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Theoretical Study of Various Fluids on the Transient Performance of a Two-Phase Closed Thermosyphon (TPCT)

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Abstract: Passive cooling heat exchangers are important for a nuclear reactor. Due to many advantages of passive cooling mechanism it may be used in nuclear reactors by employing two-phase closed thermosyphon (TPCT) during shutdown regime. Many thermosyphons can be penetrated in cooling system so that, the condenser section is exposed to air and the evaporator (evaporator) section immersed into water pool. Our study is proposed to predict the thermal behaviour of TPCT with different fluids, Water, Toluene, Acetone and Heptane. The fluid which realizes better heat transfer characteristics prediction is our study objective. The mathematical model is developed to predict thermal behaviour of these fluids to classify the most appropriate fluid. Water, acetone, heptane, and toluene are selected as working fluids in this study. Engineering Equation solver is used to formulate our model with finite difference numerical method. The results show that, Acetone has the highest heat transfer coefficient and the lowest thermal resistance in evaporator section. Water has the highest critical heat flux and highest evaporator surface temperature. Water as well has higher heat transmission coefficient in the evaporator section after Acetone. It has also low wall temperature and highest heat transfer coefficient in condenser section. Water heat transfer coefficient is three times more than the other fluids and more output cooling capacity. For the same size of TPCT, while the Acetone is the lowest thermal resistance, but water is more preferable working fluid because it is more available with low thermal resistance as well and highest cooling capacity which enables the TPCT to have more cooling capacity Validation of the proposed theoretical model is performed and showed a good evaporator and condenser surface temperature agreement with the empirical results.

Keywords: boiling; condensation; saturation pressure; saturation temperature; thermosyphon

