Original Article

Modeling of the Thermal Behaviour of Reverse Thermosyphon, utilizing Solar Energy and different working fluids

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Abstract

In solar collectors, the most important challenge is the lower performance in off-peak hours of sunlight. Thermal performance of the bubble pump enabled reverse thermosyphon integrated with a U-tube solar collector is evaluated numerically concerning average temperature along the length of the thermosyphon, thermal resistance, and efficiency. The proposed system is equipped with different nanoparticles and phase change materials at specific concentrations as two working media by passive downward heat transfer. The influences of actual variable heat flux, different fill ratios, and flow rate on thermal processes are analysed along with condensation and evaporation processes of phase change material. The results showed a significant enhancement in thermal efficiency of 71% and high operating temperature up to $98°C$. The investigated parameters were found to have a high impact on thermal performance. The best phase

change material and the nanoparticle at lower and higher heat flux and the best fill ratio for various heat flux are adjudged. Temperature distribution profiles, heat transfer, and thermal performances along with the multi-phase flow visualization of the reverse thermosyphon by CFD simulations are summarised. Quantitative estimation of the performance analysis under the high sink as well as the anti-gravity operating attributes of critical nature is highlighted.

Keywords: Thermal efficiency, CFD, anti-gravity, nanofluid, boiling, condensation, evaporation.

1. Introduction

Thermal energy recovery and its utilization has improved much by the wide use of two-phase closed thermosyphon systems. Modifications in such devices and consequent temperature profile developments by the geyser and nucleate pool boiling lead to dry-out limit, and expansion of the working fluid (Gallego, Herrera, Sierra, Zapata, & Cacua, 2020). Solar collectors are a practical solution to tap natural energy savings with eco-friendly perspectives. The flammability, limitation of temperature around 400°C, and environmental toxicity are the major disadvantages of using thermal oils (Blanco & Miller, 2017). The potential application for phase change includes low-temperature heat recovery (Li et al., 2020). Sarafraz, Tian, Tlili, Kazi, and Goodarzi (2019) showed that both the fill ratio and the tilt angle were key parameters affecting the system's thermal performance.

Kasaeian, Daneshazarian, Rezaei, Pourfayaz, and Kasaeian (2017) reported augmented heat transfer and maximum thermal efficiency of 30.4% by using 0.3%

MWCNT/EG nanofluid. 234% heat transfer coefficient augmentation was reported (Mwesigye, Yılmaz, & Meyer, 2018), using MWCNT/therminol VP-1 nanofluid. A decrease of 20-30% entropy generation, using 6% Cu-Therminol nanofluid (Mwesigye, Huan, & Meyer, 2016) is reported. Optimization of nanoparticle fraction is highly important for hydraulic and thermal performance efficiencies.

To achieve a downward passive heat transfer, designing a reverse thermosyphon with self-action and two working media is the best option. The practical applications of such devices are not found due to the need for harmful or costly refrigerants or the requirement of lower pressure in the device than the ambient. Heat-carrying action and pumping action can be performed separately with a second medium having a low boiling point. Introducing fins (Chu, Shen, & Wang, 2021), using the working medium as methanol (Huang, Lee, Tarau, Kamotani, & Kharangate, 2021) or nanofluid (Wang et al., 2020), the influence of boundary conditions (Nguyen & Merzari, 2020), and using an evaporator, smaller than the condenser (Cisterna, Fronza, Cardoso, Milanez, & Mantelli, 2021; Ng, Yu, Wu, & Hung, 2021) increase the thermosyphon thermal efficiency.

Anti-gravity flow movement of 0.38m with a heat transport of 100W in a heat pipe loop with porous wick (Tang, Zhou, Lu, & Xie, 2012) and 1m with 220W in a 10m cylindrical heat pipe (Nakamura, Odagiri, & Nagano, 2016). In most anti-gravity passive heat transfer reports, the condenser temperature is kept as the room temperature. This work is conducted with the varying heat flux and anti-gravity conditions available for a solar collector. The temperature along the thermosyphon is studied with the flow rate.

Advancement of dependable and economic means of autonomous heat transfer downwards is not yet tested in the peak annual temperature range. Simultaneous use of two working fluids in a reverse thermosyphon integrated with a bubble pump system, where water transfers heat and a low boiling substance creates pressure above atmospheric pressure and sets water in circulation is a new concept for this solution. Contrary to the thermosyphons that operate cyclically, the proposed system could be distinguished by continuous operation. Outcomes of this study could be used to evaluate the pertinence of reverse thermosyphons in the backdrop of renewable energy generation and play a dominant role in the prosperity of several countries.

2. Materials and Methods

The continuity equations are defined in Fluent (Fluent, 2020) as:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \vec{v}) = 0 \qquad 1)$$

ρ and $\vec{v}$ denotes density and velocity vector.

