and the film was thin with respect to the radius of curvature. Therefore, the simplified Nusselt's film condensation theory for vertical flat plate was adopted for condensation in the thermosyphon condenser [10]. The local heat transfer of condensation is given as:

$$T_{wc,i} = T_v - \frac{Q}{A_{c,o} h_{fc}} \tag{7}$$

$$T_{wc,o} = T_{wc,i} - \frac{Q \text{Log}(d_o/d_i)}{2\pi k_w L_c} \tag{8}$$

$$h_{cf} = \left(\frac{4}{3}\right)^{\frac{4}{3}} [Re_{cf}]^{\frac{-1}{3}} \frac{k_l}{(v_l^2 / g)^{1/3}} \tag{9}$$

heat transfer of condensation is given as:

Where Re_{cf} is the local Reynolds number of the falling condensate film, and $q_{c,i}$ is the heat flux based on the inner condenser surface area.

$$h_{cf} = \left(\frac{4}{3}\right)^{\frac{4}{3}} [Re_{cf}]^{\frac{-1}{3}} \frac{k_l}{(v_l^2 / g)^{1/3}} \tag{10}$$

As heat energy is transferred through the condensate layer by conduction, thus, the temperature gradient could be assumed linear through the film. Therefore, the average condensate film temperature can be calculated as:

$$t_{lc} = (t_{wc,i} + t_v) / 2 \tag{11}$$

2.2.3 Adiabatic section calculations

According to the assumption, the vapor temperature drop is neglected in the adiabatic section. Therefore, the vapor temperature in adiabatic is considered equal to the vapor temperature at the exit of the evaporator and the inlet to condenser sections. Owing to the thermal equilibrium between the vapor, falling liquid film and wall, the adiabatic wall and vapor temperatures is:

$$t_{wa} = t_v = t_l (L_e + L_a) \tag{12}$$

2.2.4 Calculation of heat transfer coefficient

The heat energy is transferred to the fluid inside the annular riser through the outer tube wall. Heat transfer mechanism in riser based mainly on the forced flow pattern of fluid. Heat energy is transferred by convection in sub-section having single liquid phase, while heat is transferred by nucleate boiling coefficient for subsections having two-phase mixture flow. In condenser section, the heat transfer coefficient is by condensation. Also, the heat transfer coefficient is considered by convection for the cooling water in condenser jacket.

2.2.4.1 Single-phase sub cooled heat transfer coefficient

In sub cooled section the heat transfer coefficient between the sub cooled forced flowing liquid and tube wall is calculated from Dittus-Boelter correlation [11] for forced convection as following:

$$h_{sb} = \frac{Nu_{sb} * k_{sb}}{D_H} \tag{13}$$

Where the Nussle number (Nu) is calculated by:

$$Nu_{sb} = 0.023 * Re_{sb}^{0.8} * Pr_{sb}^{0.4} \tag{14}$$

$$Re_{sb} = \frac{\rho_{sb} * u_{sb} * D_H}{\mu_{sb}} \tag{15}$$

$$Pr_{sb} = \frac{\mu_{sb} * c_{p, sb}}{k_{sb}} \tag{16}$$

$$T_{sb,i} = T_{dc,0} = T_v \tag{17}$$

$$T_{sb,0} = \frac{Q_{sb}}{m \cdot Cp} + T_{sb,i} \quad (18)$$

$$T_{wsb,i} = \frac{q}{h_{sb}} + T_{sb} \quad (19)$$

$$T_{wsb,o} = \frac{Q_{sb} \cdot \text{Log}(d_o/d_i)}{2\pi k_w L_{sb}} + T_{wsb,i} \quad (20)$$

2.2.4.2 Two-phase flow heat transfer coefficient

For the forced flow boiling in the double tube thermosyphon (FFDTT), the flow boiling, is driven by the forced circulation inside the evaporator section, caused by the pump between the annulus and down-comer. In sub cooled and saturated boiling, various empirical correlations for the heat transfer coefficient in two-phase flow have been developed. In this model, the correlation proposed by Rohsenow and Chen [12] is considered a heat transfer coefficient of two-phase. This correlation propose that the flow boiling heat transfer actually results from the combination of pool boiling heat transfer and flow convective flow heat transfer as following:

$$h_{TP} = h_{NB} + h_{FC} \quad (21)$$

Where:

$$h_{NB} = s \cdot h_{FZ} \quad (22)$$

Where, *s* is the suppression factor and *h_{FZ}* is the nucleate boiling heat transfer coefficient calculated from the Froster-Zuber equation [12].