1 INTRODUCTION

Heat pipes are the most common passive, capillary-driven of the two-phase systems. The liquid-to-vapour phase transition (boiling/evaporation and condensation) of a working fluid is a component of two-phase heat transfer. Simple two phase closed thermosyphon is a natural circulation heat pipe that, consists of a sealed pipe charged with working fluid under a vacuum condition without wick material [1]. Thermosyphon is passive two-phase heat transfer loops increase the thermal conductivity between a heat source and a heat sink by utilizing the very efficient thermal transport mechanism of evaporation and condensation. Additionally, thermosyphon heat pipes are found to have a number of general advantages that make them suitable for a variety of uses, including heating buildings with solar energy [2], water heaters, furnaces, and boilers, as well as cooling applications (such as cooling nuclear reactors, turbine blades, transformers, electronics, and internal combustion engines). In the last few decades, the thermosyphon systems were used as a passive cooling system for the spent fuel pool and nuclear reactors. Many academics have conducted theoretical and experimental research on the fundamental characteristics of two-phase closed thermosyphons. Thermosyphon functioning is affected by a number of variables, including fill ratio, working fluid, tilt, geometry, heat input, cooling water flow rate, and others. Patil Aniket D., and Yarasu Ravindra B. [3] in which they focused on aspects such as filling ratio, aspect ratio, heat load, mass flow rate, and inclination angle, all of which affect thermosyphon thermal performance, publish a review study. Many investigations were implemented in order to analyse and improve the thermosyphon's thermal performance, theoretically and/or experimentally to study the effect of mathematical Modelling, working fluid, surface geometry and mechanical modification. New heat transfer augmentation approaches such as resurfacing, ultrasonic wave utilization, and CFD analysis are also enlightening. Due to the extended application of heat pipe in the recent years, noticeable interest in the study of thermosyphon design and performance. These studies were carried out (but not limited to) by Thanya, et al [4], Filippo Cataldo and Yuri Carmelo Crea [5], M. G. Hammouda, et al [6], Ghasem, et al [7], and Myeongjin Kim, et al [8]. Several investigations were presented to study the effect of working fluids on the thermosyphon. Mozumder, et al. [9] designed, built, and tested a tiny heat pipe using the working fluids acetone, methanol, and water. Thermal resistance and total heat transfer coefficient were used to evaluate the heat pipe's performance. H. Jouhara, et al. [10] utilizing water and the dielectric heat transfer liquids FC-84, FC-77, and FC-3283 carried out a thermosyphon thermal performance experiment. S. M. Sadrameli, et al. [11] looked into the impact of working fluid occupancy inside heat pipes and the amount of heat input on thermosyphon performance. The operating fluid in a steel thermosyphon with a 16 mm diameter was toluene. Evaporator, adiabatic, and condenser section lengths were, respectively, 10, 23, and 17 cm. Russo, et al [12] investigated experimentally the thermal performance of various working fluids in thermosyphons that can be employed in thermal control of electronic equipment. Acetone, water, ethanol, and methanol were used as working fluids. A. Jay, D. Pingale, et al [13] constructed a two-phase closed thermosyphon (TPCT) with a 1 m length and outer diameter 19mm that is filled with different fluids (Propylene Glycol and Ethylene Glycol). Different fill volume ratios (40 and 60 %), and heat inputs (60 to 80 °C) were used to study the performance of thermosyphon. Rafal Andrzejczyk [14] investigated how several factors affected a wickless heat pipe's performance. With different filling ratio values (0.32, 0.51, 1.0), three working fluids—Ethanol, Water, and SES36 (1, 1, 1, 3, 3-Pentafluorobutane) were investigated. Abdelrahim Abusafa and Aysar Yasin [15] investigated a two-phase closed thermosyphon system for cooling high heat flux electronic equipment. Using R-134a as a working fluid, the performance of the thermosyphon was studied. The influence of heat flux and the refrigerant pressure were investigated on the boiling section heat transfer coefficient. H. Z. Abou-Ziyan et al. [16] investigated the thermal performance of a two-phase closed thermosyphon using water and R134a as the working fluid in both stationary and vibratory situations. The thermal performance of an inclined two phase closed thermosyphon with distilled water and aqueous solution of n-Butane as a working fluid was explored by M. Karthikeyan, et al. [17]. Analytical calculations were made by Shwetabh Singh, et al. [18] to determine the contribution of the working fluid's thermo physical parameters to the formation of entropy. A calculation of entropy has been made using acetone, pentane, heptane, ammonia, pentane, and water as working fluids. Hussain Saad Abd, et al. [19] studied the wickless heat pipe's performance properties, including thermal power, working fluids, temperature change with varying input powers, thermal resistance, and heat transfer coefficient, which were shown experimentally. Various heat loads (from 20 to 60 W) and a filling ratio of 50 % were employed with the working fluids methanol and ethanol. M. M. Sarafraz, et al. [20], conducted a series of tests to assess the thermal performance and heat transmission of n-pentane-acetone and n-pentane-methanol mixtures inside a gravity-assisted thermosyphon heat pipe. The impact of a number of variables, including the input heat to the evaporator section, the carrying fluid filling ratio, the tilt angle of the heat pipe, and the carrying fluid type, on temperature distribution and heat pipe performance was examined. D. Mishkinis, et al. [21] carried out a review of prospective solutions for the development program "High Temperature LHP" and selected potential working fluids (including two-component combinations) and LHP materials for the envelope and wick. The typical working fluids for LHP are ammonia and propylene. Results show that if the temperature is close to 125 °C, organic liquids including acetone, ethanol, and methanol are incompatible with Ni wick due to a highly severe breakdown process. In several experiments and theoretical studies, the transient behaviour of closed two-phase thermosyphons was investigated to determine its impact on thermosyphon performance. The following list includes a few of these studies. Hichem Farsi, et al. [22] performed an experimental and theoretical investigation to examine the two-phase closed thermosyphon (TPCT) behaviour in transient regimes. Two types of TPCT responses were shown by experimental data. The mathematical model had been constructed to obtain an analytical expression of the system response time. In double-phase closed thermosyphon, Zhongchao Zhao and colleagues [23] studied thermal performance of phase change heat and mass transport in transient state by using computational fluid dynamics (CFD). To simulate condensation and evaporation in the thermosyphon at various heating inputs, a computational fluid dynamic model based on the fluid volume approach was created. This thermosyphon's operating fluid is deionized water, as stated. R. Parand, et al. [24] gave a theoretical analysis of the thermosyphon behaviour in the transient domain. To determine the temperature of the thermosyphon and the time required to attain steady state condition, a computer simulation programme based on the lumped technique has been created. Jiao Yonggang, et al. [25] presented a mathematical model to predict the unsteady state start-up process of heat pipe. The temperature, velocity and pressure distribution in the working trap was solved by FLUENT. According to the preceding review, several experimental and theoretical studies look into the effect of working fluid, while others look into the transient reaction. The goal of this research is to predict the transient response of various operating working fluids on the capability of (TPCT).

