

# Numerical Investigation of Effect of Fill Ratio and Inclination angle on a Thermosiphon Heat Pipe Thermal Performance

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1 **Numerical Investigation of Effect of Fill Ratio and Inclination angle on a Thermosiphon Heat**  
2 **Pipe Thermal Performance**

3  
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9 **Abstract**

10  
11 Computational Fluid Dynamic (CFD) modelling of a heat pipe is a powerful tool that can be used to  
12 investigate the complex physical phenomena of the evaporation and condensation phase change  
13 processes inside thermosiphon heat pipes. In this work, a new CFD simulation of two phase flow inside  
14 thermosiphon heat pipe is carried out to investigate the effect of fill ratio (ratio of liquid volume to the  
15 evaporator volume) and inclination angle on its thermal performance in terms of temperature  
16 distribution and thermal resistance using FLUENT (ANSYS 15). Results of the CFD simulation were  
17 compared to published experimental data showing good agreement with maximum deviation of 4.2%  
18 and 8.1% for temperature distribution and thermal resistance, respectively. In addition, numerical  
19 results of inclination angle were also compared with experimental data in terms of thermal resistance  
20 giving maximum deviation of 1.3%. Using the validated CFD modelling, results showed that at low fill  
21 ratio and low inclination angle, there was a significant increase in the evaporator temperature.  
22 Regarding the thermal resistance, a fill ratio of 65% and inclination angle of 90° produced the lowest  
23 thermal resistance for all the heat input values used. Also, as heat input increases, the effect of the fill  
24 ratio and inclination angle becomes more significant.

25 **Keywords:** CFD simulation, Ansys fluent, two phase closed thermosiphon, fill ratio, Inclination angle

NUMECLATURE					
$C_p$	Specific heat	J/kg K	$t$	Time	s
$D$	Outside diameter of thermosiphon	m	$u$	Velocity	m/s
$E$	Total energy per unit mass	J/kg	<b>Greek symbols</b>		
$F_s$	Continuum surface force	Kg/m <sup>2</sup> s <sup>2</sup>	$\alpha$	Volume fraction	
$g$	Acceleration gravity	m/s <sup>2</sup>	$\mu$	Dynamic viscosity	Pa s
$h$	Heat transfer coefficient	W/m <sup>2</sup> K	$\rho$	Density	Kg/m <sup>3</sup>
$h_{fg}$	Latent heat	J/kg	$\sigma$	Surface tension coefficient	N/m
$L$	Height	m	<b>Subscripts</b>		
$K$	Thermal conductivity	W/m K	$con$	Condenser	
$K_C$	Surface curvature		$conv$	Convection	
$P$	Pressure	Pa	$cw,av$	Condenser wall, average	
$Q_{con}$	heat removed from condenser	W	$l$	Liquid	
$S_q$	Energy source term	J/m <sup>3</sup> s	$mix$	Mixture	
$S_{am}$	Mass source term	Kg/m <sup>3</sup> s	$Sat$	Saturation	
$T$	Temperature	K	$v$	Vapour	

26

## 27 1. Introduction

28 Heat pipes are devices for transferring heat from one point to another by evaporating and condensing  
 29 the working fluid in a sealed vessel. They have the advantages of low thermal resistance, compact and  
 30 uses small amount of working fluid thus are used in wide range of applications such as electronics  
 31 cooling, heat exchangers and solar collectors. The main sections in the heat pipe are evaporator and  
 32 condenser in which the heat is absorbed by working fluid in the evaporator side and rejected in the  
 33 condenser. The vapour condensates by giving up its latent heat to the coolant at the condenser section  
 34 and the condensate returns back to the evaporator by capillary force in the case of wicked heat pipe or  
 35 by gravity in the case of wickless heat pipe (Thermosiphon). Considerable interest has been paid to  
 36 wickless Two-Phase Closed Thermosiphon (TPCT) heat pipes due to their simple construction and low  
 37 cost [1-3].

