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Transient Analysis of Two-Phase Closed Thermosyphon (TPCT) Using AL_2O_3 Nanofluid

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Abstract: A theoretical model is developed to predict the dynamic behavior of TPCT in transient conditions. Al_2O_3/water nano-fluid is used as a working fluid with three different concentrations (1, 2 and 4%) by volume. The various transient parameters during operation from start up to shutdown condition are calculated to present the performance of the TPCT with nano-fluids. The model evaluated the average wall and fluid temperature distribution with time. The average evaporator heat transfer coefficient, overall heat transfer coefficient, effective thermal conductivity and thermal resistance of TPCT are presented. Also the effects of changing the nano-fluid concentration on all these parameters are studied. According to the model results, the performance of the TPCT increases compared with pure water. The different parameters increase about 10% with increasing nano-fluid concentration.

Index Terms— Nano-fluids , Heat transfer enhancement, Thermosyphon.

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1. INTRODUCTION

The two-phase closed thermosyphon (TPCT), which is essentially a gravity-assisted wickless heat pipe, utilizes the evaporation and condensation of the working fluid inside the heat pipes to transport heat. A TPCT uses gravity to return the condensate to evaporator. Its position is not restricted and it may be used in any orientation [1]. The TPCT has a simple structure, smaller thermal resistance, higher efficiency and lower fabrication costs. Given these advantages, the TPCT has been widely used in many fields, such as industrial heat recovery, electronic component and turbine blade cooling, solar heating systems and so on [2].

Many of these applications would benefit from a decrease in the thermal resistance of the heat-transfer fluid. Most commonly used working fluids in TPCTs are water; methanol; ethylene glycol (EG) and their mixtures which are originally poor heat transferring fluids. Since thermal conductivity of these fluids plays an important role in these energy efficient heat transfer equipments, numerous techniques have been introduced to improve it. Because of higher thermal conductivity of solids compared to those of liquids, Consequently fluids with nano-sized particles suspended in them which later called nano-fluids has been proposed by [3]. From the Argonne National laboratory, USA. By suspending nano-sized particles in a fluid, its heat transfer performance can be significantly improved with incurring either little or no penalty in pressure drop.

Since, a lot of researches have been carried out studies on the heat transfer characteristics of nano-fluids. The heat transfer characteristics of nano-fluids started with the investigation of thermal conductivity [4-5], then the single-phase flow heat transfer [6-8]. The focus mainly is on the phase-changing heat transfer of nano-fluids. Among the phase-changing heat transfer, the application of nano-fluids in heat pipes gains increasing popularity.

The involved heat pipes include the grooved heat pipe [9-10], wicked heat pipe [11-12], oscillated heat pipe [13-14], and the thermosyphon [15-20].

Xue et al. [15] studied the heat transfer performance of carbon nano-tube-water nano-fluid in a thermosyphon The mass concentration of nano-particles is 1.3158 wt.%. The thermosyphon is a copper tube with an outer diameter (O.D.) of 20 mm. The filling ratio is 20%. Results show that the thermosyphon with carbon nano-tube nano-fluid has a higher evaporation section wall temperature, incipience temperature, and excursion, as well as thermal resistance. The carbon nano-tube-water nano-fluid deteriorates the heat transfer of the thermosyphon compared with the water case.

Khandekar et al. [16-17] investigated the overall thermal resistance of a closed two-phase thermosyphon using water-based Al_2O_3 (40 to 47 nm), CuO (8.6 to 13.5 nm), and laponite clay (disks with a diameter of 25 nm and thickness of 1 nm) nano-fluids. The length and the inner diameter of the thermosyphon are 720 and 16 mm, respectively. The nano-particle mass concentration is 1.0 wt.%. Results show that all nano-fluids have inferior thermal performance compared to pure water. A mechanism analysis guesses that the increase in wettability and entrapment of nano-particles in the grooves of the surface cause a decrease of the Peclet number in the evaporator side and finally leads to poor thermal performance.