2 THEORETICAL MODEL

The two-phase closed thermosyphon (TPCT) is constructed up of an evacuated sealed tube that contains a small volume of liquid and is separated axially into three basic regions: evaporator (heating), adiabatic (thermally insulated), and condenser (cooling), as shown in Fig. 1. The heat from the evaporator component is transferred to the liquid in the thermosyphon through the pipe wall, causing it to boil. In the form of latent heat of vaporization, the heat is absorbed by the working fluid. The vapour moves upward because the evaporator zone has a higher pressure than the condenser zone. In the cooler condenser zone, the latent heat absorbed in the evaporator condenses and is released. The heat then exits the thermosyphon through the tube wall and into the outside, passing through the thin liquid coating. The flow circuit inside the tube is completed when the liquid is gravity-forced back to the evaporator section as a thin layer. The theoretical behaviour of a two-phase closed thermosyphon (TPCT) in the transient phase is investigated. The transient model was used to mimic the thermosyphon's response to various working fluids.

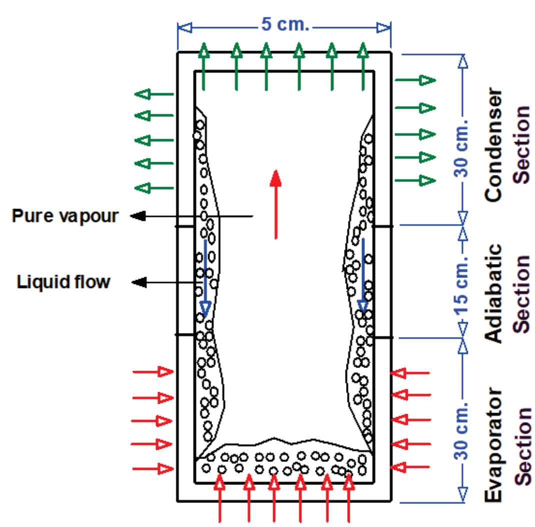

Figure 1 Two-phase closed conventional thermosyphon operation

2.1 Model Description and Assumptions

2.1.1 Thermosyphon Evaporating Section (Boiling)

A closed container makes up a thermosyphon, which collects energy along its evaporating part and extracts it along its condensation part. Look at a diameter D thin-mechanically polished wall material of thermosyphon. When heat is applied to the thermosyphon, saturated fluid is boiled at a specific internal pressure of evaporation part of length L_e. The vapour rise to condenser and condenses and falls by gravity down the wall of the condensation part of length L_c back into the evaporation part. When evaporation occurs at the surface that represents a liquid solid interface, it is termed boiling and the process occurs when the surface temperature in the boiling section T_{sb} exceeds the saturation temperature T_{sat} at adjusted pressure inside evaporation section. Heat energy is transferred from the outer surface to inside surface to the liquid by conduction and convection, which Newton's law of cooling [26] can be applied:

$$Q_{sb} = \overline{h}_{sb} A_{sb} \left(T_{sb} - T_{sat}\right) = \overline{h}_{sb} A_{sb} \Delta T_e \quad (1)$$

We can see vapour bubbles in this process, which propagates and subsequently separates from the surface. The dynamics and formation of vapour bubbles are intricately dependent on the excess temperature, the kind of surface, and the fluid's thermophysical characteristics like surface tension. The dynamics of vapour bubble production also affects the mobility of liquids close to the surface, which has a significant impact on the coefficient of heat transfer. The liquid is free of motion, and its movement at the surface is caused by free convection and mixing triggered by bubble development and detachment. Pool boiling can happen under a variety of circumstances.

There is nucleate boiling in the temperature range of 30 °C > T_e > 5 °C. Two distinct flow regimes may be observed in this range. At nucleation sites in the area, individual bubbles develop and detach from the surface. This separation causes a significant amount of fluid mixing close to the surface, significantly raising the heat transfer coefficients h and q_{sb}. In this regime, the majority of heat exchange occurs directly from the surface to a liquid that is moving at the surface rather than through the surface-born vapour bubbles.

