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Experimental investigation and comparison of heat transfer coefficient of a two phase closed thermosyphon

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Abstract

In this study, we investigated and reviewed the heat transfer equations of the evaporator and condenser of a two phase closed Thermosyphon as well as the differences between this equations for a working fluid and different conditions. The heat transfer limits of a two phase closed thermosyphon such as sonic, flooding (or entrainment) dry-out and boiling are investigated . A good agreement is observed between analytical results of this study and the analytical and experimental results of those available in the open literature. The heat transfer limits are very important in thermosyphons and have to be calculated accurately while the input heat must be lower than the heat transfer limits for the operations of thermosyphon without any problem. Also by using the experimental results a semi empirical equation for heat transfer coefficient of evaporator was proposed.

Keywords: Thermosyphon; Heat transfer limit; Flooding limit; Boiling limit; Dry-out limit.

1. Introduction

A two phase closed thermosyphon is a heat pipe needing no wicks to return the condensate working fluid from the condenser section to the evaporator one because of the gravity. However, the evaporator must be positioned at the lower part of the gravitation field, compared to the condenser. But, because of its simple structure, stable operating condition during steady state, the characteristic of a thermal diode and wide operating temperature range, it was applied to many industrial fields. The two phase closed thermosyphon is widely used because of its simple structure compared to the other types of heat pipes. Therefore, thermosyphons are being used in many applications such as : heat pipe, heat exchangers, cooling of electronic components, solar energy systems, deicing and snow melting. The analysis of two phase flow heat transfer in a two phase closed thermosyphon is very complicated. The condensation of heat transfer in the condenser was based on the Nusselt model and Faghri [1] compared it with some correlations. While in the evaporator, the nucleate boiling model made by Rohsenow in reference [2] was applied to predict the heat transfer coefficients and compared with the correlation suggested by Imura et al. [3].
Thermal performance of the thermosyphon is affected by many factors, such as the type of working fluid, filling ratio (ratio of volume of working fluid to volume of evaporator Section), aspect ratio (ratio of evaporator section length to inside diameter of pipe), inclination angle (from horizontal axis), operating pressure (or corresponding saturation temperature), and length of various sections of the pipe. Various heat transfer limits occur as a result of the dryout of liquid film in the evaporator. When the heat input to the evaporator is relatively high, intensive liquid film evaporation causes the vapor flow to move upwards, quickly, exceeding the flooding limit. The drying out of the liquid film may result from liquid

droplet entrainment from the liquid film by the interfacial shearing effect of high speed vapor flow, resulting in the entrainment limit.

2. Models of equations

The rate of heat transfer to the evaporator section of a thermosyphon can be calculated from the Eq. (1)

$$Q_{in} = V^2 / R = RI^2 \tag{1}$$

The rate of heat remove from the condenser section by the coolant water is obtained from Eq. (2)

$$Q_c = \dot{m} c_p (T_{o,w} - T_{i,w}) \tag{2}$$

The heat losses from the evaporator and the condenser are obtained by

$$Q_{loss} = \sigma A_t \varepsilon (T_{ins}^4 - T_\infty^4) + h A_t (T_{ins} - T_\infty) \tag{3a}$$

where ε = 0.085 and σ = $5.669 \times 10^{-8} W/m^2.K$ and free convection heat transfer coefficient on a vertical cylinder obtained from [2]:

$$Nu = \frac{hL_t}{k} = \left\langle 0.825 + \frac{0.387\, Ra^{1/6}}{\left[1 + \left(\frac{0.492}{\Pr}\right)^{9/16}\right]^{8/27}} \right\rangle^2 \tag{3b}$$

The experimental heat transfer coefficient of laminar film in condenser of a heat pipe is:

$$h_c = \frac{Q_c}{A_c (T_v - T_c)} \tag{4}$$

And the analytical correlation of the condenser section in reference [2] is given by:

$$\bar{h}_c = 0.943 \left\{ \frac{\rho_l g k_l^3 (\rho_l - \rho_v) \left[h_{fg} + 0.68 c_{pl} (T_v - T_c) \right]}{\mu_l L_c (T_v - T_c)} \right\}^{1/4} \tag{5}$$