The momentum equation is:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla(\rho \vec{v}\vec{v}) = -\nabla \mathrm{p} + \nabla \vec{\tau} + \rho \vec{g} + S_g \qquad 2)$$

$\vec{\tau}$ is stress tensor, g is gravity acceleration and p is pressure. The energy equation for phase change (Fluent, 2020) is:

$$\frac{\partial}{\partial x}(\rho H) + \nabla\big(\vec{v}(\rho H + p)\big) = \nabla(k \nabla \mathrm{T} + \vec{\tau}\vec{v}) S_H \qquad 3)$$

H is internal energy, h is the sum of sensible enthalpy and ΔH is latent heat, and k is thermal conductivity. Sensible enthalpy h can be found (Fluent, 2020) below:

$$\Delta H = \beta L \qquad 4)$$

β is the liquid fraction of PCM and L is the latent heat of PCM. Liquid fraction β is defined by Fluent (2020) as:

$$\beta = 0, if\ T < T_{solidus} \qquad 5)$$

$$\beta = 1, if\ T < T_{liquidus} \tag{6}$$

$$\beta = \frac{T - T_{solidus}}{T_{liquidus} - T_{solidus}}, \text{if } T_{solidus} < T < T_{liquidus} \tag{7}$$

$T_{solidus}$ and $T_{liquidus}$ are material temperatures at solid and liquid phases and the temperature of the PCM is denoted by T.

The equation for the stress tensor $\vec{\tau}$ is as follows (Fluent, 2020):

$$\vec{\tau} = \mu \left[(\nabla \vec{v} + \nabla \vec{v}^T) - \frac{2}{3} \nabla \cdot \vec{v} I \right] \tag{8}$$

I is unit tensor and μ is molecular viscosity. For natural convection and mushy region, source term S_g was considered as follows (Pawar & Sobhansarbandi, 2020):

$$S_g = \rho \vec{g} \beta \left(T - T_{ref} \right) - \frac{(1-\beta)^2}{\beta^3 + \epsilon} A_{mushy} \vec{v} \tag{9}$$

The second term in equation 9 relates to the porosity of medium in each cell where liquid fraction is considered (Pawar & Sobhansarbandi, 2020). A_{mushy} is mushy zone constant (mushy zone is mixed solid-liquid region) set at 100000. To avoid a division by zero, ϵ was set as 0.001 (Fluent, 2020).

Heat removal factor (F_R) is the ratio of actual heat transfer to maximum heat transfer. The Hottel–Whillier–Bliss (Hottel & Whillier, 1958; Bliss, 1959; Whillier, 1967) equations were used to evaluate the instantaneous thermal efficiency (η_{th}), using a modified efficiency curve model. The absorbed radiation is the product of the incident radiation and the transmittance-absorptance product (τ_α).

Relationship of absorptance of the absorber (α) cover transmittance (τ), overall heat transfer coefficient (U_L, W/m^{2}K), inlet fluid temperature (T_i, K), ambient temperature (T_a, K) and global solar radiation (G_T, W/m^2) are given as follows (Duffie & Beckman, 2013):

$$\eta_{th} = \frac{Q}{A_c G_T} = \frac{\dot{m} C_p (T_0 - T_i)}{G_T} \tag*{10)}$$

Q is the rate of solar energy gained (W), $\dot{m}$ is the fluid flow rate (l/hr), C_p is heat capacity (J/kgK), T_i, T_a and T_0 are temperature (K) of the inlet, ambient, and outlet respectively, A_c is surface area (m^2).

$$\eta_{th} = F_R \tau_\alpha - F_R U_L \frac{(T_i - T_a)}{G_T} \tag*{11)}$$

Nanofluid closed circulation is maintained by keeping the evaporator with a tilt angle of 65°. Fluid inlet, surface, ambient, and outlet temperatures, friction factor, wind velocity, and global solar radiation are found. The flow rates of nanofluids are varied for the study. The Reynolds number in the collector $(Re = 4\dot{m}/\pi d\mu)$ of nanofluids with the viscosity (μ) is calculated by the tube hydraulic diameter/inner diameter (d).

Increased resistance of fluid causes pressure drop decrease. The friction factor is:

$$f = \frac{\Delta P}{\left(\frac{l}{d}\right)\left(\frac{\rho V^2}{2}\right)} \tag*{12)}$$

l is tube length (mm), ΔP is pressure drop, v is velocity, and ρ is density.

Nusselt number is:

$$Nu = \frac{hd}{k} \tag*{13)}$$

Heat transfer coefficient (W/m^2K) $h = Q/A_c(T_s - T_b)$, T_s is outlet fluid temperature after time t, T_b is arithmetic average of outlet and inlet temperatures, k is the thermal conductivity (W/mK). $A_c = \pi dL, T_s = T_1 + T_2 + T_3/3$ and $T_b = T_o + T_i/2$.