Eq. (2) is developed by Rohsenow [27] and nucleate boiling uses this correlation widely. All liquid properties are evaluated at saturation temperature.

$$q_{sb} = \mu_l h_{fg} \left[\frac{g\left(\rho_l - \rho_v\right)}{\sigma_l}\right]^{0.5} \left(\frac{c_{p,l} \Delta T_e}{C_{s,f} h_{fg} Pr_l^n}\right)^3 \quad (2)$$

Critical heat flux represents an important characterization of boiling curve. It is necessary to operate evaporation process near to this point, but we as well consider the danger of dissipating heat in excess of this amount. Zuber [28] developed and, obtained an expression, which can be approximated as:

$$q_{max} = C_{s,f} h_{fh} \rho_v \left[\frac{\sigma_l g\left(\rho_l - \rho_v\right)}{\rho_v^2}\right]^{0.25} \quad (3)$$

At excess temperatures, a vapour film blankets and covers the surface until separation between the surface and liquid. Because conditions of laminar film condensation, it is common to consider film, boiling correlations on results obtained from condensation phenomena. One such result, which applies to film boiling on a cylinder diameter D, is of the form [26].

$$\overline{Nu}_{sb} = \frac{\overline{h}_{sb} D}{k_v} = C_{s,f} \left[\frac{g(\rho_l - \rho_v) h_{fg} D^3}{\nu_v k_v (T_{sb} - T_{sat})}\right]^{0.25} \quad (4)$$

Where $\Delta T_e = T_{sb} - T_{sat}$.

The transient analysis takes place according to energy balance between boiling heat flux to the surface of boiling section and heat transferred to interior fluid by convection and conduction. This transient transfer occurs according to Eq. (5) [26].

$$(\rho V C_p)_{sb} \left(\frac{dT_{sb}}{dt}\right) = C_{s,f} h_{fg} A_{sb} \left[\frac{\sigma_l g(\rho_l - \rho_v)}{\rho_v^2}\right]^{0.25} - \\ - \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma_l}\right]^{0.5} \left[\frac{c_{p,l}(\Delta T_e)}{C_{s,f} h_{fg} Pr_l^n}\right]^3 \quad (5)$$

The 60 % of maximum heat flux is our assumption [26] to be absorbed by heating section and it is assumed sufficient to boil the fluid at assigned vacuum pressures.

2.1.2 Condensing Section of Thermosyphon

Condensation happens when the temperature of a vapour is decreased below its saturation temperature. In industrial equipment like thermosyphon, the process results from direct contact between a cool surface and vapour. Its latent energy is released, heat energy is transferred to the surface, and the formation of condensate transforms the vapour into liquid phase.

The total heat transfer to the surface may be obtained by using Eq. (6) with and expressed as:

$$Q_{sc} = \overline{h}_{sc} A_{sc} (T_{sat} - T_{sc}) \quad (6)$$

The average Nusselt number in condensing section is:

$$\overline{Nu}_{sc} = \frac{\overline{h}_{sc} L}{k_l} = 0.943 \left[\frac{\rho_l g(\rho_l - \rho_v) h_{fg} L^3}{\mu_l k_l (T_{sat} - T_{sc})}\right]^{0.25} \quad (7)$$

When the flow in tube is in the laminar, wavy regime, Kutateladze [29] advises a correlation:

$$\overline{Nu}_{\mathrm{sc}}=\frac{\overline{h}_{\mathrm{sc}}\left(\frac{\nu_1^2}{g}\right)^{1/3}}{k_1}=\frac{Re_\delta}{1.08\,Re_\delta^{1.22}-5.2} \tag{8}$$

Where,

$$Re_\delta=\frac{4q_{\mathrm{sc}}}{h'_{\mathrm{fg}}\mu_1\pi D} \tag{9}$$

And h'_{fg} is the modified latent heat of vaporization and expressed by:

$$h'_{\mathrm{fg}}=0.68\,h_{\mathrm{fg}}cp_1\,(T_{\mathrm{sat}}-T_{\mathrm{sc}}) \tag{10}$$

And Re is Reynolds number based on liquid film thickness δ and its range for laminar flow is $30\le Re\le 1800$.

The heat removed by condensing section depends on the heat absorbed by boiling section, so the transient equation yields the form [26]:

$$\left(\rho VC_{\mathrm{p}}\right)_{\mathrm{sc}}\left(\frac{\mathrm{d}T_{\mathrm{sc}}}{\mathrm{d}t}\right)=\mu_1 h_{\mathrm{fg}}\left[\frac{g\left(\rho_1-\rho_{\mathrm{v}}\right)}{\sigma_1}\right]^{0.5} * \\ *\left(\frac{C_{\mathrm{p,1}}\Delta T_{\mathrm{e}}}{C_{\mathrm{s,f}}h_{\mathrm{fg}}\,Pr_1^{\mathrm{n}}}\right)^3\pi DL_{\mathrm{sb}}-\overline{h}_{\mathrm{sc}}\pi DL_{\mathrm{sc}}\left(T_{\mathrm{sat}}-T_{\mathrm{sc}}\right) \tag{11}$$