This equation can be written in dimensionless form of average Nusselt and Reynolds number for liquid film:

$$Nu^\bullet = \frac{\bar{h}_c}{k_l} \left[\frac{\nu_l^2}{g} \left(\frac{\rho_l}{\rho_l - \rho_v} \right) \right]^{1/3} = 0.925 \operatorname{Re}_{l,\max}^{-1/3} \qquad \operatorname{Re}_{l,\max} \prec 325 \tag{6}$$

$$\operatorname{Re}_{l,\max} = \frac{Q_c}{\pi D \; \mu_l h_{fg}} \tag{7}$$

Several correlations of heat transfer coefficient in evaporator section of a two phase closed thermosyphon were investigated. Rohsenow in reference [2] reported a model for nucleate boiling as:

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$$q_s = \mu_l A h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left(\frac{c_{pl}(T_e - T_v)}{C_{sf} h_{fg} \Pr^n} \right)^3 \tag{8}$$

where $q_s = h_e (T_e - T_v)$

[For example, for water-copper $C_{s,f} = 0.0068$, $n = 1.0$ where $C_{s,f}$ is the solid-fluid constant]

And the correlation that Imura et al. [3] suggests for the evaporator section of a two phase closed thermosyphon is:

$$\bar{h}_e = 0.32 \left(\frac{\rho_l^{0.65} k_l^{0.3} c_{pl}^{0.7} g^{0.2} q_e^{0.4}}{\rho_v^{0.25} h_{fg}^{0.4} \mu_l^{0.1}} \right) \left(\frac{p_v}{p_{atm}} \right)^{0.3} \tag{9}$$

In which q_e is the applied heat flux in the evaporator section.

Although thermosyphons are very efficient heat transfer devices, they are subject to a number of heat transfer limitations. These limitations determine the maximum heat transfer rate that a thermosyphon can achieve under certain working conditions. These limits can arise from the cessation of capillary pumping and the attainment of sonic velocity in the heat pipe vapor.

The sonic limit is the highest possible heat transport rate that can be sustained in a heat pipe for a specific vapor temperature at the evaporator end of a heat pipe. The sonic limit is reached when the vapor leaving the evaporator attains sonic velocity. An expression of sonic limit is given by Dunn and Reay [4]:

$$Q_{c,90} = \rho_v h_{fg} A_v \sqrt{\frac{\gamma R_v T_v}{2(\gamma + 1)}} \tag{10}$$

where $Q_{c,90}$ is the heat transfer rate limits at a vertical position.

The correlation of flooding limit investigated by Faghri [1] is given as:

$$Q_{c,90} = K h_{fg} A \left[g \sigma (\rho_l - \rho_v) \right]^{1/4} \times \left[\rho_v^{-1/4} + \rho_l^{-1/4} \right]^{-2} \tag{11}$$

where

$$K = \left(\frac{\rho_l}{\rho_v} \right)^{0.14} \tanh^2 (Bo)^{1/4} \tag{12}$$

The dry out limit is reached in a two phase closed thermosyphon at the bottom of the evaporator section when the fill charge ratio is very small. For this, the entire amount of working fluid should be circulating throughout the thermosyphon.

Rosler et al. [5] used a thermosyphon with R113 as the working fluid employing filling ratios between 0.02 and 0.4 and experimentally and theoretically correlated an equation for dry out limit which is a function of filling ratio, the properties of fluid and heated length:

$$FR = 1 - \frac{1}{1 + C \left[\frac{(Q_{c,90} / A h_{fg})^2}{2 \sigma \rho_v} \left(\frac{3 Q_{c,90} \mu_l L_e}{A \rho_l (\rho_l - \rho_v) g h_{fg}} \right)^{1/3} \right]^{3/4}} \tag{13}$$

where C is the exponential constant. This equation can be solved for $Q_{c,90}$ as a function of filling ratio and fluid properties and the evaporator length.