Transmittance for visible wavelength and absorptance of the plate are $\tau = 0.89$ and $\alpha = 0.89$.

The thermal resistance of the evaporator R_e (°C/W) is:

$$R_e = \frac{T_{ew} - T_{ev}}{Q_e} \quad 14)$$

T_{ew} is wall temperature, T_{ev} is vapor temperature and Q_e is input power. Condenser thermal resistance (R_c) is:

$$R_c = \frac{T_{cv} - T_{cw}}{Q_e} \quad 15)$$

T_{cw} is wall temperature of condenser, Q_e is input power and T_{cv} is vapor temperature. Total thermal resistance (R) of the heat pipe is (Mousa, 2011):

$$R = \frac{T_{ew} - T_{cw}}{Q_e} \quad 16)$$

Heat transfer coefficient at the evaporator h_e (W/m^2K) is:

$$h_e = \frac{Q}{A(T_{ew} - T_e)} \quad 17)$$

The simulation performance of thermosyphon is illustrated by thermal resistance on an overall basis.

Heat transfer rate Q on an overall basis is:

$$Q = \frac{T_e - T_c}{R_{eq}} \quad 18)$$

where T_e and T_c are evaporator and condenser average wall temperatures, R is thermal resistance (K/W) and Q is power throughput.

The filling ratio FR is:

$$FR = \frac{V_l}{Al_e} \times 100 \quad 19)$$

A is the internal cross-section area, l_e is the length of an evaporator, and V_l is liquid volume.

3. CFD modeling

A pressure-based solver, a SIMPLE algorithm with PRESTO!, least-square cell-based method, and second-order upwind mode were used. The evaporator section is indicated by the distance between 0 and 200mm, the distance between 300 and 500 mm indicates the condenser section and the middle section is the adiabatic region. A UDF code is used for creating a closed-loop piping system and another UDF for the phase change process.

The simulation was conducted with a constant heat flux of a solar thermal collector, placing the thermosyphon above it. Reverse thermosyphon having two working media (Figure 1) consist of a U-tube (U) with a fin (F) of the solar thermal collector providing heat flux, bubble pump (B), an evaporator (E), a separator (S), and a condenser (C); which are interconnected to become a closed system of liquid. The evaporator is partly filled with a liquid medium (LM) of heat transfer and pumping medium (PM). A film of the pumping medium of 9-10mm on the bulk of the liquid medium is introduced as a layer. Liquid medium shows maximum flow rate during slug flow (Hanafizadeh, Karimi, & Saidi, 2011) in a two-phase study of a bubble pump.

Stagnation (on-demand) operation was used for preliminary simulation with no circulation of HTF throughout the system till achieving maximum energy storage and circulation of HTF is initiated in the system at a later time of the day (Papadimitratos, Sobhansarbandi, Pozdin, Zakhidov, & Hassanipour, 2016). This is a new system of study with no prior experimental work or data.

Figure 1 Circuit diagram of bubble pump enabled reverse thermosyphon with two working media.

4. Mesh geometry and independence

GAMBIT software is used to generate two-dimensional geometry and meshing. The first grid size was 0.01 mm and the growth ratio was 1.2. 36 cells constituting one cell layer are set apart for top and lower walls since heat conduction does not take place through these. Grid-independence results for the reverse thermosyphon charged with water for heat input of 100 W for a mesh size of 19500 (cells) shows that the evaporator, separator, and condenser registered mean temperatures of 30.51, 26.03, and 21.48°C respectively and that for a mesh size of 69276 were 29.16, 25.86 and 22.45°C respectively and that for a mesh size of 129944 were 29.32, 25.89 and 22.47 respectively. Thus, the mesh size selected for the numerical study is 69276. The solid region contains 15092 cells and 54184 quad cells for the fluid region. As a result, 69,276 cells are generated. Fifteen cell layers are selected to analyse the film of liquid getting developed near the left and right wall regions. The mesh sizes of 69276 and 129944 revealed very similar values of mean temperature for evaporator, separator, and condenser.

5. Initial and boundary conditions

- Constant heat flux at evaporator wall with non-slip at inner walls.
- Zero heat flux at upper and lower ends as well as a separator.
- Convection heat transfer with heat transfer coefficient values from the CFD simulation of the condenser at the walls of the condenser.
- Interfaces between solid and fluid regions of the heat pipe are assumed as the coupled wall.
- Physical properties at 298.15 K are assumed temperature independent; except density and surface tension of liquid phase.