Noie et al. [18] studied the Al_2O_3-water nano-fluid in a thermosyphon. The thermosyphon is made of a copper tube with an inner diameter of 20 mm and a length of 1,000 mm. The length of the evaporator and the condenser is 350 and 400 mm, respectively. The nano-particle volume concentration is 1% to 3%. Results show that the nano-fluid can enhance the heat pipe efficiency by 14.7%, and the thermosyphon shows a more uniformly distributed temperature.

Paramatthanuwat et al. [19] studied the heat transfer of Ag-water nano-fluid in a thermosyphon. The effects of filling ratio (30%, 50%, 80%), the operating temperature (40°C, 50°C, 60°C), the ratio of length and diameter (5,10, 20), and the diameter (7.5, 11.1, and 25.4 mm) on the heat transfer performance were investigated in detail. Results show that the heat transfer capacity can be enhanced by 70% by adding Ag nano-particles.

Teng et al. [20] studied the heat transfer performance of the Al_2O_3-water nano-fluid (mass concentrations of 0.5%, 1.0%, and 3.0%). The thermosyphon is made of a copper tube with an inner diameter of 8 mm and a length of 600 mm. The authors investigated the effects of inclination, filling ratio, and mass concentration on the heat transfer performance. The thermosyphon efficiency can be enhanced by 16.8% at the mass concentration of 1.0%.

It is obvious from the previous studies that, most investigations focus on the experimental performance characteristics of TPCT with nano-fluid. Therefore, it was seen appropriate to indulge in a theoretical transient study, which could predict the behavior of DTT from startup, the steady state to shutdown condition. The main objective of the current study is to develop a theoretical model that can predict the dynamic behavior of the TPCT by tracing various transient parameters during operation from start up to shutdown condition.

2. Theoretical Model

A model has been performed to describes the thermal and phase flow of closed two-phase thermosyphon (TPCT). This model presents a theoretical investigation of thermosyphon behavior in the transient regime. The transient model was adopted to simulate the response of thermosyphon with pure water and nano-fluid with different concentrations. The transient thermal behavior of (TPCT) has been utilized to obtain a mathematical expression of the system response. Figure (1) show TPCT, which is basically divided axially into three basic regions: evaporator (heating), adiabatic (thermally insulated) and condenser (cooling) sections. The thermosyphon main tube made of copper with 26 mm inner diameter, 1 mm thickness and 970 mm long. The evaporator and condenser section lengths are 550 mm and 250 mm respectively, while the adiabatic section is 170 mm long. Individually, each region is mathematically and thermally treated due to variation of the heat transfer processes. In addition, the evaporator region is filled of a liquid (filling ratio=1). When the power is on, the heat generated in the heater is rising its temperature with time. The heat transferred to the wall causing its temperature to rise and with time transfer the heat to the liquid. On reaching the saturation temperature, any heat added causes the saturated fluid to evaporator. Then the heat rate is carried by the vapor flows from evaporator to condenser, which is rejected to the heat sink. The model is analyzed in one-dimension, where the axial coordinate x is mainly measured from the evaporator bottom.

2.1. Equations of the Model

A model describing both thermal and phase flows of the TPCT has been performed by Farsi,etal [21]. It is based on a spatial discretization similar to that proposed by Reed and Tien [22] and Dobran [23]. A thermal model has been derived from the previous model. This simple model has been developed in order to provide analytical expressions of the variables in the system. On the other hand, to give an expression of the TPCT response time as a function of the various parameters. The heat balance equation for each wall and fluid give:

$$C_w \frac{d}{dt}T_w = Q_e - h_e S_e (T_w - T_f) \tag{1}$$

$$C_f \frac{d}{dt}T_f = h_e S_e (T_w - T_f) - h_c S_c (T_f - T_{wat}) \tag{2}$$

The solution of the above equations is obtained by using finite difference -Euler method in step of Δ t. The solution leads to the following expressions of the average evaporator wall temperature and the average fluid temperature respectively:

$$C_w\,(T_w^{n+1} - T_w^{n}) = \Delta t * Q_e - \Delta t * h_e S_e\,(T_w^{n+1} - T_f^{n+1}) \quad (3)$$