$$h_{FZ} = \frac{0.00122 \cdot \Delta T_{sat}^{0.24} \cdot \Delta P_{sat}^{0.75} \cdot c_{pl}^{0.45} \cdot \rho_l^{0.49} \cdot k_l^{0.79}}{\sigma^{0.5} \cdot H_{fg}^{0.24} \cdot \mu_l^{0.29} \cdot \rho_g^{0.24}} \quad (23)$$

The convective part of the total heat transferred transfer is as:

$$h_{FC} = F \cdot h_l \quad (24)$$

Where:

- F: the two-phase heat transfer coefficient multiplier.
- h_l: the single phase liquid convective heat transfer coefficient calculated from Dittus-Boelter equation and is given as following:

$$h_l = \frac{Nu_l \cdot k_l}{D_H} \quad (25)$$

The flow condition can be determined by the aid of the Reynolds number (Re) of the fluid, which is related to the mass flow rate of liquid:

$$Nu_l = 0.023 \cdot Re_l^{0.8} \cdot Pr_l^{0.4} \quad (26)$$

$$Re_l = \frac{\rho_l \cdot (1-x) \cdot u_l \cdot D_H}{\mu_l} \quad (27)$$

$$Pr_l = \frac{\mu_l \cdot c_{pl}}{k_l} \quad (28)$$

$$x = \frac{Q_b}{m \cdot H_{fg}} \quad (29)$$

$$Bo = q / (G \cdot H_{fg}) \quad (30)$$

$$X_{tt} = \left[\frac{1-x}{x} \right]^{0.9} \cdot \left(\frac{\rho_v}{\rho_l} \right)^{0.5} \cdot \left(\frac{\mu_l}{\mu_v} \right)^{0.1} \quad (31)$$

$$F = 1 + (2400 \cdot (Bo^{1.16})) + (1.37 \cdot \left(\frac{1}{X_{tt}} \right)^{0.86}) \quad (32)$$

$$S = \left[\frac{1}{1 + (1.15 \cdot 10^{-6}) \cdot (Re_b)^{1.17} \cdot (F)^2} \right] \quad (33)$$

$$T_{wb,i} = \frac{q}{h_{TP}} + T_b \quad (34)$$

$$t_{wb,o} = \frac{Q \text{Log}(d_o/d_i)}{2\pi k_w L_b} + t_{wb,i} \quad (35)$$

3.1 Solution procedures

The numerical, one-dimensional, steady state model is proposed to avoid the complex lumped modeling, as well as reducing the program computational time. The model primary depends on the basis of the previous empirical relationships and the theoretical results. Generally, the accurate prediction of axial temperatures of vapor, liquid and wall, in the thermosyphon sub-regions, is the main goal of this model.

The solution primarily starts at the heat sink point, and then precedes toward the condenser, adiabatic and evaporator subsections. The procedures of solution sequence are briefly stated as:

1. Input the initial conditions: working fluid properties, thermosyphon dimensions, heat sink and source characteristics and the cooling parameters.
2. Initialize the cooling Reynolds number, *Re_{cw}* and determine the flow regime (laminar or turbulent), then the average heat transfer coefficient, *h_{cw}* is calculated between the cooling medium and the condenser wall.
3. As a first step, the local outer and inner wall temperature *t_{wc,o}* & *t_{wc,i}* are estimated along the condenser section.
4. Assuming laminar falling condensate film and the local condensation heat transfer coefficient *h_{cf}* is

- calculated, and then the vapor temperature t_v distribution is found in the axial direction of the condenser section.
5. The vapor temperature in the adiabatic section is taken equal to that at the inlet to the condenser section t_{v,L_e+L_a} , and due to the thermal equilibrium, the falling film is also taken at the same temperature $t_{wa,o}$.
 6. A continuous, smooth and laminar thin liquid film in the evaporator is assumed, then the local evaporation heat transfer coefficient h_{ef} , and both inner and outer wall temperature, $t_{wef,i}$ & $t_{wef,o}$ are calculated.
 7. The local liquid pressure in the evaporator pool, p_{lp} , due to the action of the hydrostatic head, is evaluated; consequently the corresponding saturation temperature distribution along the pool height t_{lp} is determined.
 8. The local inner wall temperature $t_{wep,i}$ in the liquid pool subsection, is found by calculating the local non-boiling and boiling heat transfer coefficient formed the average evaporator heat transfer coefficient h_{ep} , consequently the outer wall temperature $t_{wep,o}$ is determined.