38 Although many experimental studies have been performed to examine the impact of working fluid fill  
 39 ratio and inclination angle on the performance of different types of heat pipes, limited number of these  
 40 studies have tested the performance of two phase closed thermosiphon. Noie [4] studied the effect of

41 filling ratio and the evaporator aspect ratio (evaporator length to evaporator diameter) on the heat  
42 transfer performance of the TPCT for a range of heat input. It was found that changing the fill ratio can  
43 reduce the evaporator wall temperature depending on the aspect ratio. Jiao et al [5] developed an  
44 analytical model to investigate the effect of filling ratio on the steady state heat transfer characteristics  
45 of a vertical wickless heat pipe and compared the results with their experimental work. They reported  
46 that the fill ratio depends on geometrical parameters and heat input. Jouhara and Robinson [6]  
47 investigated experimentally the effect of using different working fluids namely, water, FC-84 and FC-  
48 3283 and two filling ratios (100% and 50%) on the performance of thermosiphon heat pipe. A small  
49 size thermosiphon of 10 W with different working fluids (water, methanol and acetone) and liquid fill  
50 at various input energy has been investigated by Mozumder et al [7]. The study showed that the effect  
51 of charging liquid can be indicated by temperature difference, thermal resistance and overall heat  
52 transfer coefficient. The influence of the charged liquid and adiabatic length on the thermal  
53 performance of a long heat pipe charged with R-134a has been examined by Sukchana and Jaiboonma,  
54 2013 [8] who concluded that the optimum liquid charge and heat flux suitable for shorter adiabatic  
55 section were 15 % and  $5.92 \text{ kW/m}^2$ , respectively. Chehade et al [9] tested effects of fill ratio, inlet  
56 cooling water temperature and mass flow rate in condenser jacket on the performance of the two-phase  
57 closed loop Thermosiphon. They concluded that the best fill charge ratio is between 7% and 10% and  
58 the fastest start up occurs by using the optimal fill ratio.

59 An experimental study has been performed by Manimaran et al [10] to examine the effect of heat  
60 input, charge fill ratio, and angle of inclination on thermal characteristics of a heat pipe, who reported  
61 that the lower thermal resistance was obtained at fill ratio 75% and vertical orientation. Sadeghinezhad  
62 et al and Ghanbarpour et al [11, 12] investigated the effect of different nanofluids and inclination angle  
63 on the thermal characteristics of a sintered wick and screen mesh heat pipe, respectively. They reported  
64 that the orientation has a strong effect on the thermal performance of a heat pipe and the lower thermal

65 resistance is obtained at an angle of  $60^\circ$ . The effect of inclination angles on thermal performance of  
66 ammonia pulsating heat pipe and copper nanofluid heat pipe has been performed by Xue Zhihu and Qu  
67 Wei, and Senthilkumar et al [13, 14], respectively. They demonstrated that the thermal performance of  
68 studied heat pipes increases as the inclination angle increases. Nazarimanesh et al [15] performed an  
69 experimental study to investigate the thermal of performance sintered heat pipe at various degree of  
70 inclination. They found that the lowest thermal resistance for base working fluid is achieved at an angle  
71 of  $90^\circ$ .

72 There have been limited published CFD research work conducted to analyse TPCT heat pipes despite  
73 their numerous applications [16]. Fadhl et al [16] developed a CFD model to simulate condensation and  
74 evaporation processes inside the TPCT. CFD results were compared with experimental data in terms of  
75 temperature distribution along the heat pipe and thermal resistance at different heat inputs. They  
76 reported that the thermal performance of thermosiphon heat pipe improved by increasing heat input  
77 over 172 W.

78 Alizadhdakel et al [17] have reported experimentally the effect of input energy and fill ratio on the  
79 performance of a wickless heat pipe. They have also carried out a CFD simulation to investigate the  
80 phase change phenomena with effect of noncondensable gases throughout thermosiphon, and compered  
81 the results of experiment and CFD model. An optimum value for fill ratio of 50% was concluded for  
82 the studied thermosiphon and heat input range. A three dimension CFD analyses to investigate the  
83 effect of water with different concentrations of nanoparticles on the thermosiphon heat pipe  
84 performance has been performed by Humic and Humic [18]. Results showed that the concentration of  
85 nanoparticles in water had a considerable effect on the heat transfer characteristics of The TPCT. Fadhl  
86 et al [19] carried out a CFD simulation of a wickless heat pipe with R134a and R404a as working  
87 fluids, and Results were compared with published experimental data in terms of temperature  
88 distribution along the wall of TPCT. They found that thermal characteristics of both fluids inside the