$$C_f\,(T_f^{n+1} - T_f^{n}) = \Delta t * h_e S_e\,(T_w^{n+1} - T_f^{n+1}) - \Delta t * h_c S_c\,(T_f^{n+1} - T_{wat}) \quad (4)$$

$$T_w^{n+1} = \frac{C - B * T_f^{n+1}}{A} \quad (5)$$

$$T_f^{n+1} = \frac{F - \dfrac{D * C}{A}}{E - \dfrac{D * B}{A}} \quad (6)$$

Where these constant are:

$$A = C_w + \Delta t * h_e * S_e$$
$$B = -\Delta t * h_e * S_e$$
$$C = \Delta t * Q_e + C_w * T_w^n$$
$$D = -h_e * S_e * \Delta t$$
$$E = C_f + \Delta t * h_e * S_e + \Delta t * h_c * S_c$$
$$F = C_f * T_f^n + \Delta t * h_c * S_c * T_{wat}$$

Equation (5) and (6) depend mainly on the nine variables Q, Cw, Cf, Se, Sc, he, hc, Tw and Δt.

2.1.1. Heat Load, Q

The net hear load is accurately calculated by the rate of the heat removal in the cooling jacket of the condenser as a following:

$$Q = [m^* Cp^*(T_o - T_i)]_{cw} \quad (7)$$

2.1.2. Average Evaporator Heat transfer Coefficient, he

The process of the heat transfer in the liquid pool of the thermosyphon generally assumed as a common nucleate boiling. The nucleate boiling heat transfer coefficient calculated from Forester-Zuber equation [24]:

$$h_e = \frac{0.00122 * \Delta T_{sat}^{0.24} * \Delta P_{sat}^{0.75} * c_{pl}^{0.45} * \rho_l^{0.49} * k_l^{0.79}}{\sigma^{0.5} * H_{fg}^{0.24} * \mu_l^{0.29} * \rho_g^{0.24}} \quad (8)$$

2.1.3. Average Condenser Heat Transfer Coefficient, hc

The average heat transfer coefficient between the condenser surface and cooling water can be calculated in the case of laminar and turbulent flow regimes [25] respectively as:

For laminar

$$h_{cw} = 1.86(\mathrm{Re.\ Pr})_{cw}^{0.33}(d_h/L_c)^{0.33}(\mu_{cw}/\mu)^{0.14}(k_{cw}/d_h) \quad (9)$$

For turbulent

$$h_{cw} = 0.023(\mathrm{Re}_{cw})^{0.8}\ (\mathrm{Pr}_{cw})^{0.33}\ (k_{cw}/d_h) \quad (10)$$

Where, the flow is assumed as turbulent at Recw ≥ 3000,

$$\mathrm{Re}_{cw} = \frac{4\dot{m}_{cw}}{\pi d_h \mu_{cw}}$$

2.2. Equivalent Overall Heat Transfer Coefficient of Thermosyphon, U

The performance of the thermosyphon can be hypothetically expressed by an overall heat transfer coefficient. The equivalent overall heat transfer coefficient is calculated from the following equation:

$$U = q_{ax} / \Delta T_i \quad (11)$$

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

$$\Delta T_i = Tw_e - Tw_c \quad (12)$$

2.3. Effective Thermal Conductivity (Keff)

Effective thermal conductivity is an appropriate means for measuring the performance of two-phase closed thermosyphon. TPCT can achieve high thermal conductivity due to liquid vapor phase change to transport heat from evaporator to condenser section. The effective thermal conductivity can be evaluated by the following relation as:

$$K_{eff} = q_{ax} * Lp / \Delta Ti \quad (13)$$

Where:

$$Lp = 1/2(L_e + L_c) + L_{ad}$$

2.4. Thermal Resistance of Thermosyphon

The thermal resistance of thermosyphon with pure water and different nano-fluids, where calculated by the following relation:

$$R_{th} = (T_e - T_c) / Q \quad (14)$$

Where, Te and Tc are average values of the evaporator and condenser temperatures.

2.5. Nano-fluid Properties Correlations

The thermosyphon operation is based on vaporization and subsequent condensation of the working fluid. So, the properties of working fluid have effect in the performance of TPCT. Thermophysical properties including the thermal conductivity, the viscosity, and the surface tension of nano-fluid have been introduced by S.Lee, et al. [26].