Busse [6] carried out a two dimensional analysis of viscous limit in a two phase closed thermosyphon and found that

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$$Q = \frac{D_e^2 h_{fg} \rho_v P_v}{64 \mu_v L_v} \tag{14}$$

where P_v *and* ρ_v are the pressure and density of vapor and this expression changes with vapor characteristics and inside diameter of evaporator.

The transfer of heat into a heat pipe produces a temperature gradient across the wall of the tube. At the interface between the wall and the vapor space, the liquid temperature on the wall is equal to or greater than that of the adjacent saturated vapor. The liquid temperature rises steadily with distance from the interface, reaching a maximum value at the outer surface. The heat transfer rate corresponding to the incipient boiling condition is called the boiling limit.

Grobis et al [7] proposed empirical correlations for the maximum radial heat flux at the boiling limitation in a two phase closed thermosyphon.

$$\frac{Q_{c,90}}{Q_{c,\infty}} = C^2 \left[0.4 + 0.006 D_i \sqrt{\frac{g(\rho_l - \rho_v)}{\sigma}} \right]^2 \tag{15}$$

where $Q_{c,\infty}$ is the critical heat flux for pool boiling that is:

$$Q_{c,\infty} = 0.142 A \sqrt{\rho_v} \left[g \sigma (\rho_l - \rho_v) \right]^{1/4} \tag{16}$$

And

$$\begin{aligned} C &= 0.538 \left(\frac{D_i}{L_c} \right)^{-0.44} \left(\frac{D_i}{L_e} \right)^{0.55} \psi^{0.13} \quad for \quad \psi \le 0.35 \\ C &= 3.54 \left(\frac{D_i}{L_c} \right)^{-0.44} \left(\frac{D_i}{L_e} \right)^{0.55} \psi^{-0.37} \quad for \quad \psi \succ 0.35 \end{aligned} \tag{17}$$

3. Description of experiments procedures

For experimental investigation of heat transfer specifications in two phase closed thermosyphon a setup has been fabricated. Figure 1 shows the appearance of a two phase closed thermosyphon for testing and investigating the thermal performance.

In these experiments two copper tubes of 1000 mm length with inside diameter of 15 and 25 mm and 2 mm thickness were employed (Figure 1).

The working length of the thermosyphon consists of three parts; a lower part of 430 mm as the evaporator section, the middle part of 160 mm as the adiabatic section, and a upper part of 410 mm as the condenser section. The 410 mm long water jacket was surrounding the condenser section. Inlet and outlet connections located obliquely across each other to introduce swirl flow. Eight Ni-Cr thermocouples were installed mechanically to the surface of the evaporator and adiabatic and condenser to monitor the temperature distribution. A personal computer and a data logger were used to show the temperatures measured by thermocouples. The two phase closed thermosyphon was surrounded by 40 mm thickness of glass wool for insulating and stopping the heat transfer to the environment. Table 1 shows the specifications of two phase closed thermosyphon for this study.

An electrical resistance of 1000 W was employed to produce the heat input of evaporator and the accuracy of monitoring for voltage and electrical current was $\pm 2\%$. The temperature of inlet and outlet coolant water was measured by digital thermometer and the coolant mass flow rate of water was measured by a Rotameter. Also a vacuum pump was employed to produce vacuum condition inside the thermosyphon.

The measured parameters were: heat input of the evaporator section Q_{in}, heat output of the condenser section Q_c, temperature of inlet and outlet of the coolant water, the coolant mass flow rate of water, and the surface temperatures of evaporator, adiabatic and condenser sections. The experimental limits were as follows:

- Heat input (Q_{in}) between 100 and 200 W
- Aspect ratio (AR) 28.6 and 17.2
- Filling ratio (FR) between 15 and 100%
- Inclination angle (φ) between 15 $^{\circ}$ and 90 $^{\circ}$
- Coolant mass flow rate ($\dot{m}$) between 0.00997 and 0.03 kg/s

At first, the experiments were performed taking AR as 28.6 for various filling ratio, inclination angle, heat input, and coolant mass flow rate and measure the pointed parameters and then the experiments were repeated for AR=17.2 .