6. Validation

As shown in Figure 2, as the heat flux increases from 100.41 to 376.14W/m^2, the temperature between evaporator and condenser for computational results were compared with the results of experimental work (Fadhl, Wrobel, & Jouhara, 2013) and observed that the temperature increases, and the thermal resistance decreases. The average deviation percentage for temperature between evaporator and condenser for experimental and computational is 3.7%. For temperature between evaporator and condenser, the highest deviation between experimental and computational is observed at a heat flux of 100.41W/m^2, the deviation after that from a heat flux of 172.87W/m^2 to 376.14W/m^2 is 14%. In the experiment, only two thermocouple positions were used to record the average temperature of the evaporator section (Fadhl, Wrobel, & Jouhara, 2013) and this might be the reason for the large deviation at low heat flux. Therefore excluding the lowest heat flux of 100.41W/m^2, the computational results obtained for temperature are extremely close to the experimental values for temperature between evaporator and condenser. In the case of heat inputs beyond 170 W, the thermal resistance shows relatively independent of the heat input (Fadhl, Wrobel, & Jouhara, 2013). In the case of lower heat inputs, the thermal resistance tends to increase. In CFD software, the ideal adiabatic condition was considered and so the deviation was obtained while comparing both the results.

Figure 2 Validation of computational results.

7. Results

Figure 3 depicts the distribution of temperature on the thermosyphon surface and when ammonia is used, the stability of the distribution of temperature became lower than the cases of using water. The flow rate was kept as 10l/h with a filling ratio of 60%. Water that has a higher evaporation temperature than ammonia might be the major cause of this behaviour. The flow rate of cooling water gets improved, though the distribution of temperature at the evaporator surface gets decreased than that of water. When we use ammonia, the reverse effect is noticed. Using ethylene glycol having better heat transfer capacity, the stability of temperature distribution and mean surface temperatures of condenser and evaporator are higher. The melting and freezing onset difference of roughly $30°C$ makes Erythritol a strong candidate as an HTF due to Erythritol's ability to stay in liquid form for a longer period to prevent thermal expansion from crystallizing and transferring heat for a longer period.

Figure 3 Average temperature of Thermosyphon surface using PCM.

Ethylene glycol showed an efficiency of 89% for the 200W heat input. The efficiency was observed as 37.8 % for the 500W heat input (Figure 4). As the heat input increases, the thermal efficiency decreases. Though Erythritol showed less thermal efficiency than Ethylene glycol and Ammonia, it showed the highest thermal efficiency of 37.8 % at 500W heat input. As the average real heat input is higher than 200W most times, Erythritol is adjudged as a better PCM in Thermosyphon with variable heat input from U-tube solar collector. Using phase change materials, by increasing heat input, thermal efficiency is lowered as the very high latent heat will be turned to superheated steam.

Figure 4 Thermal Efficiency of PCM in Thermosyphon.

The highest thermal efficiency in Figure 5 is 58% at 500 W heat input for 0.2% Ag. The thermal efficiency of 0.2% Ag (29.6%) is much lower at lower heat input but it becomes the highest at high heat input. The thermal efficiency of the nanofluids increases with an increase in heat input from 200W to 500W. The thermal efficiency observed for 0.2% MWCNT is 52% at 500W heat input and the thermal efficiency observed for 200W is 38%. The thermal efficiency at 200W is higher than that of 0.2% Ag. Hence, the Ag nanoparticle shows better thermal efficiency at higher heat input and the MWCNT nanoparticle shows better thermal efficiency at lower heat input. Using nanofluid materials, by increasing heat input, thermal efficiency also is enhanced.

Figure 5 Thermal Efficiency of Thermosyphon using different nanoparticles.

In Figure 6, it is observed that the thermal resistance decreases as heat flux increases from 100 to 500W/m^2. The average thermal resistance is 0.37 for 100W/m^2 and 0.07 for 500W/m^2 by using Erythritol. Similar trends were reported (Solomon, Roshan, Vincent, Karthikeyan, & Asirvatham, 2015; Sözen et al., 2016). Thermal resistance decreases with heat flux non-linearly as the heat transfer mechanism changes to nucleate boiling from convection.

Figure 6 Thermal Resistance in Thermosyphon using Erythritol as PCM.

Identical results of thermal efficiency improvement of 95.94 - 137.07% were seen by using Ethylene glycol, and upon changing the flow rates to 30l/h from 10l/h (Figure

7), compared to water. Similar behaviour is noticed at almost all the heat puts investigated in this study. Hence, the thermal efficiency elevation or reduction depends on the flow rate. Karthikeyan, Vaidyanathan, and Sivaraman, (2010) reported an elevation, and Sözen et al., (2016) reported a reduction in thermal efficiency with the rise in flow rate. The reduction of the thermal efficiency consequent on flow rate enhancement might be due to the flow reversal at the outlet which reduced both the flow rate and the temperature rise.

Figure 7 Thermal efficiency enhancement in Thermosyphon using different PCM and flow rates.