It is clear from the program sequence that the present numerical solution is a straight forward calculation method without any iteration, which makes the program consume less time and proceeds quickly.

4. Discussion of the Analytical Modeling Results

The results of the analytical modeling are basically discussed and analyzed to evaluate the thermosyphon performance at the steady state operation of a closed two-phase double tube thermosyphon with forced convection at evaporator. This result analyzes the axial outer surface wall temperature distribution for all the thermosyphon sections. Also the effects of changing the transported thermal energy (heat rate) are conducted. The overall heat transfer coefficient, average evaporator heat transfer coefficient, effective thermal conductivity, vapor mass flow rate and the mean vapor temperature are presented. Also some parameters such as non-boiling, boiling regions along evaporator section, boiling and non-boiling heat transfer coefficient along evaporator section and the mean vapor quality are discussed. The results were performed at normal operation of the thermosyphon, without occurrence of any limit in the previously specified operation ranges.

4.1 Axial Wall and Fluid Temperature Distribution

The results were conducted at steady state operation to measure the local outside wall surface and vapor temperature distribution. These measures are of great significance to interpret the results and analyze the thermosyphon performance. Also temperature distribution is necessary to

monitor the operation of thermosyphon, in order to avoid wall dry out or overheating.

Experiments have been performed at different ranges of governing operating parameters, such as: the heat load varied from 1000 to 1500(W) also forced sub cooling flow rate $m\text{-sub}$ changed from 0.025 to 0.1 (kg/s). Various representative plots of the axial wall and fluid temperature distribution along the thermosyphon length are readily illustrated in Figs. (2) and (3) for water as working fluid. the local wall temperature rises first along the convection region and then decreases along the boiling region. It is quite natural that the forced sub cooling flow rate through the evaporator causes an upward flow in the annulus hot channel and return downward flow in the inner tube. The temperature of the fluid in the riser is thus expected to increase as the fluid flows up along the evaporator. However, it is shown that the fluid temperature in the annulus increases initially in the axial distance starting from the bottom, goes through a maximum at a certain distances from the bottom, and then decreases continually until it reaches the lowest value at the top of evaporator. This behavior may be physically understood by considering that, as the temperature of the fluid in the annulus becomes equal to the saturation temperature at a certain distance from the bottom, nucleate boiling commences. The boiling of the working fluid becomes possible in the annulus, due to the higher heat transfer coefficient. The same results of axial temperature distribution were found at the hot channel which, is agree with the evaporator wall temperature distribution.

As the adiabatic section is thermally insulated, the surface wall temperature remains constant for the thermosyphon. Also the results indicated the nonuniformity of wall temperature over the condenser section with distance. This results from the variation of the convection boundary conditions of the cooling water. In the condenser jacket, the cooling water enters the condenser with low temperature. As the cooling water flows towards the outlet, the temperature increases axially. In addition, the thermal resistance of the condensate film decreases with distances, due to the reduction of film thickness, which leads to reduction of temperature difference across the condensate film. Consequently, the wall temperature also rises axially. The great sensitivity of temperature variation with distance is sharply attributed to the large radial heat flux value in condenser section, which reaches about 2.4 times that of the evaporator radial heat flux.

While keeping the cooling flow rate constant in the cooling jacket for a liquid filling ratio 100% in the evaporator, effects of the heat load level on the steady state operation of thermosyphon are determined by changing the heat load applying on the evaporator. Fig. (2) plots the wall and fluid temperatures along the thermosyphons sections at constant cooling water rate of 0.046 kg/s and liquid filling ratio 100% considering the change in heat load. As seen from the figure, the local outer wall and fluid temperature increases for all thermosyphon sections: evaporator pool, adiabatic and

condenser, whenever the radial heat flow increases. Fig. (3) indicates that the axial local wall and fluid temperature of the forced flow double tube thermosyphon decreased during increase in forced flow rate through the evaporator section at constant heat load and cooling flow rate.