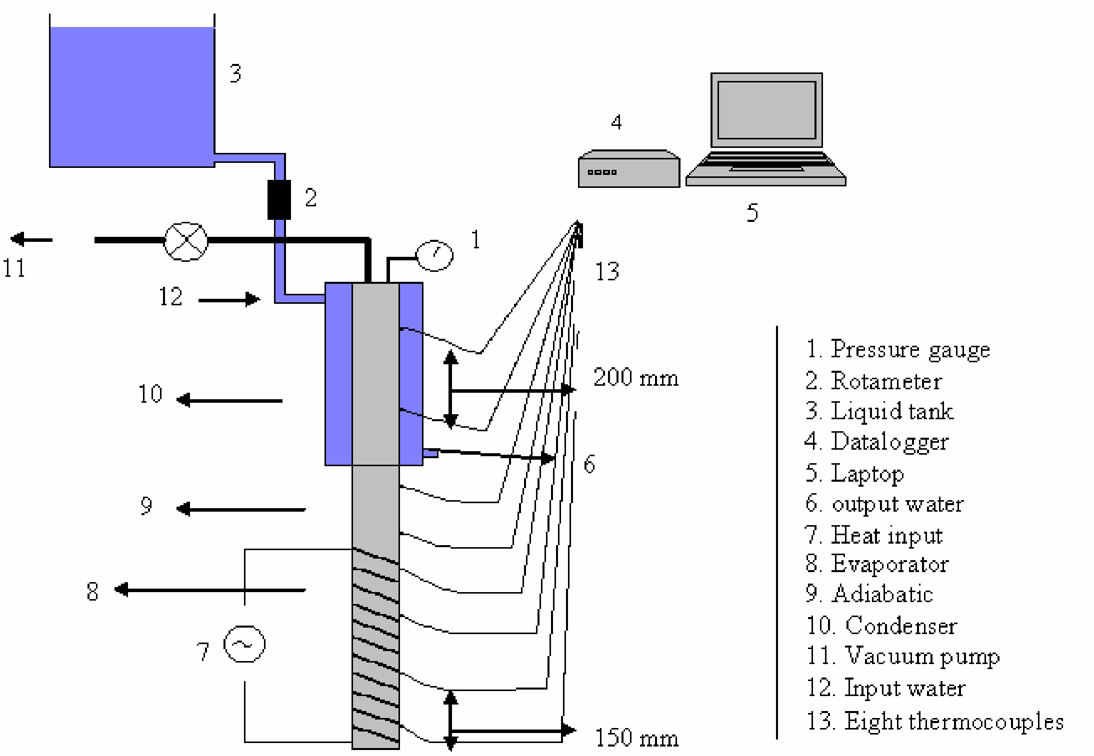

Figure 1. Experimental appearance of thermosyphon

Table 1. Specifications of the two phase closed thermosyphon

ID	OD	Evap L	Adia L	Cond L	Material	Working Fluid
15 mm	17 mm	430 mm	160 mm	410 mm	Copper	ethanol
25 mm	27 mm	430 mm	160 mm	410 mm	Copper	ethanol

4. Results and discussion

Figure 2 illustrates the experimental data of Park et al [8] for C_6F_{14} as the working fluid and three fill charge ratio. It can be seen that the Nusselt number increases by increasing the fill charge ratio. It can be explained that during the steady state condition, the heat transfers directly from the high temperature working fluid to the lower part of the condenser without evaporating process.

Noie et al. [9] investigated the heat transfer coefficient of condenser versus the inclination angle for different filling ratios. They used Eq(4) and found that the condensation heat transfer increases as the filling ratio increases and the maximum condensation heat transfer coefficient takes place at φ=30° for FR=22% and 30% and at φ=45° for FR=15%. Figure 3 shows their experimental data for heat transfer coefficient.

Figure 4 shows the heat transfer coefficient in evaporator versus heat flux for correlations of Rohsenow [2], Eq(8) for C_6F_{14}, $C_{s,f} = 0.004$ and Imura [3], Eq (9) and the experimental data of Park [8] for various fill charge ratios. This Figure shows that in about 1000-30000 (W / m^2) evaporator heat flux, the heat transfer coefficient increases from 300-3400 (W/m^2k) and the experimental data are at good agreements with Rohsenow [2] correlation.