As the heat input increases from 100 to 500W, the difference in temperature and the temperature between the evaporator and condenser increases (Figure 8). The average temperature between evaporator and condenser is 83.02$°C$ for Ethylene glycol and 48.52$°C$ for Ammonia. The average temperature of the evaporator of the Thermosyphon by using ethylene glycol as the PCM was 122.25$°C$.

When Ammonia and Ethylene glycol are used as PCM in working fluid water with a flow rate of 10l/h and 200W heat input, the mean difference in temperature was 12.9$°C$ for Ethylene glycol and 11.3$°C$ for Ammonia. When the heat input was changed to 500W, these values were 16.57$°C$ and 22.9$°C$ respectively (Figure 8). When the flow rate was changed to 30l/h, these values were 17.5$°C$ and 4.7$°C$ respectively with 200W. When the flow rates were similar, a rise in heat input makes the differences in temperature higher, and when the flow rate rises, the temperature differences get lowered concerning the working fluids investigated (Sözen et al., 2016). Ethylene glycol was found to be a better PCM at lower heat input and Ammonia was found to be a better PCM at higher heat input.

Figure 8 Average Cooling Temperature of Thermosyphon using different PCM and flow rates.

As shown in Figure 9 as the heat flux increases from 100 to 500W/m^2, the temperature between evaporator and condenser increases with a very high rate of increase from 100 to 200W/m^2. The average difference in temperature is 68°C for 0.2% MWCNT with Erythritol as the PCM is highest for fill ratio 0.8 with 500W/m^2 heat flux. As the fill ratio increases from 0.3 to 0.8, the temperature between the evaporator and condenser increases. The thermal efficiency has registered a maximum of 70.5% for fill ratio 0.5 with 500W/m^2 heat flux. Even though the fill ratio of 0.8 showed a thermal efficiency at par with a fill ratio of 0.5 from 100 to 250W/m^2 heat flux, it showed a less thermal efficiency after 250W/m^2 heat flux input. Thermal efficiency increases with an increase in heat flux.

Figure 9 Thermal efficiency and difference in Temperature in Thermosyphon.

The temperature profile from the start of the evaporator (Es) at 20mm length of Thermosyphon to the end of the condenser (Ce) using 0.2% MWCNT and Erythritol as PCM with heat input 100-500W is shown in Figure 10. The temperature increases as heat input increases in the evaporator with the highest value of 97.99°C for 500W at 170mm length (end of the evaporator, Ee). The adiabatic section also shows a decrease in temperature from its start (As) to end (Ae) for the lower heat input, whereas a slight increase is registered with the higher heat inputs from 220-270mm length. A similar trend is observed in the Condenser for all the heat inputs, whereas a slight increase is observed with 200W from 350-450mm length. The lowest condenser temperature values, 50.49°C

for 500W, and 30.5°C for 100W were noticed. The whole length of the Thermosyphon has shown a temperature variation of 20-42°C.

Figure 10 Surface Temperature of Thermosyphon using nanoparticles and PCM.

8. Conclusion

The present study used a combination of heat transfer and storage in a single unit, in which the U-tube effectively replaces the thermosyphon heat pipe. A simple condenser section is created by extending one side of the U-tube geometry where the Heat Transfer Fluid in the U-tube transfers heat to heat water as if in a simple heat exchanger configuration.

Erythritol is adjudged as a better PCM in Bubble pump enabled Reverse Thermosyphon with variable heat flux from U-tube solar collector than Ethylene glycol and Ammonia concerning the average temperature of Thermosyphon surface. Ag nanoparticle shows better thermal efficiency at higher heat flux and MWCNT nanoparticle shows better thermal efficiency at lower heat flux at a concentration of 0.2%. Based on the flow rate, the thermal efficiency tends to either decrease or increase. Based on the flow rate for ethylene glycol the efficiency decreases with increases in flow rate, but for water and ammonia, it increases. The reduction of the thermal efficiency consequent on flow rate enhancement might be due to the flow reversal at the outlet which reduced both the flow rate and the temperature rise. Ethylene glycol was found to be a better PCM at lower heat flux and Ammonia was found to be a better PCM at higher heat flux, compared to water, concerning cooling temperature.

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Figure

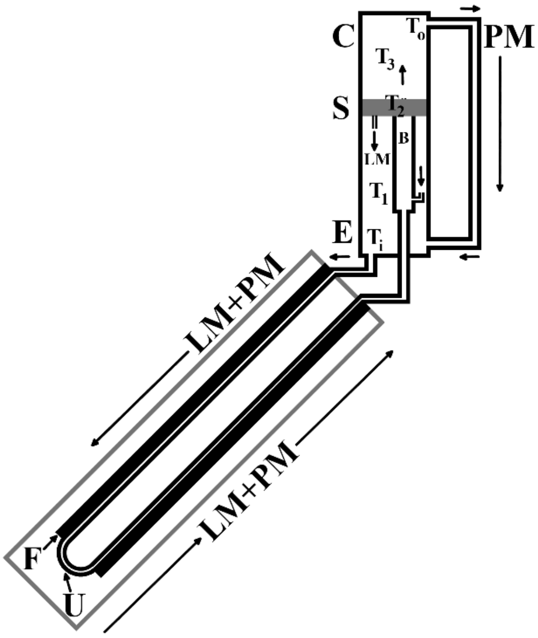

Figure 1 Circuit diagram of bubble pump enabled reverse thermosyphon with two working media.