The density of nano-fluids is calculated as:

$$\rho_{nf} = \rho_{np} * \phi + \rho_l (1 - \phi) \quad (15)$$

The specific heat of nano-fluids is calculated as:

$$cp_{nf} = cp_{np} * \phi + cp_l (1 - \phi) \quad (16)$$

The thermal conductivity of nano-fluids is evaluated as:

$$k_{nf} = k_l (1 + 7.4 * \phi) \quad (17)$$

The viscosity of nano-fluids is calculated from following correlation:

$$\mu_{nf} = \mu_l (1 + 2.5 * \phi) \quad (18)$$

The latent heat of nano-fluids is the same as that of water.

A computer simulation program based on the method was developed to estimate temperatures and the other parameters of the thermosyphon. The equations are solved by Engineering Equation Solver program (ESS).

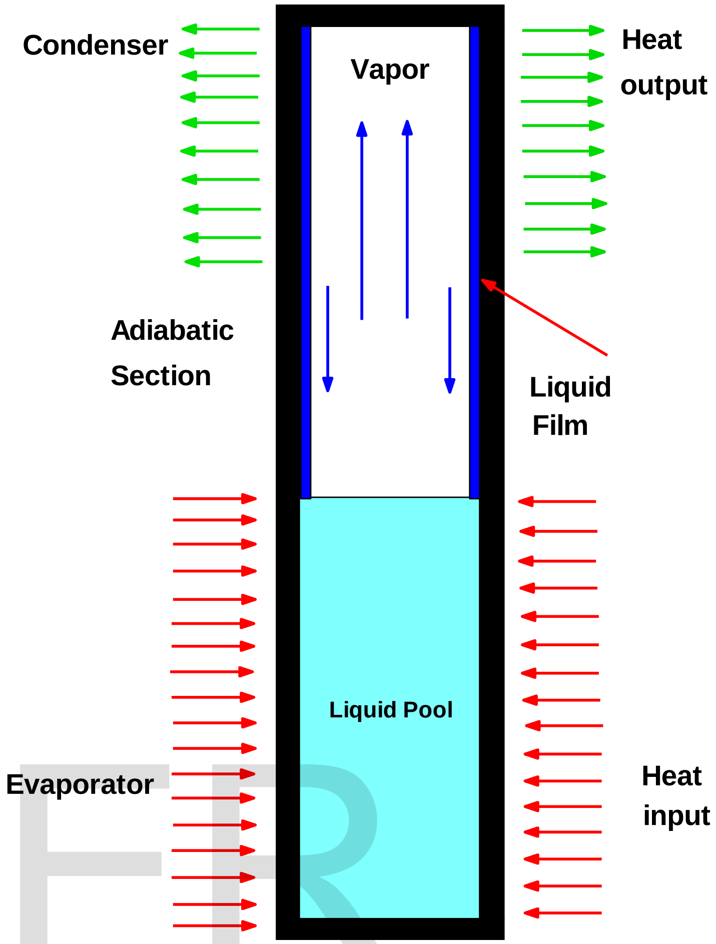

Fig. (1) A Schematic of Two-Phase Closed Conventional Thermosyphon.

3. Discussion of the Analytical Modeling Results

The results describing the time-dependent behavior of the thermosyphon during transient operation of start-up from initial conditions to steady state and shutdown are basically discussed and analyzed. These result analyze the average wall and fluid temperature distribution for the various thermosyphon operation stages. The average evaporator heat transfer coefficient, overall heat transfer coefficient, effective thermal conductivity, thermal resistance of TPCT are presented. Also the effects of changing the nano-fluid concentration on all these parameters were studied.

Figure (2) illustrates the distribution of the transient average wall and fluid temperatures through heat up, steady state and shutdown. At the beginning of the heating process, the heat transfers first to the evaporator wall of the thermosyphon. The wall temperatures increase from the ambient temperature with time and it begins to transfer heat to the fluid. With time the fluid temperature rises along the pool region in the evaporator and starts to evaporated.