89 thermosiphon differ significantly from that of water. A numerical CFD analysis and experimental  
90 work to investigate cooling water flow rate, input energy and orientation on the thermal performance of  
91 a thermosiphon heat pipe have been carried out by Abdullahi [20]. Results show that the heat transfer  
92 characteristics of the TPCT increase as inclination angle and input energy increase. Kim et al [21]  
93 implemented a CFD simulation to study the effect of the condensation frequency on the mass transfer  
94 rate during phase change inside a thermosiphon heat pipe. The study concluded that the condensation  
95 frequency should be considered as  $0.1 \times (\rho_l / \rho_v)$  to accurately simulate the mass transfer process during  
96 condensation and evaporation phenomena.

97 From all mentioned experimental investigations, it can be concluded that the best fill ratio and  
98 inclination angle for any heat pipe depend on many factors such as geometry, heat input, type of liquid  
99 and operating conditions. Therefore, according to these parameters, the suitable inclination angle and  
100 liquid charge ratio change from one heat pipe to another and investigations to identify the best fill ratio  
101 and inclination angle is needed whenever any of these parameters is changed. For that reason, a  
102 numerical study should be used to specify optimum charging ratio and orientation before the  
103 experimental work to reduce time and cost of these investigations. In addition, all stated numerical  
104 CFD studies were not employed to analyse these effects. Thus, in the present study, a new CFD model  
105 was developed to investigate the influence of five different values of fill ratio (25%, 35%, 65%, 80%  
106 and 100%) of water and inclination angle range of (10, 30, 50, 70, and 90°) on the thermal performance  
107 of a two-phase closed thermosiphon at various values of heat input. Consequently, wide range of  
108 affecting parameters can be modelled to investigate their effect on the performance of the heat pipe.

109

## 110 **2. GOVERNING EQUATIONS**

111 Many researchers have used Volume of Fluid (VOF) model to solve numerically a multiphase flow  
112 because it is easier compared with finite volume method. Reasons behind that are that the location of

113 the interface between phases varies for each computational step, and physical properties at the interface  
114 are also changeable which make the numerical simulation computationally expensive. Thus, solving  
115 these problems can be achieved using VOF model by defining the motion of all phases and tracking the  
116 location of the interface accordingly [16-28]. In the VOF model, movement of different fluids can be  
117 tracked by solving a single set of Navier-Stocks equations for the volume fraction of each fluid  
118 throughout the computational cell [28]. Therefore, the existence of a certain phase in any control  
119 volume can be easily specified from the volume fraction according to the following three cases:

120  $\alpha_l = 1$ : The cell is full of vapour

121  $\alpha_v = 0$ : The cell is full of liquid

122  $0 < \alpha_v < 1$ : The cell contains a mixture of liquid and vapour

123 The third case means

$$124 \quad \alpha_l + \alpha_v = 1 \quad [1]$$

125 Where  $\alpha_l$  and  $\alpha_v$  are volume fractions of liquid and vapour respectively.

126 In order to define the motion of the fluid inside the TPCT during evaporation and condensation  
127 processes, the governing equations of mass continuity, momentum and energy with source terms are  
128 solved using Fluent Ansys.

129

## 130 **2.1 Continuity Equation**

$$131 \quad \frac{\partial}{\partial u}(\rho) + \nabla \cdot (\rho \vec{u}) = 0 \quad [2]$$

132 Where,  $\rho$  and  $u$  are the density and velocity of the fluid.

133 To track the interface between phases, solution of eq. (2) for the volume fraction is needed. Therefore,  
134 for the secondary phase (liquid phase) of VOF model, this equation can be written as follow:

135

$$136 \quad \frac{\partial}{\partial u}(\alpha_l \rho_l) + \nabla \cdot (\alpha_l \rho_l \vec{u}) = S_{am} \quad [3]$$

137 Where,  $S_{am}$  is the mass source term that can be used to find the mass transport from one phase to  
 138 another during the evaporation and condensation processes. The above equation solves for the  
 139 secondary phase ( $l$ ) only and the volume fraction for the primary phase ( $v$ ) can be calculated using eq.  
 140 (4):

$$141 \quad \sum_{k=1}^2 \alpha_k = 1 \quad [4]$$

142

## 143 **2.2 Momentum Equation**

$$144 \quad \frac{\partial}{\partial u}(\rho \vec{u}) + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla p + \rho \vec{g} + \nabla \cdot \left[ \mu (\nabla \vec{u} + \nabla \vec{u}^T) - \frac{2}{3} \mu \nabla \cdot \vec{u} \right] + F_S \quad [5]$$