4.2 Heat Flux and Forced Flow rate Effect on the Vapor Quality

The effect of heat load on the exit vapor quality, x (calculated from equation. 29) is summarized and plotted in Figs. 4-5. The transported heat ranged from 1000 to 1500W for water, also forced sub cooled flow rate ranged from 0.05 to 0.1 kg/s. The figures show the great dependence and nonlinear proportionality of the vapor quality on the change of transmitted thermal loads and forced sub cooled water flow rate. Also from the figures the vapor mass flow rate at evaporator end of the forced flow double tube thermosyphon indicates the non-boiling (liquid) and boiling length (two phase). As a result of the increase in heat flux, the change in liquid temperature along the tube length required a shorter length of hot channel to attain the saturation value. This means that the non-boiling length is inversely proportional to the heat flux; on the other hand the non-boiling length is directly proportional to the increase in forced flow rate thought the evaporator. In the Figures for the water as working fluid the non-boiling length ratio differs from 66 % to 53 % of the total evaporator length, corresponding to a heat load of 1000 to 1500 (W) at constant forced flow rate at 0.025 kg/s. While for constant heat load at 1000 (W) and the forced flow rate ranged from 0.025 to 0.1(kg/s) the non-boiling length ratio varies from 65% to 5% of the total evaporator length. It is substantially concluded that at relatively high forced flow, the evaporator has a sub cooled hot channel (riser), while at lower forced flow the riser becomes fully saturated.

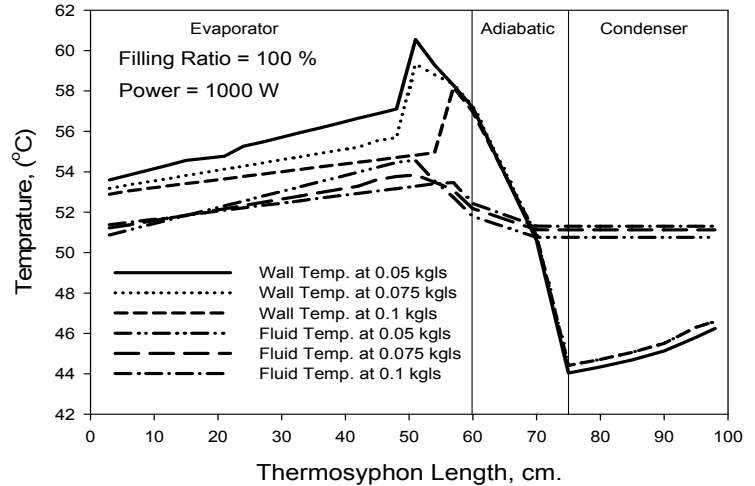


Figure. 3. Axial Outer Wall and Fluid Temperature Distribution for (FFDTT) at different Forced Sub cooling in Steady State Operation.

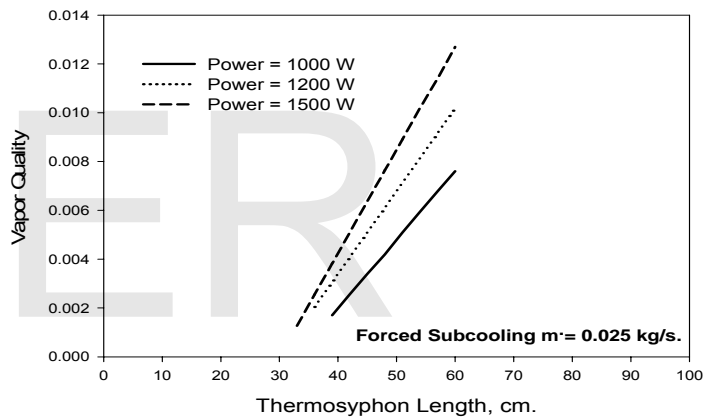


Figure. 4. Mean Vapor Quality of (FFDTT) for Different Loads at Steady State Operation at Constant Forced Flow rate.