The wall and fluid temperature increase to certain value and when all temperatures are constant the TPCT reach to steady state. Following the period of steady state, the shutdown is initiated by the power is off. The wall and fluid temperature decrease during shut down until reach to the ambient temperature.

The effect of several concentration levels were investigated on the thermosyphon wall temperature variation. As shown in the figure, the temperature changes with different nano-particle concentration. By increasing the nano-particle concentration, the temperature gradient in evaporator becomes less. This happens mainly due to the direct relation of particles concentration with the nano-fluid thermal conductivity. As the particle concentration raises (i.e. effective thermal conductivity of the working fluid increases) the temperature amplitude drops in the evaporator section. This happens owing to the higher heat flux passes through TPCT as the fluid thermal conductivity increases.

The calculated average evaporator heat transfer coefficient is plotted in figure (3). With the time for the different transient operations. Throughout the heat-up phase, he increases gradually from the bottom to the top of the evaporator. The wall/working fluid temperature increase gradually forming the bubble nuclei. While the fluid temperature increases with time the bubbles has a higher intensity near the free surface of the liquid pool. The increase of the bubble movement leads to increase of the evaporator heat transfer coefficient. At steady state, all temperatures are constant that the evaporator heat transfer coefficient of the TPCT constant with time. When the shutdown is initiated, wall and fluid temperature decrease and according to that the evaporator heat transfer coefficient decreases with time.

The figure also represents the effect of concentration levels on average evaporator HTC. During nucleate pool boiling, the single-phase convective transport is patently affected by many complementary effects such as bubble departure diameter and frequency, nucleation site density and the average rise velocity of individual bubbles. Generally, the bubble formation is related to the surface wettability and roughness. It is cleared from previous study about nano-fluid boiling that the nano-particles changed the surface condition of the evaporator section. The sizes of nano-particles are smaller than the cavities of the clean surface. Then, the Al_2O_3 nano-particles that may deposit on nucleation sites could create more new active nucleation sites by splitting a single nucleation site into multiple ones and enhanced the boiling heat transfer. Beside this, the irregular nano-pores formed between deposited Al_2O_3 nano-particles would affect the bubble diameter and release frequency, and then bubbles may be continuously generated. Therefore, the average heat transfer coefficient increases with the nano-fluid concentration. As shown from the figure, the evaporating HTC of nano-fluid increases maximally by 12% at the operating temperature of 65°C. The transient condenser heat transfer coefficient is evaluated at different nano-fluid concentrations. As shown in figure (4), the nano-fluid concentration has little effect on the condenser heat transfer coefficient. This effect is due to that, when the nano-fluid evaporates the nano-particles aren't transported upwards in the condenser zone by the vapor inertia. But the fact that even micron sized pollutant particulates do not settle down for extended periods in the atmosphere strongly indicates that the possibility of nano-particle transport in the condenser section cannot be ruled out. According to the above discussion, nano-

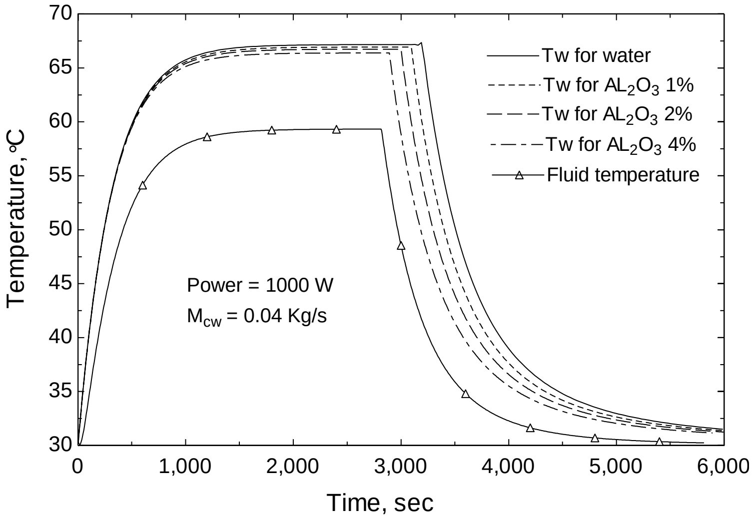

Fig. (2)Transient Average Outer Wall and fluid Temperature at Different Nanofluid Concentration.