145 Where, the fluid properties  $\rho$  and  $\mu$  are expressed by eq. (6) and eq.(7) respectively. According to the  
 146 VOF model, the physical properties are determined for the mixture only based on the value of volume  
 147 fractions of liquid and vapour.

$$148 \quad \rho = \alpha_l \rho_l + \alpha_v \rho_v \quad [6]$$

$$149 \quad \mu = \alpha_l \mu_l + \alpha_v \mu_v \quad [7]$$

150  $F_S$  is the Continuum Surface Force (CSF) acting on the interface between two phases which was  
 151 proposed by Brackbill [29] and is used in Fluent Ansys to include the effect of surface tension. This  
 152 term can be expressed as follow [30]:

$$153 \quad F_S = 2\sigma \frac{\alpha_l \rho_l K_{c_v} \nabla \alpha_v + \alpha_v \rho_v K_{c_l} \nabla \alpha_l}{\rho_l + \rho_v} \quad [8]$$

154 Where,  $\sigma$  is the interfacial tension between two phases,  $K_{c_l}$  and  $K_{c_v}$  are surface curvatures of liquid and  
 155 vapour respectively that can be written in the following forms:



156  $kc_l = \frac{\Delta\alpha_l}{|\nabla\alpha_l|}$  [9],  $kc_v = \frac{\Delta\alpha_v}{|\nabla\alpha_v|}$  [10]

157

158 **2.3 Energy Equation:**

159

160  $\frac{\partial}{\partial t}(\rho E) + \nabla \cdot [\vec{u}(\rho E + p)] = -\nabla \cdot (k\nabla T) + S_q$  [11]

161 Where,  $E$  and  $K$  are the internal energy and thermal conductivity which can be computed from Eq. (12)  
 162 and Eq. (13) respectively, again, for mixture only.

163  $k = \alpha_l k_l + \alpha_v k_v$  [12]

164  $E = \frac{\alpha_l \rho_l C_{p_l} + \alpha_v \rho_v C_{p_v}}{\alpha_l \rho_l + \alpha_v \rho_v} (T - T_{sat})$  [13]

165 Where,  $k_l$  and  $k_v$  are the thermal conductivity of liquid and vapour and  $C_{p_l}$  and  $C_{p_v}$  are the specific  
 166 heat of liquid and vapour respectively.  $S_q$ , is the energy source term which can be employed to  
 167 determine the heat transfer during the phase change which is calculated from mass source term  $S_{am}$  and  
 168 the latent heat ( $h_{fg}$ ) as follow:

169  $S_q = S_{am} h_{fg}$  [14]

170 Single momentum equation and energy equation will be solved all over the control volume for both  
 171 fluids. Accordingly, the computed velocity and temperature will be shared between two phases.

172

173 **2.4 Phase Change Equations**

174 In order to model the transport phenomenon inside the thermosiphon represented by mass and heat  
 175 transfer from one phase to another during evaporation and condensation processes, source terms  
 176 proposed by De Schepper et al [22] need to be added to the continuity and energy equations used by the

177 VOF model in Fluent Ansys. As stated previously, a single volume fraction equation will be solved for  
 178 each cell for secondary phase while the volume fraction for the primary phase will be obtained from  
 179 eq.(4). Therefore, to describe the mass transfer related to the evaporation process, two equations are  
 180 needed, one for liquid phase and another for vapour phase as follow:

181

182 Evaporation  $T_{mix} > T_{sat}$

183 Liquid phase:

$$184 \quad S_{\alpha M} = -0.1\alpha_l \rho_l \left| \frac{T_{mix} - T_{sat}}{T_{sat}} \right| \quad [15]$$

185 Vapour phase:

$$186 \quad S_{\alpha M} = 0.1\alpha_l \rho_l \left| \frac{T_{mix} - T_{sat}}{T_{sat}} \right| \quad [16]$$