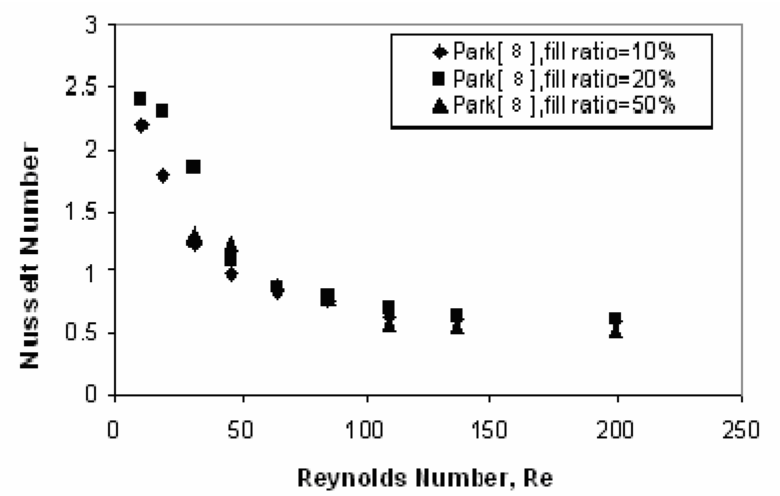

Figure 2. Nusselt number versus Reynolds number for three fill charge ratio for C_6F_{14} at $T_v = 50℃$

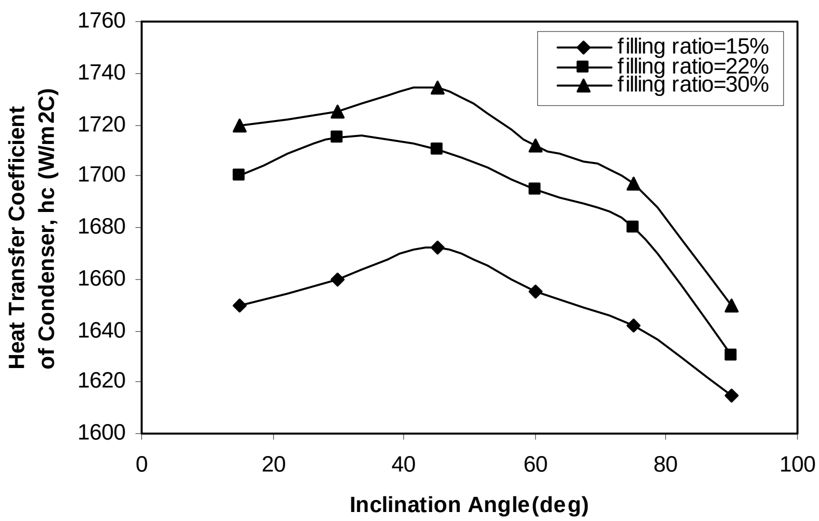

Figure 3. Heat transfer coefficient of condenser versus inclination angle for different filling ratios[9]

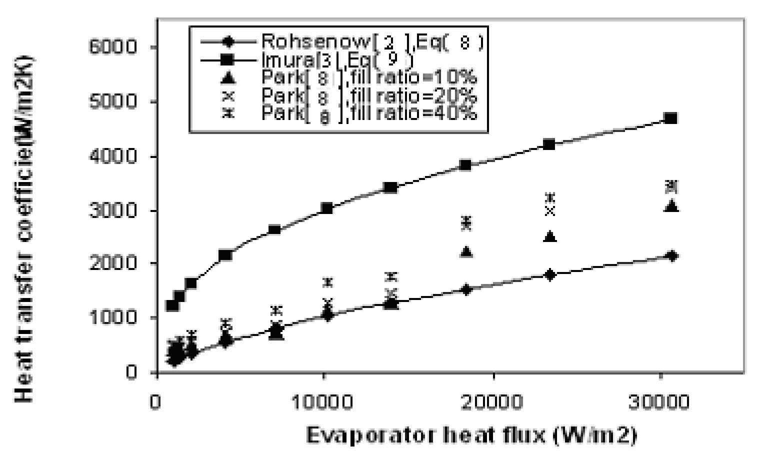

Figure 4. Heat transfer coefficient versus heat flux in evaporator for different fill charge ratios for C_6F_{14} *at* $T_v = 50℃$