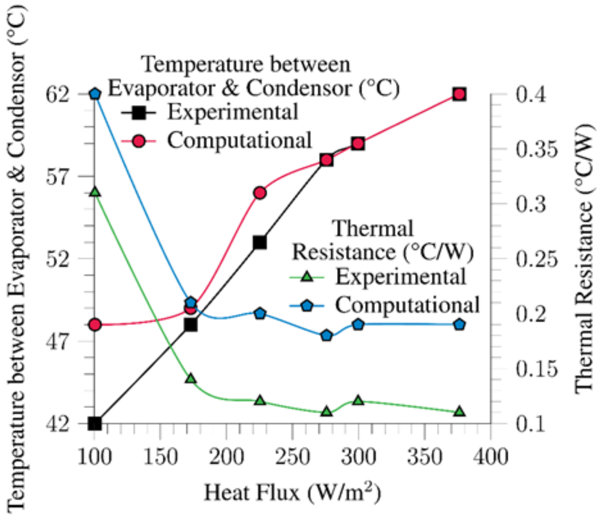

Figure 2 Validation of computational results.

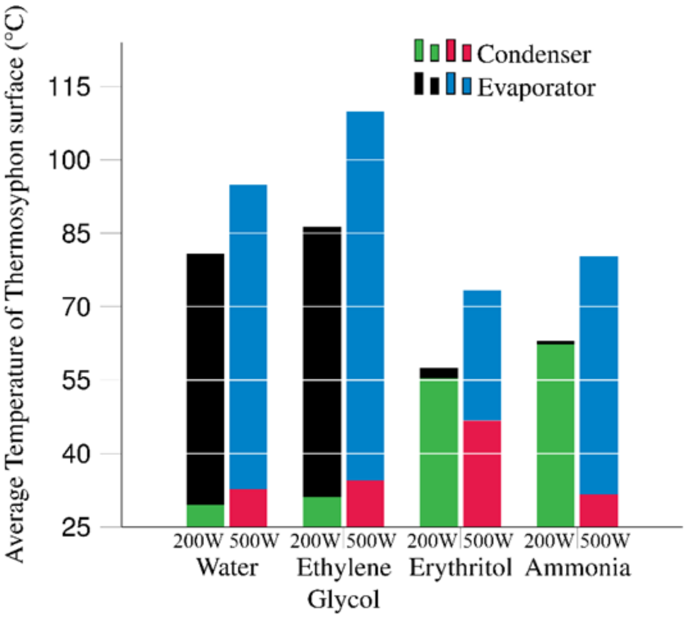

Figure 3 Average temperature of Thermosyphon surface using PCM.

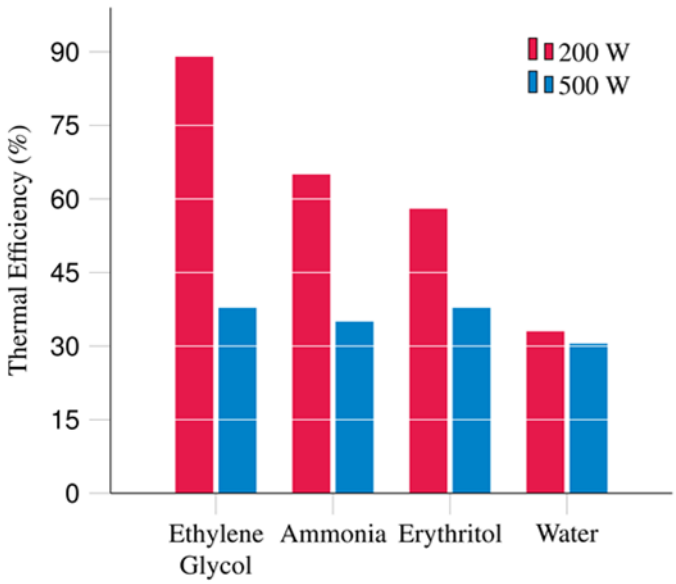

Figure 4 Thermal Efficiency of PCM in Thermosyphon.