187 Similar to the evaporation process, two expressions are also required to represent the mass transfer  
 188 during the condensation process. Again, one for liquid and another for vapour as follow:

189 Condensation  $T_{mix} < T_{sat}$

190 Liquide phase:

$$191 \quad S_{\alpha M} = 0.1\alpha_v \rho_v \left| \frac{T_{mix} - T_{sat}}{T_{sat}} \right| \quad [17]$$

192 Vapour phase:

$$193 \quad S_{\alpha M} = -0.1\alpha_v \rho_v \left| \frac{T_{mix} - T_{sat}}{T_{sat}} \right| \quad [18]$$

194 Accordingly, the energy source term  $S_q$  that needs to be added to the energy equation (eq. (11)) to  
 195 represent the amount of heat transfer from one phase to another during the evaporation and  
 196 condensation processes can be determined from eq. (14) as follow:

197 Evaporation

$$198 \quad S_q = -0.1\alpha_l \rho_l \left| \frac{T_{mix} - T_{sat}}{T_{sat}} \right| h_{fg} \quad [19]$$

199 Condensation

$$200 \quad S_q = 0.1\alpha_v \rho_v \left| \frac{T_{mix} - T_{sat}}{T_{sat}} \right| h_{fg} \quad [20]$$

201 Where,  $T_{mix}$  and  $T_{sat}$  are the temperature of mixture and saturation temperature respectively. Equations  
202 (15-20) are set in a sub-program and linked to the Fluent to add the calculated mass source terms  
203 (eqs.15-18) and energy source terms (eqs.19 and 20) to the mass conservation equation (3) and energy  
204 equation (11) respectively in the VOF model in order to completely model the phase change process.

205

### 206 **3. CFD SIMULATION SET UP**

#### 207 **3.1 Geometry and Mesh**

208 Geometry of a vertical two-dimension wickless heat pipe has been generated using workbench design  
209 modular (Ansys 15). The geometry represents a copper tube with a total height of 400 mm, outer and  
210 inner diameters of 22 and 20.2 mm respectively. The thermosiphon is divided into two sections,  
211 evaporator and condenser with height of 200 mm each as illustrated in Fig. 1. These dimensions are  
212 chosen to be similar to geometry of a previous experimental work by Abdullahi [20] to validate the  
213 CFD simulation.

214 Workbench design modular (Ansys 15) was also used to mesh the geometry where Control edge sizing  
215 technique was employed to control the grid in every domain and to govern cell sizes near inner walls  
216 and inside the solid domain (walls) with bias factor of 10 used in these regions to ensure that the flow  
217 and heat transfer can be correctly captured in these areas. The number of cells in the fluid domain was  
218 24522 and 9620 grids in the solid domain. The mesh size and type are shown in Fig. 2.a.

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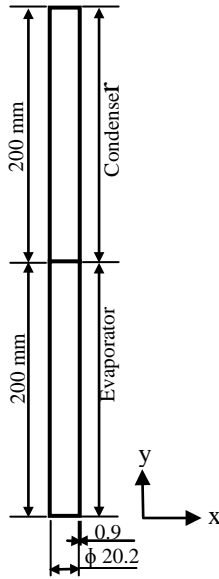


Fig.1. Heat pipe geometry

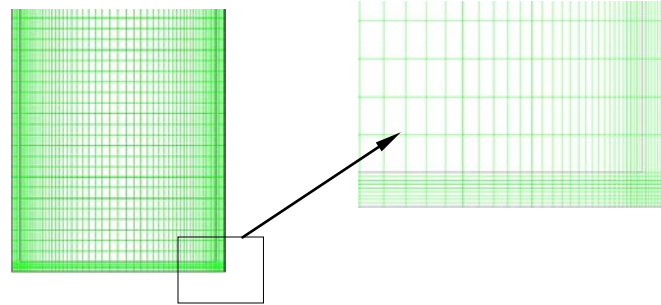


Fig.2.a. Mesh of zoomed section

### 3.2 Initial and Boundary Conditions

Five different filling ratios and inclination angles are used in this study namely, 25%, 35%, 65%, 80% and 100% of the evaporator volume, and 10, 30, 50, 70 and 90°, respectively. To set up the fill ratio for each case, the corresponding evaporator height is initially patched with liquid while the remaining height is patched with vapour. In addition, the inclination angle is defined as the inclination of thermosiphon from the horizontal axis and can be set up by multiplying y-component of acceleration gravity with sine of the angle and x-component with cosine as shown in fig.2.b.