Noie [10] investigated the heat transfer coefficient of evaporator of a two phase closed thermosyphon versus the average temperature of evaporator surface for different filling ratios and compared them with the correlations of Rohsenow [2] and Imura [3]. For his investigation, the experimental results were found to be in reasonable agreement with correlations of Rohsenow [2] and Imura [3]. Figure 5 shows the experimental data of Noie [10] for heat transfer coefficient of evaporator.

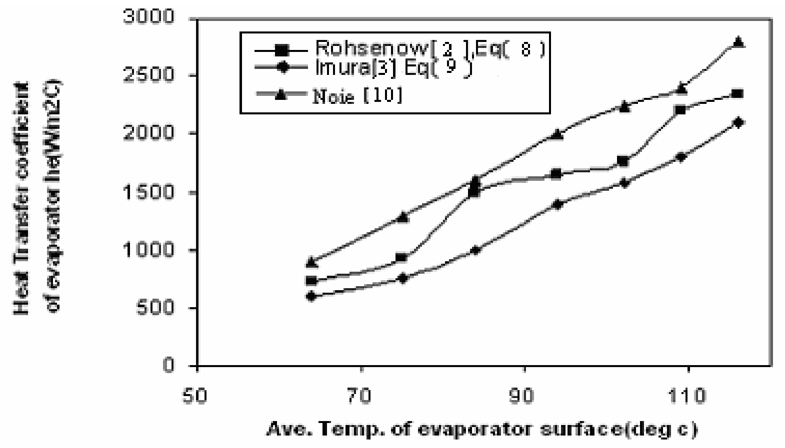

Figure 5. Heat transfer coefficient of evaporator versus average temperature of evaporator surface for different filling ratios

There is no scarcity of experimental data on limiting heat pipe heat transport rates in the literature. Such data are frequently presented along with analytical predictions of heat transport limits. In principle, if the calculated and experimental limits are in a reasonable agreement, the analytical methods may be considered to be valid.

Figure 6 presents the comparison of experimental results of Rosler et al [5] with the theoretical predictions of Eq (13) by the dryout regions and R113 as the working fluid. The continues lines at Figure 6 show the theoretical results for three different vapor temperature. As it can be seen, for small fill ratios ($FR \leq 80\%$), the experimental data [5] agree very well with the theoretical predictions [5].

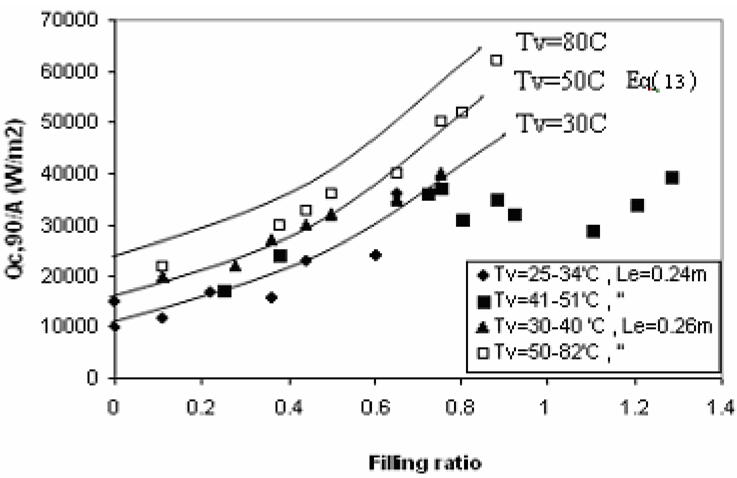

Figure 6. Dryout limit of experimental data[5] and the Eq(13) for R113 for different Tv

Figure 7 shows the comparison of boiling limit from Grobis [7], Eq (15) for R113 as the working fluid at $T_v = 30^\circ C$ and the experimental data [7]. It can be seen that there are agreements with the analytical and the experimental data.