2.2 Solution Methodology

The solution of the previous Eqs. (5), (11) is modelled by finite difference method (Euler) using Engineering Equation Solver (EES) [31] which is used to formulate the model. The solution illustrates the average change in evaporator and condenser wall temperatures with time. The finite difference form of above equations yield to:

For evaporator section:

$$(\rho VC_{\mathrm{p}})_{\mathrm{sb}}\left(\frac{T_{\mathrm{sb}}^{t+1}-T_{\mathrm{sb}}^{t}}{\mathrm{d}t}\right)=C_{\mathrm{s,f}}h_{\mathrm{fg}}A_{\mathrm{sb}}\left[\frac{\sigma_1 g(\rho_1-\rho_{\mathrm{v}})}{\rho_{\mathrm{v}}^2}\right]^{0.25}- \\ -\mu_1 h_{\mathrm{fg}}\left[\frac{g(\rho_1-\rho_{\mathrm{v}})}{\sigma_1}\right]^{0.5}\left[\frac{c_{\mathrm{p,1}}(T_{\mathrm{sb}}^{t+1}+T_{\mathrm{sb}}^{t}/2)}{C_{\mathrm{s,f}}h_{\mathrm{fg}}\,Pr_1^{n}}\right]^3 \tag{12}$$

For Condenser section:

$$(\rho VC_{\mathrm{p}})_{\mathrm{sc}}\left(\frac{T_{\mathrm{sc}}^{t+1}-T_{\mathrm{sc}}^{t}}{\mathrm{d}t}\right)= \\ =\mu_1 h_{\mathrm{fg}}\left[\frac{g\,(\rho_1-\rho_{\mathrm{v}})}{\sigma_1}\right]^{0.5}\left(\frac{c_{\mathrm{p,1}}\,\Delta T_{\mathrm{e}}}{C_{\mathrm{s,f}}h_{\mathrm{fg}}Pr_1^{n}}\right)^3\pi DL_{\mathrm{sb}}- \\ -\overline{h}_{\mathrm{sc}}\pi DL_{\mathrm{sc}}\left[T_{\mathrm{sat}}-(T_{\mathrm{sc}}^{t+1}+T_{\mathrm{sc}}^{t}/2)\right] \tag{13}$$

Initial and boundary conditions: Temperature of the evaporator surface is assumed initially (Time=0) at the same temperature of the surrounding reactor pool temperature, while the condenser wall temperature is assumed as the surrounding air temperature in the reactor hall. The limit values of temperatures are the saturation temperature values corresponding to each fluid distinct vapour pressure which are selected at 40 °C. The TCPT is under different vacuum pressure related to each fluid type and that realizes 40 °C saturation temperature. This value keeps the pool temperature at 40 °C and enables TCPT to absorb accumulated residual heat. Tab. 1 illustrates these pressures.

Table 1 Vapour pressure for various types of fluids

Fluid type	Vapour pressure (Pa)	Saturation temperature (°C)
Water	7375	40
Acetone	56520	
Toluene	7903	
Heptane	12325	

2.3 Model Validation

The model validation is illustrated in Fig. (2), and compared with experimental results by Mohamed S. El-genk [30] while employing heat pipe to absorbing heat load of 600 watt via evaporating section and dissipating it by natural convection through condensing section to air. The pressure inside horizontal thermosyphon is kept at atmospheric pressure 1 bar and the material is copper. The model prediction shows reasonable results for evaporator and condenser wall surface temperatures with the measured values. The figure illustrates that; the difference between experimental results and model predictions do not exceed 5%.

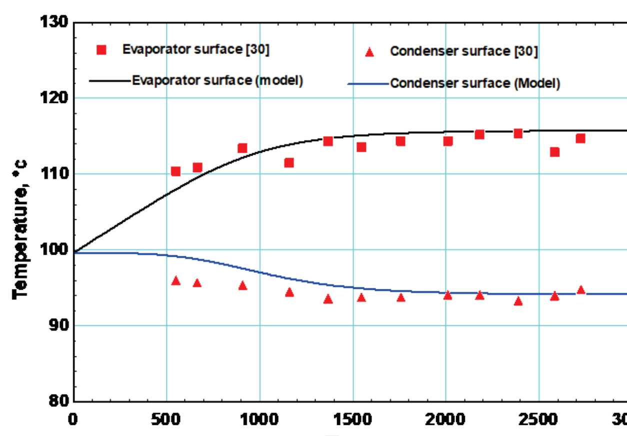

Figure 2 Evaporator and condenser surface temperatures at 600 Watt

3 RESULTS AND DISCUSSION

The theoretical model is design to study the transient behaviour of thermosyphon and the effect of the various working fluids on the thermosyphon feature. Saturation temperatures are adjusted to be the same for four fluids type by varying vacuum pressure inside heat pipe to satisfy the same saturation temperature. The used thermosyphon is characterized by this information below:

Saturation temperature: 40 °C
Dimension:
Evaporator length: $30 + \pi d$ cm
Condenser length: $30 + \pi d$ cm
Adiabatic length: 15 cm
Diameter (d): 5.0 cm
Thickness: 3.0 mm
Material: Aluminium 6061
Nucleate boiling flux: 10 % of critical heat flux $C_{s,f} = 0.013$, and $n = 1.0$.

The results of the analytical modelling are primarily addressed and examined in order to evaluate the performance of a closed two-phase thermosyphon with different working fluids in transient operation.

3.1 Critical Heat Flux

The critical heat flux of considered fluids is shown in Fig. (3), which is calculated from Eq. (3) at saturation temperatures 40 °C and dependent on thermodynamic properties of each fluid. As shown in the figure; for the same condition of operation; the water has the greatest critical heat flux followed by Acetone, while Heptane and Toluene are the lowest and are relatively the same values. This behaviour is returned to the latent heat of the fluids, where it is the major parameter appears on the values.

It is assumed in model that the nucleate boiling heat flux is to be maintained at 10 % of the critical heat flux. Consequently this flux will be greater in Water and Acetone but will be lower in case of Heptane and Toluene which are the same.

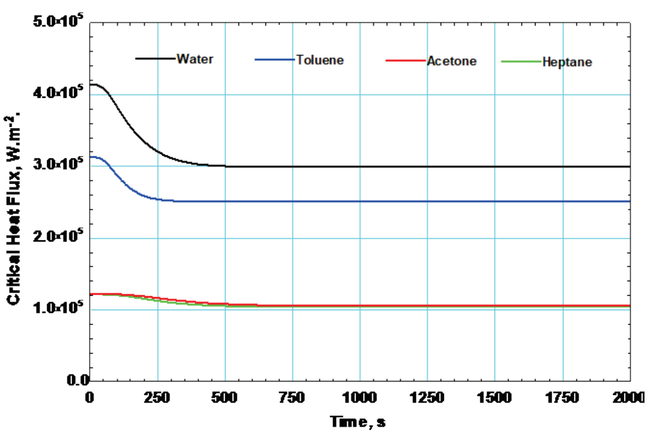

Figure 3 Critical heat flux of various fluids

3.2 Transient Evaporator and Condenser Temperature

The Fig. 4 illustrates the temperatures predicted by the mathematical model for evaporator wall surface versus time using various working fluids. It is shown from the figure that, at the beginning of calculation (when the wall surface temperature is reached approximately to the saturation temperature of the fluid about 40 °C) the evaporator surface wall temperature increases rapidly. This increasing in the temperature is due to the increase of heat flow from available temperature difference. However, the rise in temperature decreases until steady state condition is realized. This is an obvious fact due to the decrease in the driving force, namely the temperature difference. It is concluded from the Fig. 5 that, the wall temperature in the evaporator in case of water realizes higher temperature at steady state condition than the other fluids, this behaviour referred to the highest latent heat of vaporization of water and it means that water absorbs more heat and has more cooling capacity.

Also shown from the Fig. 5, the evaporator surface temperature achieves 15 °C higher than the saturation temperature for water while Heptane record only 7 °C rise due to the higher thermal conductivity of water than Heptane. The transient temperatures calculated by the mathematical model for condenser wall surface is illustrated in Fig. 5 for various working operating fluids. The figure demonstrates that, the condenser wall surface temperature decreases below the saturation temperature due to condenser cooling heat dissipation. With time, the rate of temperature decrease, however, slows until a steady state condition is established. This is a clear reality because the driving force, notably the temperature difference between wall and cooling air, has decreased. The condenser wall temperature in the Heptane reaches the steady state condition faster and has a lower temperature compared to the other fluids.