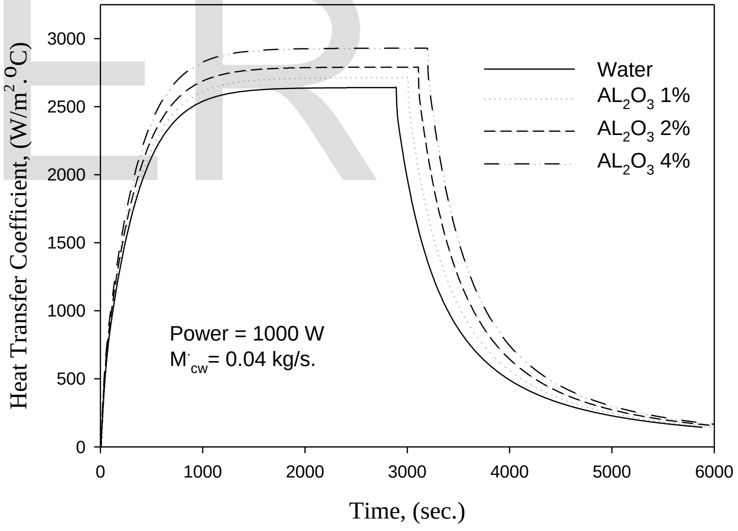

Fig. (3). Transient Average Evaporator Heat Transfer Coefficient at Different Nanofluid Concentration.

fluid can enhance the evaporating HTC of the thermosyphon but has no effect on the condensing HTC. The heat transfer characteristics of nano-fluid result mainly from the changes of the thermo physical properties of nano-fluids. The overall heat transfer coefficient is plotted in figure (5) viruses the time for heat up , steady state and shutdown. As shown in the figure, there is a direct relation between the overall heat transfer coefficient and heat transfer rate. Although, the temperature difference ΔTi increases with the time, 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 in the heat flux is relatively higher than the rate of increase of temperature difference (ΔTi). Consequently, the equivalent overall heat transfer coefficient increases at heat-up region. At steady state, the overall heat transfer coefficient is constant for period of time. when power is off at shutdown, the heat transfer rate from evaporator to condenser decreases and also the overall heat transfer coefficient.

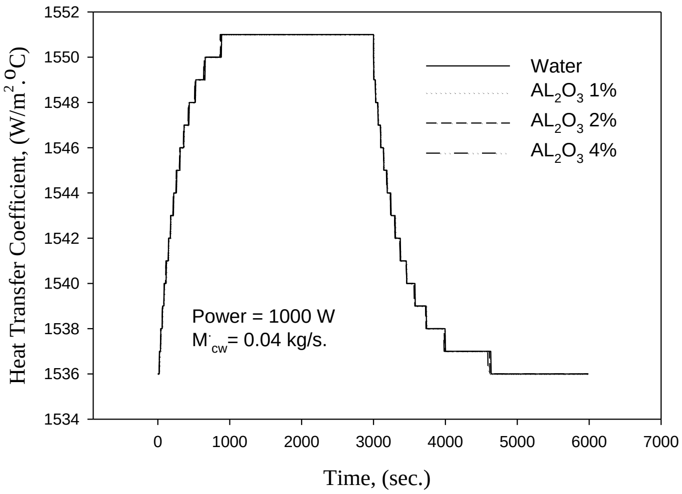

Fig. (4). Transient Average Condenser Heat Transfer Coefficient at Different Nanofluid concentration.

Also the figure illustrates the relation between the overall heat transfer coefficient and the nano-fluid concentration, the increase of nano-fluid concentration from 1 to 4 % increase the overall heat transfer coefficient by nearly 11%.

The effect of nano-particles on two-phase flow heat transfer enhancement may be illustrated through two reasons; the suspended nano-particles increased the thermal conductivity of base fluid and the interactions among the nano-particles itself on one hand and between nano-particles and the inner surface of the thermo syphon. These reasons increase the heat transfer rate in evaporator and from evaporator to condenser. Therefore the increasing of nano-particle concentration causes an increasing in the overall heat transfer coefficient.