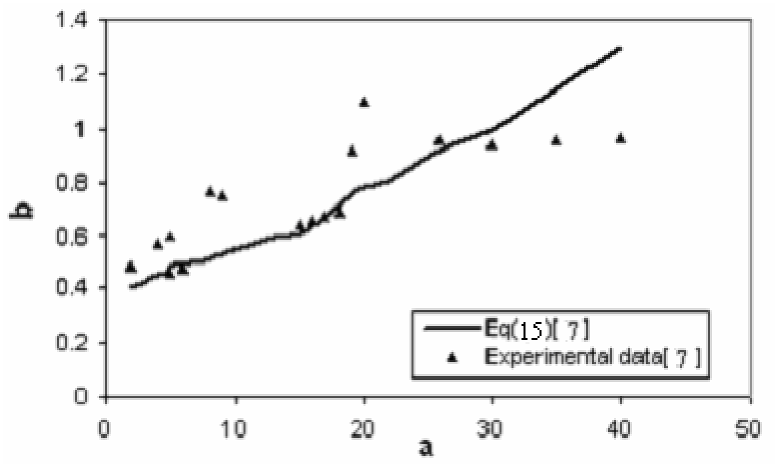

Figure 7. Boiling heat transfer for water in two phase closed thermosyphon where $a = R\sqrt{\frac{g(\rho_l - \rho_v)}{\sigma}}$ and $b = \frac{1}{C}\sqrt{\frac{Q_{c,90}}{Q_{c,\infty}}}$

5. A new correlation for heat transfer coefficient of evaporator

By doing the experiments, it is clear that changing Temperature and some thermodynamic properties affect the heat transfer coefficient of evaporator inside the thermosyphon.

The method of fitting used in this paper is the least square method. This method fits a set of data points (x_i, y_i) to a function that is a combination of any number of functions of the independent variable x. The goal of nonlinear regression is to determine the best-fit parameters for a model by minimizing a chosen merit function. Where nonlinear regression differs is that the model has a nonlinear dependence on the unknown parameters, and the process of merit function minimization is an iterative approach. The process is to start with some initial estimates and incorporates algorithms to improve the estimates iteratively. The new estimates then become a starting point for the next iteration. These iterations continue until the merit function effectively stops decreasing. The nonlinear model to be fitted can be represented by:

$$y = y(x; a) \tag{18}$$

The merit function minimized in performing nonlinear regression the following:

$$\chi^2(a) = \sum_{i=1}^{N}\left\{\frac{y_i - y(x_i; a)}{\sigma_i}\right\}^2 \tag{19}$$

where σ_i is the measurement error, or standard deviation of the *i*th data point. For understanding how the results calculated we have:

The *ith* predicted, or fitted value of the dependent variable Y, is denoted by $\hat{Y}_i$. This value is obtained by evaluating the regression model $\hat{Y} = f(X, \hat{\beta}_j)$, where $\hat{\beta}_j$ are the regression parameters, or variables. Then the residuals $(Y_i - \hat{Y}_i)$ and sum of the residuals $\sum_{i=1}^{n}(Y_i - \hat{Y}_i)$ calculated and then, the average of residuals and residual of sum of squares calculated.

SSE = Residual or Error Sum of Squares (Absolute) = $\sum_{i=1}^{n}(Y_i - \hat{Y}_i)^2$

SSE_R = Residual or Error Sum of Squares (Relative) $= \sum_{i=1}^{n}[(Y_i - \hat{Y}_i)^2 * W_i]$ where $W_i = 1/\sigma_i^2$ normalized so that $\sum_{i=1}^{n} W_i = n$.