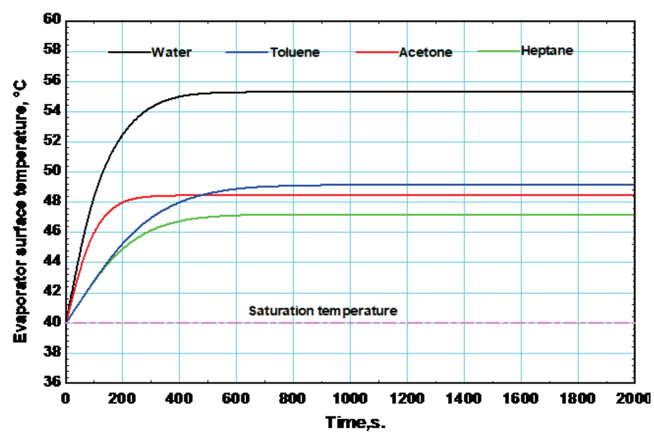

Figure 4 Transient Evaporator Surface Temperature for different working fluids

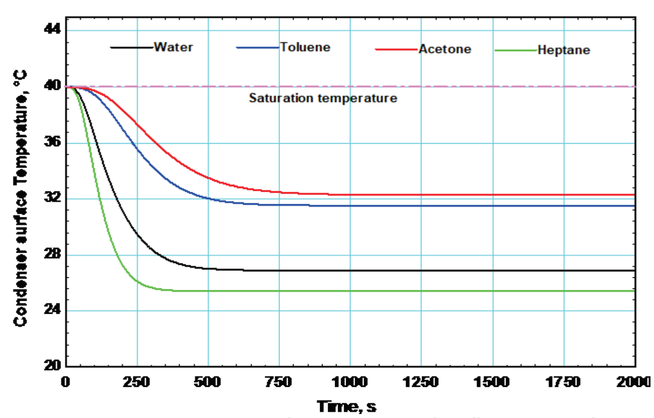

Figure 5 transient condenser surface temperatures for different working fluids

3.3 Evaporator and Condenser Heat Transfer Coefficient of Thermosyphon

The average heat transfer coefficients (h_e) and (h_c) in the evaporator and condenser are used to determine the degree of heat transfers within the thermosyphon. The average heat transfer coefficients are calculated using a mathematical model for both boiling, condensation and steady-state situations.

The transient heat transfer coefficient of the evaporator for the various thermosyphon working fluids is plotted in Fig. 6. At steady state, acetone has a higher heat transfer coefficient relative to other fluids and Toluene has a lowest heat transfer coefficient. As indicated in the figure, the evaporator heat transfer coefficient of Acetone has the greatest value of 2965 W/m^2·K, while the heat transfer coefficients for other operating fluids range from 1950, 1449 and 1152 W/m^2·K for Water, Heptane and Toluene respectively.

Coefficients are calculated using Newton's law of cooling. The values of condenser heat transfer coefficient are predicted from Eq. (8) based on some physical properties like kinematic viscosity, Reynolds number and thermal conductivity.