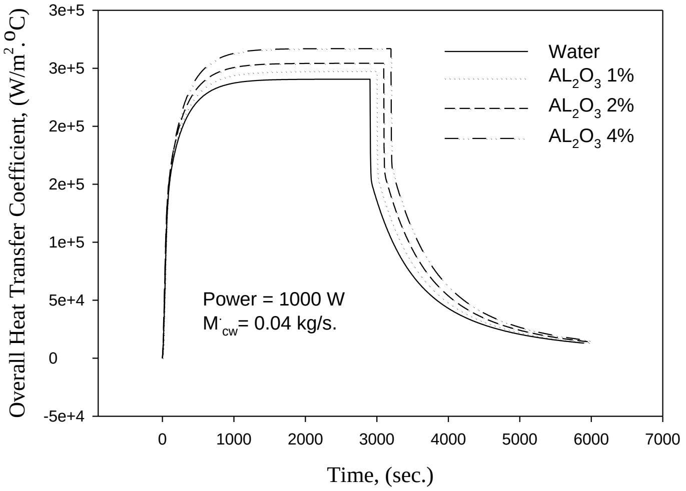

Fig. (5). Transient Overall Heat Transfer Coefficient of the Thermosyphon at Different Nanofluid Concentration.

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. Figure (6) presents the variation of the effective thermal conductivity of the TPCT versus time. As it is observed from the figure, the effective thermal conductivity is extremely dependent on the heat flow rate, where it increases with heat transferred that increase with time. At steady state condition the effective thermal conductivity of thermosyphon is constant with the constant heat load and temperatures. At shut down, the effective thermal conductivity is decreasing due to the power off and therefore, the heat transferred from evaporator to condenser decreases.

The figure also shows the relation between the effective thermal conductivity and the nano-fluid concentrations at constant heat load and constant cooling water flow rate. It was shown that, effective thermal conductivity is increased by 10% during increasing in the nano-fluid concentrations. The increase of the effective thermal conductivity is referred to the increase in the thermal conductivity of the working fluid.

In comparison with the thermal conductivity of copper and silver the thermal conductivity of the thermosyphon (water fluid) is more than 300 folds higher than that for copper and silver. With added the nano-particle to the fluid, the thermal conductivity of the TPCT reaches more than 350 folds higher than that for copper. 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.

The thermal resistances of the TPCT using both the water and nano-fluids are represented in the transient operation in Fig. (7).

Thermal resistance is evaluated at the different stages of the operation of the TPCT from heating up, steady state and shutdown. The results show that, the decrease of the thermal resistance with time and with increasing the nano-fluid concentration. To interpret the thermal resistance decreasing with nano-fluid concentration increasing, it may be said that the total thermal resistance of a TPCT between evaporator and condenser section consisted of thermal resistance in the thermosyphon wall, the thermal resistance due to evaporation and condensation (evaporator and condenser sections) and the thermal resistance in the two-phase flow through heat TPCT length. The wall thermal resistance is independent of the working fluid. Thermal resistances at the evaporator and condenser sections were influenced by several parameters, such as surface condition of thermosyphon inner wall. Considering previous discussion about nano-fluid and boiling heat transfer coefficient, one can be found that thermal resistance at Evaporator section decreased because of increasing liquid thermal conductivity, density, active nucleation site density, bubble release diameter and frequency. on the other hand, also, the diffusion and collision intensification of nano-particles in nano-fluid near duct wall due to increase in concentration of nano-particles leads to rapid heat transfer from wall to nano-fluid. The increase in heat transfer with the nano-fluid concentration leads to the decrease in the thermal resistance of thermosyphon.