σ_i = the standard deviation of the i_{th} data point Y_i and n is the number of data points, or observations.
The principle behind nonlinear regression is to minimize the residual sum of squares by adjusting the parameters $\hat{\beta}_j$ in the regression model to bring the curve close to the data points. This parameter is also referred to as the error sum of squares, or *SSE*. If the residual sum of squares is equal to 0.0, the curve passes through every data point.
Thus, the correlation that proposed to indicate the effect of those parameters on the heat transfer coefficient of evaporator is:

$$h_e = a\left(\frac{q}{h_{fg}}\right)^b \left(\frac{\mu_l}{\sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}}\right)^c (C_p)(pr)^d(\rho_l)^e \tag{20}$$

In Eq (20) the constants a, b, c, d and e are undefined and by using software as Datafit which fits the results of experiment from one to more independent variables that in Table 2 the value, upper limit and lower limit of constants in Eq (20) was shown. Hence, by analyzing the results of experiments, Eq (20) converts into Eq (21) as

$$h_e = 8.09 \frac{k_l^{1.59}\ \sigma^{1.09}\ \rho_l^{0.66}\ q^{0.94}}{h_{fg}^{0.94}\ (\rho_l - \rho_v)^{1.09}\ C_p^{0.59}\ \mu_l^{3.77}\ g^{1.09}} \tag{21}$$

Table 2. Value and limits of constants in Eq (20)

Variable	Value	Lower limit	Upper llimit
a	8/09E-20	-372/235	372/397
b	0/939901	0/380878	1/498924
c	-2/1827	-281/978	277/6128
d	-1/59859	-335/766	332/5689
e	0/661656	-715/295	716/6181

Figure 8 shows the comparison between the experimental and theoretical results of the heat transfer coefficient for AR= 17.2 FR=80% , φ=90° and $\dot{m} = 0.0199 kg/s$ for water as working fluid and it is clear that there are significant agreements with them.

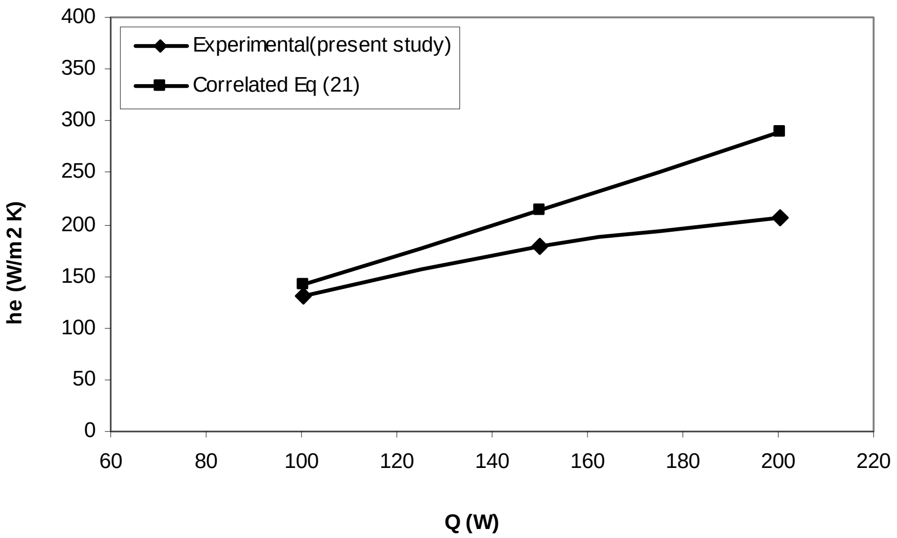

Figure 8. Comparison between experimental and correlated Eq

6. Conclusion

In this study the heat transfer coefficient of condenser and evaporator and the heat transfer limits were investigated. The analytical heat transfer coefficient and the heat transfer limits and the experimental data from several researches were compared with each other. Based on these findings, the following conclusions can be drawn:

1. There are good agreements between analytical and experimental results of previous works.
2. The condensation and evaporation heat transfer coefficient increases as filling ratio increases.
3. Maximum of the heat transfer coefficient of condenser is at about $40° < \varphi < 50°$.
4. The critical heat flux as the dryout limit increases as the fill charge ratio increases.
5. A new correlation for predicting the heat transfer coefficient of thermosyphon according to thermodynamic properties of working fluid was proposed and there is a good agreement between its results and the experimental results.

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