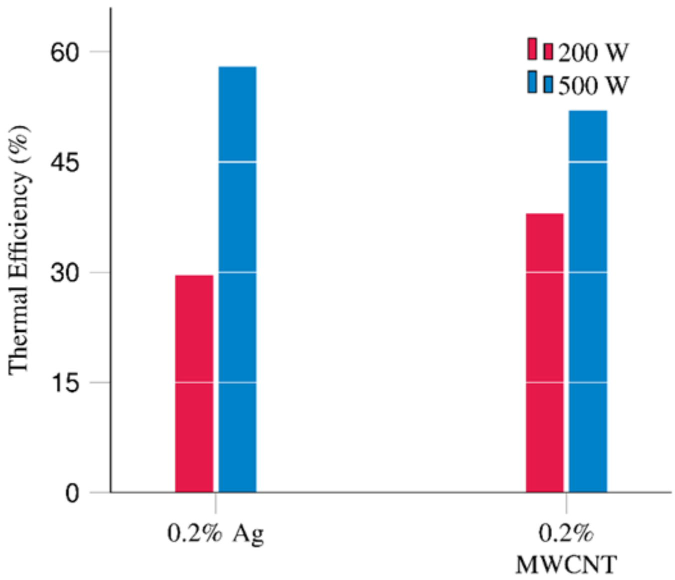

Figure 5 Thermal Efficiency of Thermosyphon using different nanoparticles.

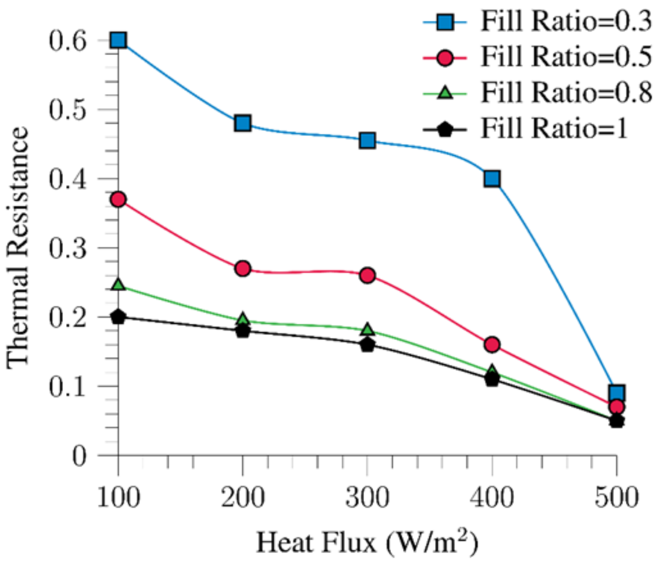

Figure 6 Thermal Resistance in Thermosyphon using Erythritol as PCM.

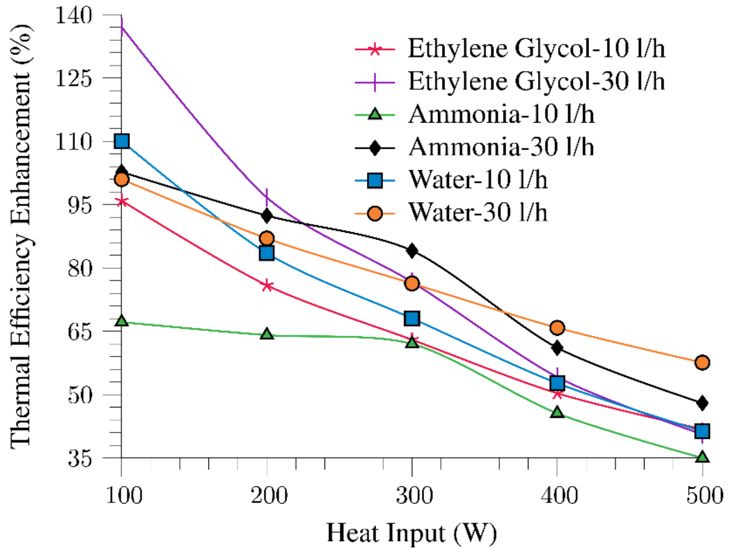

Figure 7 Thermal efficiency enhancement in Thermosyphon using different PCM and flow rates.