4. Conclusion

The transient thermal performance of TPCT using Al_2O_3/water nano-fluid as the working fluid under different concentrations was theoretically studied. Different performance parameters are calculated to evaluate the advantage of using Nano-fluids. The results indicated that, temperature distributions on the TPCT were lower level using nano-fluid compared to pure water. Heat transfer coefficients and thermal conductivity are enhancement about 12% with increasing nano-fluids concentration. The thermal resistance of the TPCT was less when nano-fluids concentration increases. The higher thermal performance TPCTs loaded with nano-fluid proved its potential as substitute for conventional ones with pure water. This finding makes nano-fluid attractive as working fluid in heat pipe and thermosyphon technology.

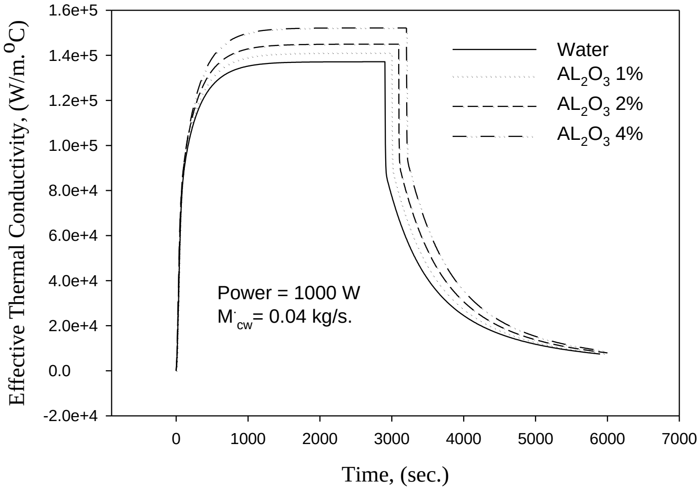

Fig. (6). Transient Effective Thermal Conductivity of the Thermosyphon at Different Nanofluid Concentration.

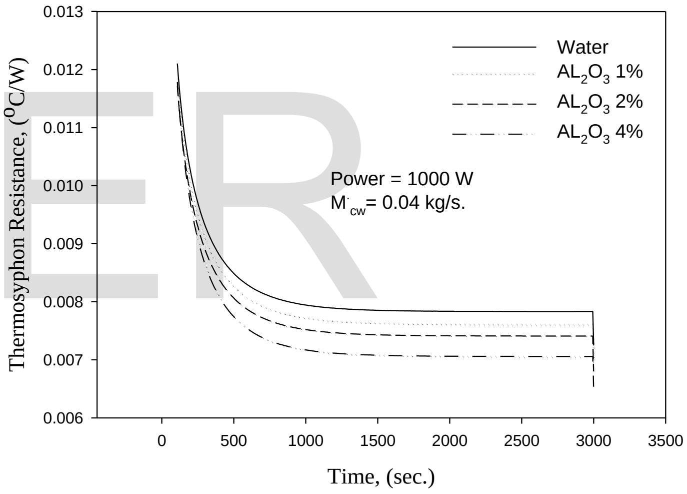

Fig. (7). Transient Thermosyphon Thermal Resistance of at Different Nanofluid Concentration.

5. Nomenclature

A	Cross section 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/s2),
h	heat transfer coefficient (W/m2.°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),
$\dot{m}$	water flow rate (kg/s),
P	pressure (N/ m^2),
q	heat flux (W/m^2),
Q	heat load (W),
S	radial surface area,(m^2)
t	time
T	temperature (°C),
ΔT	temperature difference (°C),
U_{eq}	equivalent overall heat transfer coefficient (W/ m^2 °C),
x	Axial distance (m).

Greek Symbols

φ	nano-fluid concentration,
μ	dynamic viscosity (N.s/m^2)
ρ	density(kg/m^3),
σ	surface tension (N/m),

Dimensionless Groups

Pr	Prandtl number (C_p μ/k),
Re	Reynolds number (4q L/h_{fg} μ).

Subscripts

a	adiabatic,
am	ambient,
ax	axial,
c	condenser,
cw	cooling water
e	evaporator,
eq	equivalent,
I,i	inner, initial,
j	jacket,
l	liquid(water)
o	outer,
m	mean,
nf	nano-fluid,
np	nano particle
p	pool,
r	radial,
sat	saturation,
tp	two-phase
v	vapor,
w	wall
wat	water
β	thermal expansion coefficient(K^{-1})

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