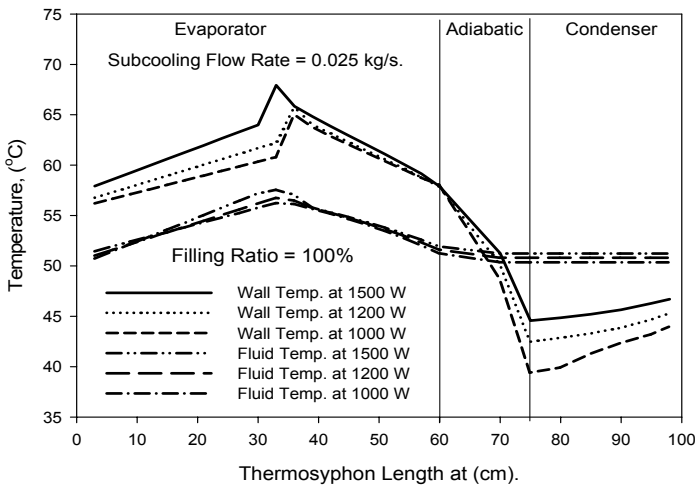


Figure.2. Axial Outer Wall and Fluid Temperature Distribution for (FFDTT) at different Heat load in Steady State Operation.

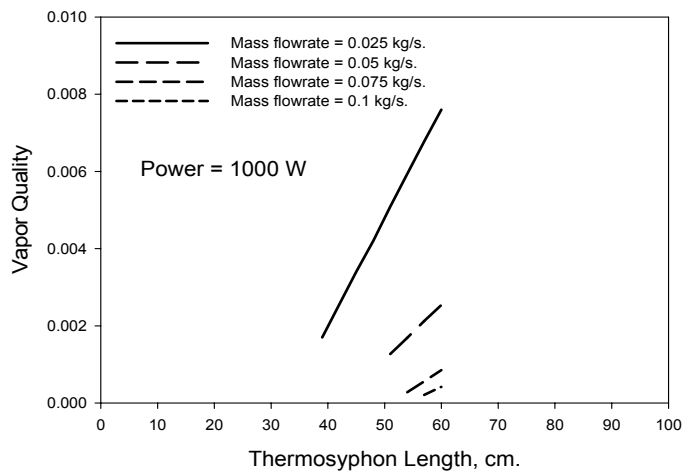


Figure. 5. Mean Vapor Quality of (FFDTT) for Different Forced Flow rate at Steady State Operation at Constant Load.

4.3. Evaporator heat transfer coefficient of forced-flow thermosyphon

Pool boiling mechanism is commonly adopted for the evaporation section of the ordinary thermosyphon, but for thermosyphon with forced-flow double tube evaporator, the forced flow boiling mechanism is considered inside the evaporation section. Single and two-phase heat transfer coefficients were observed. The superiority of FFDTT may return to the dynamics of the vaporization and flow of the working fluid in the annular gap, which cause a rapid flow in the annulus. The proper annular gap can effectively control the growth of the bubbles, and the natural circulation formed in the annular gap which enhances the separation of bubbles from the boiling surface so as to augment the heat transfer. In addition, due to the relatively smaller sectional area for the two-phase flow, there will be larger number of bubbles, and more volumetric concentration in each unit area, that creates more turbulence in the fluid. Fig. 6 and 7 indicates the relationship between the results of the single and two-phase heat transfer coefficients and radial heat flux in addition to the sub cooling flow rate. The boiling and non-boiling heat transfer coefficients values were calculated from equations (13) and (21) respectively in the range of heat load from 500 to 2000(W) and 0.025 to 0.1 (kg/s) for sub cooling flow rate. The average single-phase heat transfer coefficient is theoretically calculated by Stephan equation, the single-phase heat transfer coefficient directly increases with increases of heat load as well as increasing in sub cooling flow rate.

It is noted that the non-boiling heat transfer is higher than the boiling heat transfer coefficient for low heat flux refer to the forced flow sub cooled. Also as the heat flux increases, the initiation point of boiling shifts downward leading to remarkable increase of the two-phase subsection length, this augments the heat transfer coefficient.