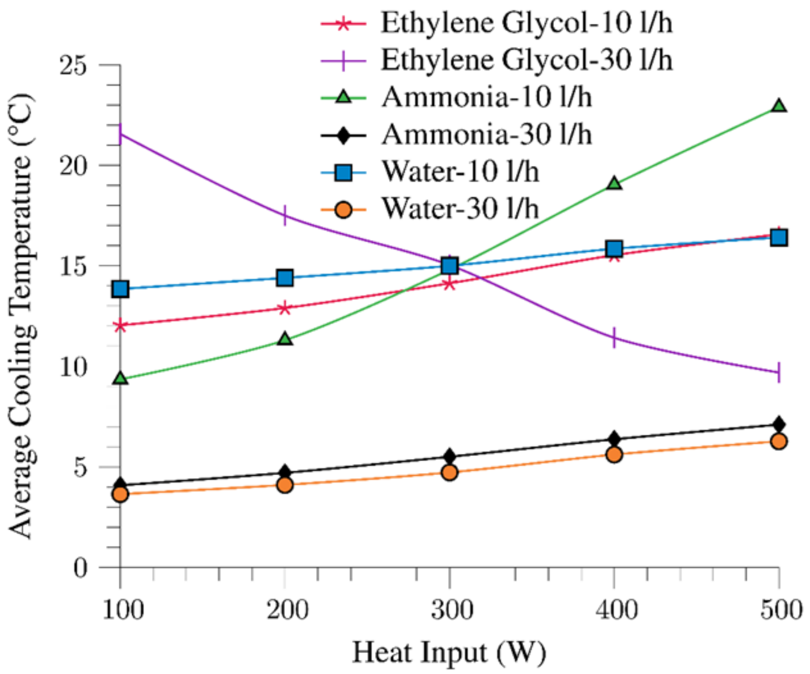

Figure 8 Average Cooling Temperature of Thermosyphon using different PCM and flow rates.

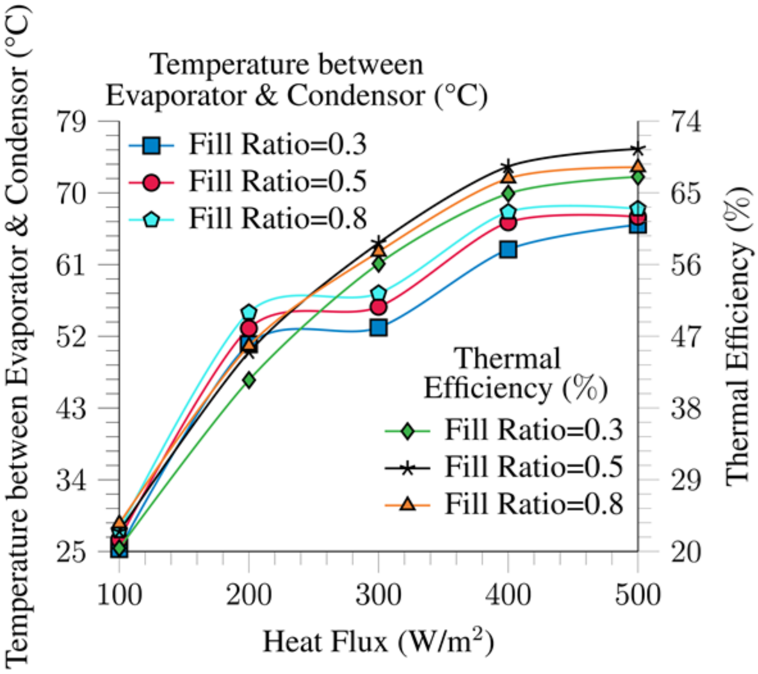

Figure 9 Thermal efficiency and difference in Temperature in Thermosyphon.

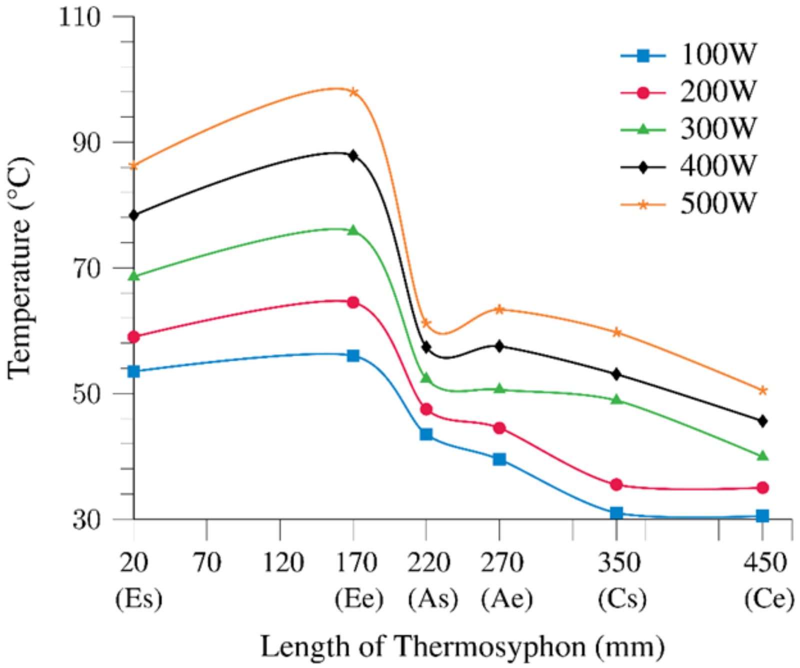

Figure 10 Surface Temperature of Thermosyphon using nanoparticles and PCM.