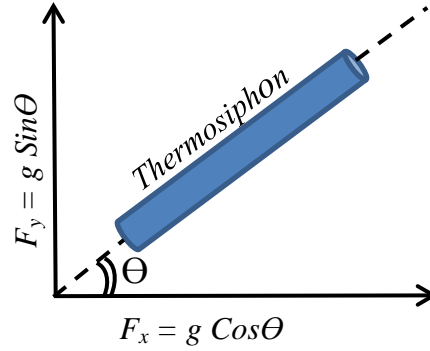


Fig.2.b. Inclination angle of thermosiphon heat pipe

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243 The initial temperature of both evaporator wall and liquid should be selected slightly above the boiling  
 244 point which was chosen to be 373 °K to insure that the boiling process occurs once simulation time  
 245 starts to reduce computational time [30] and the condenser wall and fluid temperatures were set as 290  
 246 °K (condenser cooling temperature). Operating temperature should be set to be the smallest temperature  
 247 in the system (290 °K) and operating density must be set as 0 Kg/m<sup>3</sup> when ideal gas is used and as the  
 248 smallest density in the system when constant gas density is used [30]. In addition, saturation  
 249 temperature and operating pressure were set to be 373 °K and 101325 Pa, respectively.

250 At the internal walls of evaporator and condenser sections, a non-slip boundary condition is applied,  
 251 while a constant heat flux is imposed at the outer wall of the evaporator to simulate the heat added to  
 252 the thermosiphon. Three values of heat flux were employed 2858, 5910 and 7346 W/m<sup>2</sup> corresponding  
 253 to heat transfer rates 39, 81 and 101W respectively, which is taken from [20]. The top and the bottom  
 254 ends of the thermosiphon is assumed to be insulated, which means no cooling or heating effect applied  
 255 at these walls. As a result, a zero heat flux is defined at these ends. To model the heat removed from the  
 256 condenser section, a convection boundary condition is applied at the outer wall of the condenser  
 257 section. Thus, the heat transfer coefficient between cooling water and the condenser's wall needs to be  
 258 calculated from the following relation:

259

$$h_{conv} = \frac{Q_{cond}}{2\pi DL_{cond}(T_{cw,av} - T_m)} \quad [21]$$

260 Where,  $h_{conv}$  is the convection heat transfer coefficient between the cooling water and the condenser's  
261 wall,  $Q_{cond}$  is the heat removed from the condenser section,  $T_{cw,av}$  is the average wall temperature of the  
262 condenser section and  $T_m$  is the mean temperature of the cooling water. Values of  $Q_{cond}$ , and  $T_m$  are  
263 obtained from Abdallahi [20] experimental work.

264

265 To include the effect of the interfacial force between liquid and vapour, the term  $F_s$  is added to the  
266 momentum equation eq. (5) by activating the CSF in the fluent. Consequently, the value of the surface  
267 tension in eq. (6) can be computed from the following formula [16]:

$$268 \quad \sigma = 0.09805856 - 1.845 \times 10^{-5}T - 2.3 \times 10^{-7}T^2 \quad [22]$$

269

### 270 **3.3 Solution Methods and Techniques**

271 In present analysis, the VOF model is used to simulate the multi-phase flow, while the gravitational  
272 acceleration of  $9.81 \text{ m/s}^2$  is activated to include a body force term. The water liquid is chosen to be a  
273 secondary phase (liquid phase) and its density can be determined from the following relation [16]:

$$274 \quad \rho_l = 859.0083 + 1.252209T - 0.0026429T^2 \quad [23]$$

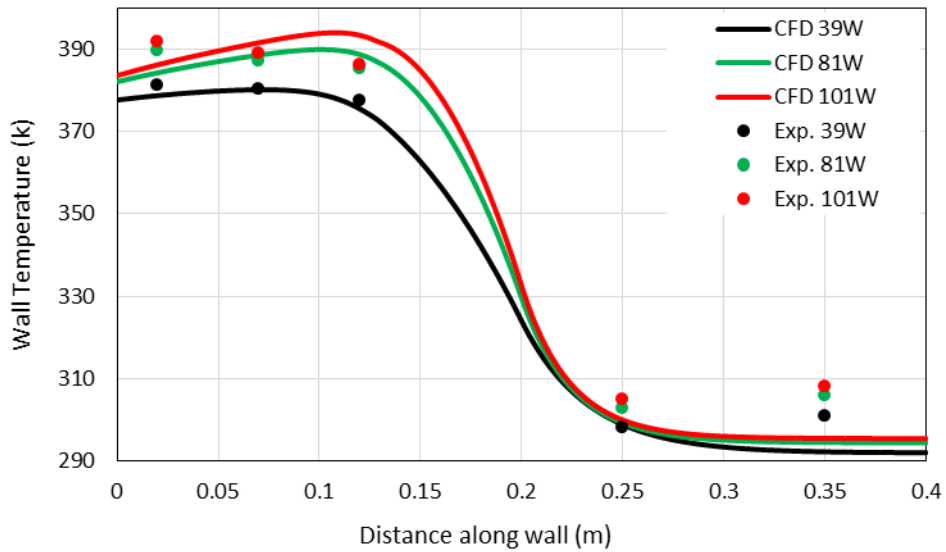
275 A transient solution with a time step of 0.001s is employed for all cases due to dynamic behaviour of  
276 the two-phase flow [17, 22]. A combination of the SIMPLE algorithm for pressure-velocity coupling  
277 and first-order upwind scheme for the calculation of the momentum and energy are used. For  
278 determination of the volume fraction and pressure, Geo-Reconstruct and PRESTO discretisation are  
279 chosen, respectively [16, 17]. The solution is considered to be converged when the residuals of the  
280 mass and velocity components are reduced to  $10^{-4}$  while the residuals of the temperature variables are  
281 reduced to  $10^{-6}$ .

## 282 **4. RESULTS AND DISCUSSION**

283 **4.1 Validation of the CFD Solution**

284 To validate the CFD simulation, same geometry and boundary conditions as Abdullahi [20] have been  
285 adopted. Therefore, the temperature distribution along the wall and the thermal resistance of the  
286 thermosiphon for the stated three different heat inputs which are determined from CFD modelling have  
287 been compared with those obtained from Abdullahi [20] experimental work.

288 A comparison of the temperature distribution along thermosiphon wall between the CFD modelling  
289 (current work) and the experimental work [20] is illustrated in fig.3 for three input energies. It is shown  
290 that the CFD simulation (solid lines) predicts well the experimental results (marks). However, there is a  
291 slight deviation (maximum 4.2%) at the bottom of the evaporator and the top of the condenser where  
292 the difference becomes larger at larger heat input.



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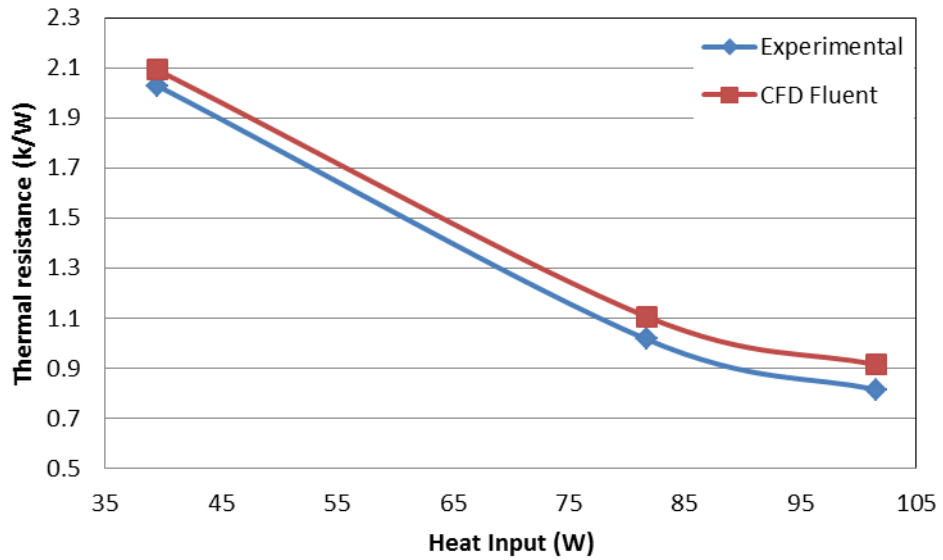
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295 Fig.3. comparison of Variation of temperature along the wall of thermosiphon between experimental  
296 data and CFD results (Vertical orientation)