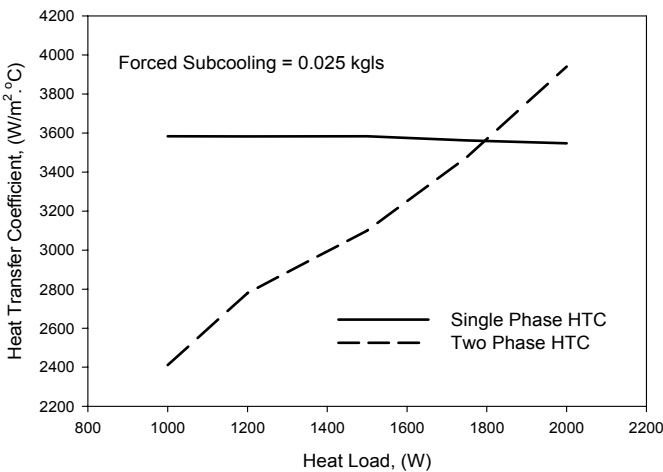


Figure 6. Single and Two-phase Heat Transfer Coefficient of (FFDTT) for Different Loads at Steady State Operation at Constant Forced Flow rate.

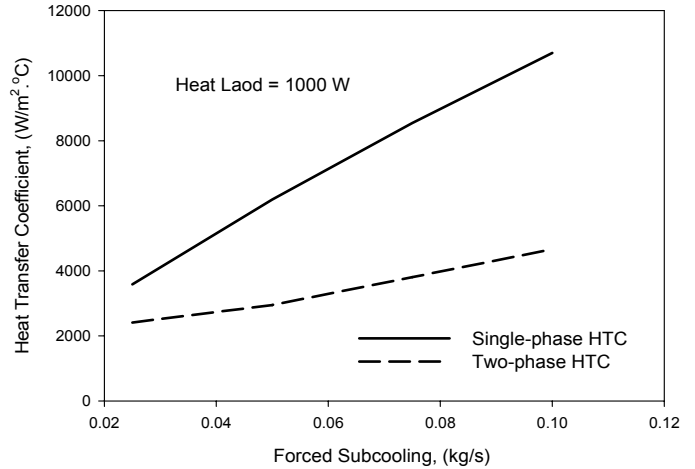


Figure 7. Single and Two-phase Heat Transfer Coefficient of (FFDTT) for Different Forced Flow rate at Steady State Operation at Constant Load.

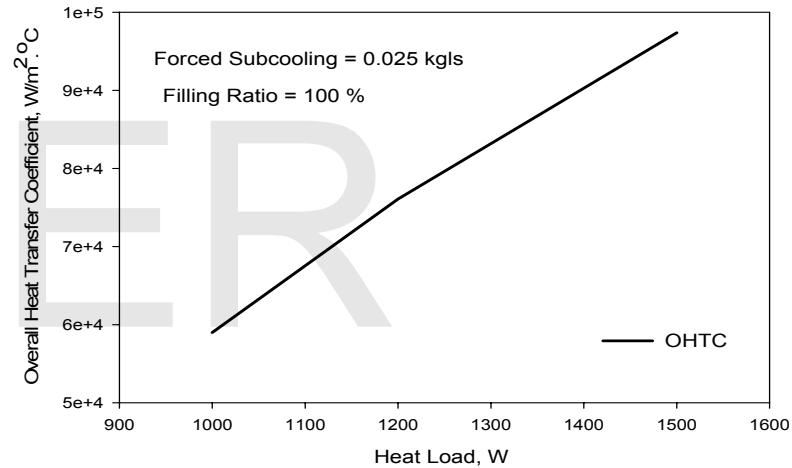


Fig. 8. Overall Heat Transfer Coefficient (OHTC) of (FFDTT) for Different Loads at Steady State Operation at Constant Forced Flow rate.

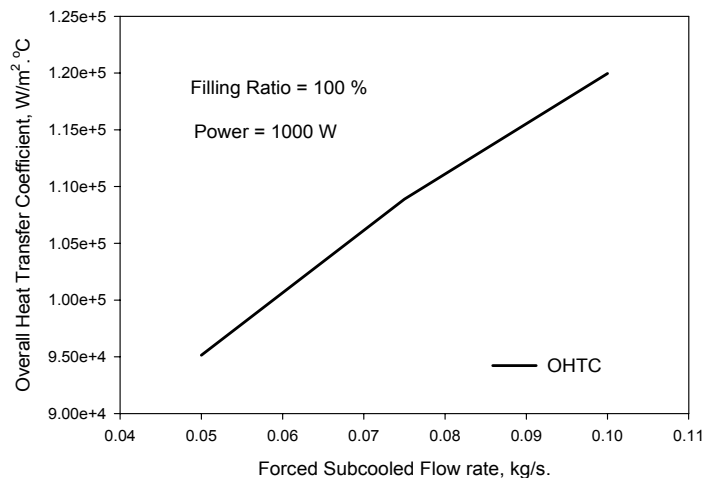


Figure 9. Overall Heat Transfer Coefficient (OHTC) of (FFDTT) for Different Forced Flow rate at Steady State Operation at Constant Load.