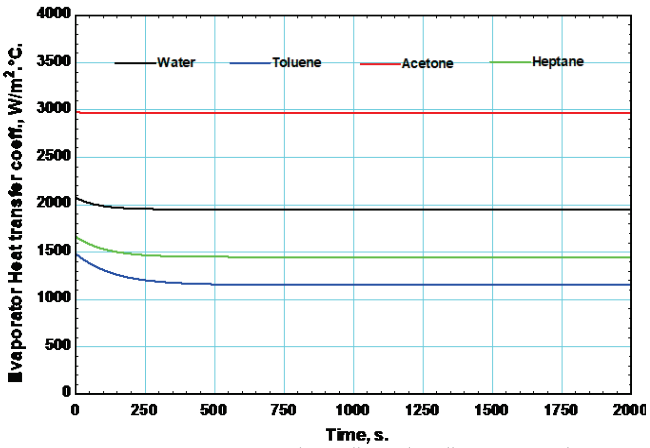

Figure 6 Evaporator heat transfer coefficient for different working fluid

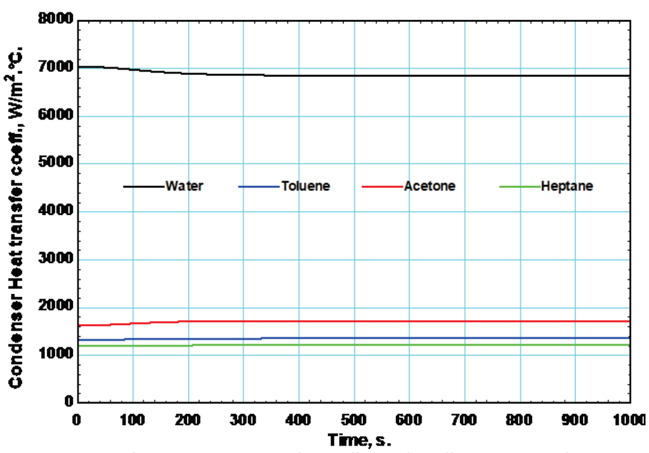

Figure 7 Condenser heat transfer coefficient for different working fluid

The transient heat transfer coefficient of the condenser in TPCT is shown in Fig. 8 for the various operating working fluids. As indicated in the diagram, the condenser heat transfer coefficient of Water has the greatest value then Toluene, Acetone and heptane respectively. The heat transfer coefficient of water is more than three times that of other working fluids.

3.4 Thermosyphon Resistance

Thermal resistance is an important parameter to the thermosyphon performance. The thermal resistance could be calculated using the below formula: $R = \dfrac{T_{sb} - T_{sc}}{Q}$, where Q is the heat absorbed by the heat pipe.

Fig. 8 shows the transient resistance of TPCT for the various operating fluids. As illustrate in the figure, the Toluene has the highest resistance than other fluids and the Acetone has the lowest. While the Acetone is the lowest resistance, the water is more used due to that, it is more available with no cost.

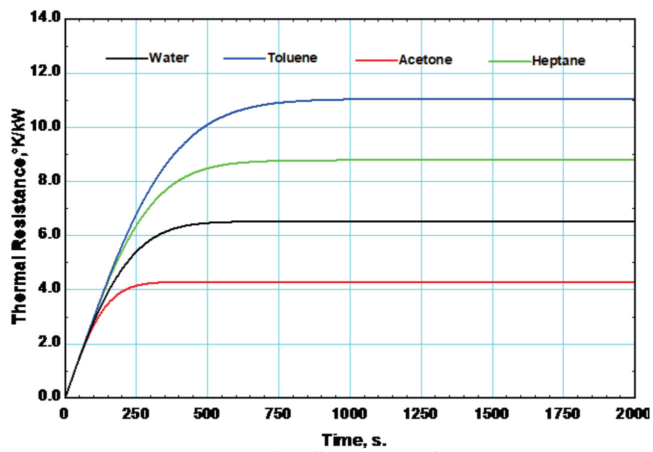

Figure 8 Heat pipe resistances for different working fluid at constant heat load

4 CONCLUSION

A mathematical model is developed to describe and study transient thermal behaviour of the two phase closed thermosyphon (TPCT) with different working fluids. The best fluid performance with TPCT is the main objective of the study. Evaporator and condenser surface temperature are the most important key results. Water, Toluene, Acetone and Heptane are used as working fluids in the present study. A validation of the mathematical model is presented using water as a working fluid by comparing the surface temperature of evaporator and condenser which are predicted from the model with experimental results [30]. The performance of TPCT with different fluids is presented. According to the results, Acetone has the highest heat transmission coefficient in the evaporator section. Its heat transfer coefficient is twice of Heptane and is slightly closer to water. Water has the highest critical heat flux, highest evaporator surface temperature and has relatively high evaporator heat transfer coefficient which means that water absorbs more heat from surrounding heat load. In condenser, water has the maximum heat transfer coefficient, which is three times more than the other fluids. Water has lower heat resistance than heptane and Toluene and little higher than Acetone. Water is preferable working fluid in TPCT because it transfers more heat flux in the evaporator, dissipates more heat in condenser, has low thermal resistance and it is available with no cost.

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Nomenclature:

A Area, m^2
C_p Specific heat, J/kg·°C
D Diameter, m
L Length, 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
q Heat flux, W/m^2
Q Heat load, W
t Time, s
dt Time difference, s
T Temperature, °C
dT Temperature difference, °C
C Nucleate boiling coefficient
n Nucleate boiling exponent

Greek Symbols
μ Dynamic viscosity, $N{\cdot}s/m^2$
β Thermal expansion coefficient, K^{-1}
v kinematic viscosity, m^2/s
ρ Density, kg/m^3
σ Surface tension, N/m
Δ Film thickness, m

Dimensionless Groups
Pr Prandtl number,
Re Reynolds number,
Nu Nusselt Number
Subscripts
a Adiabatic,
b Boiling
c Condenser,
e Evaporator
f Fluid,
l Liquid
s Surface
sat Saturation
v Vapour
sb Surface of boiling
sc Surface of condensation
s,f Surface-fluid combination