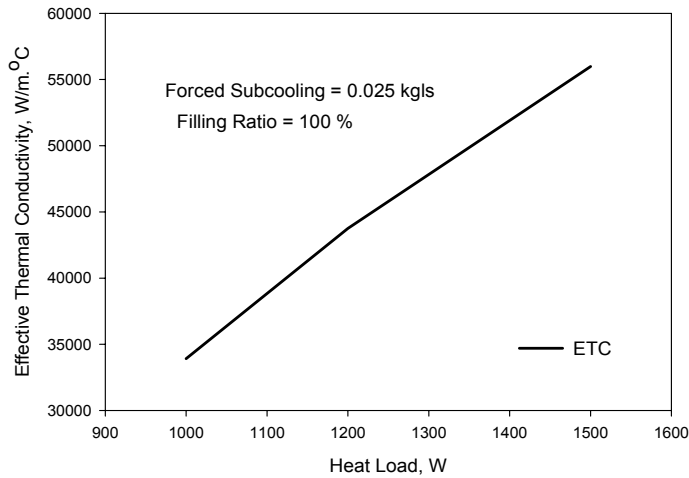


Fig. 10. Effective Thermal Conductivity (ETC) of (FFDTT) for Different Loads at Steady State Operation at Constant Forced Flow rate.

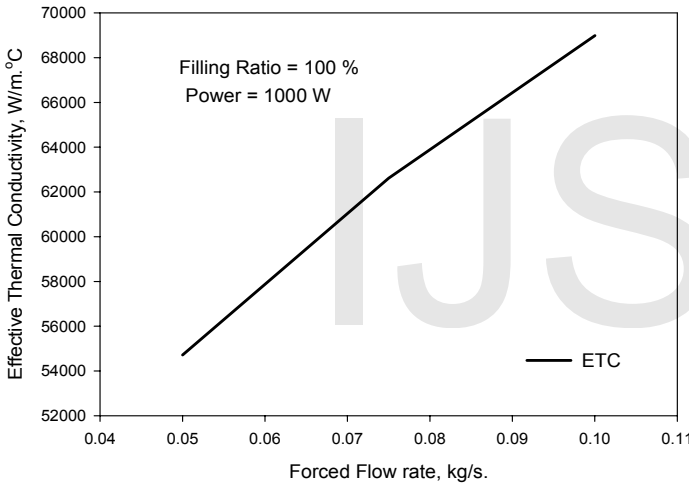


Fig. 11. Effective Thermal Conductivity (ETC) of (FFDTT) for Different Forced Flow rate at Steady State Operation at Constant Load.

4.4 The equivalent overall heat transfer coefficient (OHTC)

Thermosyphon can be considered as single conductor transporting heat from heat source to heat sink. Subsequently, the performance of that conductor (thermosayphon) can be hypothetically expressed by an overall heat transfer coefficient. Therefore, the equivalent overall heat transfer coefficient is generally defined as:

$$U = qax / \Delta Ti \tag{36}$$

The temperature difference ΔTi between the mean evaporator and condenser inner wall regions, is given as:

$$\Delta Ti = t_{we,i} - t_{wc,i} \tag{37}$$

Therefore, the previous presented temperature profiles are of great importance in calculating the equivalent overall heat

transfer coefficient of thermosyphon.

The overall heat transfer coefficient is plotted in Figs. 8 and 9 viruses the axial heat flux, and various forced sub cooling. As shown in the figure, there is a direct relation between the overall heat transfer coefficient and axial heat flux. Although, the temperature difference ΔTi increases too, as the heat flux increase, however, the temperature difference increases considerably less than the increase in the rate of heat transfer. This means that the rate of change dq in the heat flux is relatively higher than the rate of increase $d(\Delta Ti)$ of temperature difference as clearly indicated in Fig. 8, which gives the relation between the axial heat flux and temperature difference. Consequently, the equivalent overall heat transfer coefficient increases. Also, the addition of more heat augments the vapor bubble nuclei, size and number of detached bubbles from heat surfaces, which ultimately enhance the turbulence, and the subsequent equivalent heat transfer coefficient of thermosyphon. Fig. 9 also, illustrates the effect of changing the forced sub cooling, and as seen the overall heat transfer coefficient increased when forced sub cooling increases for the same value of heat flow.

4.5 Effective thermal conductivity (ETC)

The effective thermal conductivity is an appropriate means for measuring the performance of a forced-flow two-phase closed thermosyphon and it is determined as:

$$K_{eff} = qax * Lp / \Delta Ti \tag{38}$$

$$Lp = 1/2(L_{evap} + L_{cond}) + L_{ad} \tag{39}$$

As can be seen from this equation that the effective thermal conductivity of thermosyphon is not referred to, the working fluid nor the pipe metal thermal conductivity. It mainly depends on the heat flow and the dimensions of the thermosyphon. Fig. 10 presents the variation of the effective thermal conductivity of the current vertical water/copper thermosyphon versus heat load at constant forced sub cooling. As it is observed from figure, the effective thermal conductivity is extremely dependent on the heat load, where it increases with heat load rise. Also, Fig. 11 which gives the relation ship between the effective thermal conductivity and the forced sub cooling at constant heat load, and it was shown that it is increased during increasing in the forced sub cooling.

In comparison with the thermal conductivity of copper and silver the thermal conductivity of the forced flow boiling double-tube thermosyphon is more than 170 folds higher than that for copper and silver. This can be explained by the principle of phase change in thermosyphon. This clearly shows the thermosyphon has the capability to transport a high amount of heat energy.

5. Conclusion

A mathematical model has been developed for forced flow double-tube closed two-phase thermosyphon, in order to convert pool boiling to forced flow boiling mechanism inside thermosyphon in addition to describe thermal behavior in steady-state regime. An influence analysis of the thermosyphon response to various operating conditions has been performed. It has been shown that the model developed in this work can be used as an efficient tool in designing thermosyphons. The current of the derived mathematical model accurately predict the local wall and fluid temperature, evaporator heat transfer coefficients, the overall heat transfer coefficient as well as the lengths of the non-boiling and boiling region also the vapor quality. It was found that the single phase heat transfer coefficient is higher than the two-phase heat transfer coefficient refer to the forced sub cooling.

NOMENCLATURE:

A	area, (m ²).
c _p	specific heat, (J/kg. °C).
d	diameter, (m).
d _h	hydraulic diameter, (m).
g	acceleration of gravity, (9.81 m/s ²).
h	heat transfer coefficient, (W/ m ² . °C).
h _{fg}	latent heat of vaporization, (j/kg. °C).
k	thermal conductivity, (W/m. °C).
k _{eff}	effective thermal conductivity, (W/m. °C).
L	length, (m).
m.	water flow rate, (kg/s).
P	pressure, (N/ m ²).
q	heat flux, (W/ m ²).
Q	heat load, (W).
t	temperature, (°C).
ΔT	temperature difference, (°C).
u	overall heat transfer coefficient, (W/ m ² . °C).
U _{eq}	equivalent overall heat transfer coefficient (W/m ² °C).
x	Axial distance, (m).

Greek Symbols

α	void fraction.
ε	configuration (geometry) factor.
μ	dynamic viscosity, (N.s/ m ²).
β	thermal expansion coefficient (K-1)
ν	kinematic viscosity, (m ² /s).
ρ	density, (kg/m ³).
σ	surface tension, (N/m).

Dimensionless Groups

Pr	Prandtl number, (C _p μ/k).
Ra	Rayleigh number, (βgdi ⁴ qe/κlα _l ν).
Re	Reynolds number, (4q L/hfg μ).

Subscripts

a	adiabatic,
am	ambient,
ax	axial,
c	condenser,
cw	cooling water
e	evaporator,
eq	equivalent,
f	film,
i, i	inner, initial,
j	jacket,
l	liquid
o	outer,
m	mean,
nc	natural convection,
p	pool,
r	radial,
sat	saturation,
tp	two-phase
v	vapor,
w	wall.

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