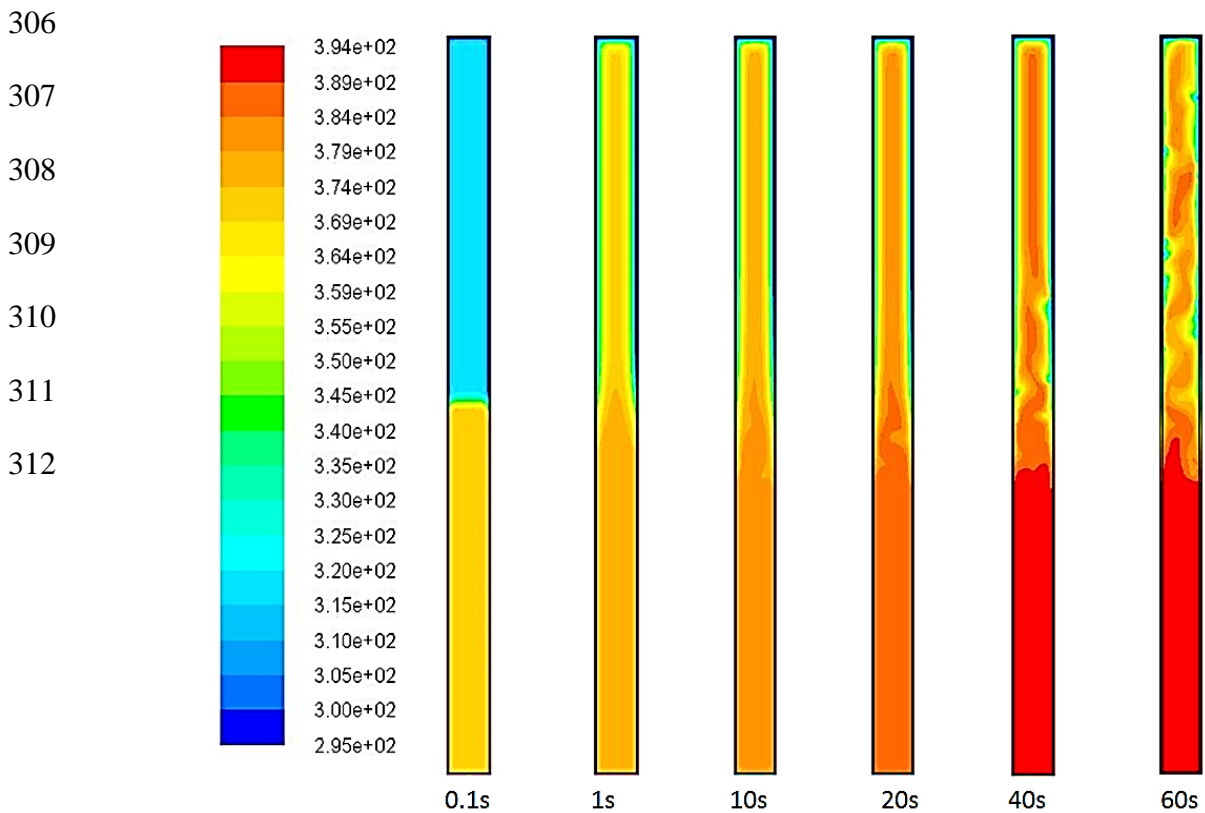
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298 Figure (4) presents a comparison of the thermal resistance between CFD simulation and experimental  
299 study [20] at different heat inputs. It is observed that the CFD solution over predicts the experimental

300 results by 8.1%. This is due to higher evaporator temperature and lower condenser temperature  
 301 obtained from the CFD solution, which yield higher thermal resistance. However, the same trend has  
 302 been achieved in which the thermal resistance decreases with increasing the heat input.



303  
 304 Fig.4. Comparison of Variation of the thermal resistance with heat input between experimental data and  
 305 CFD results (Vertical orientation)





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Fig.5. Temperature contours at various simulation times (101W, 65 % and 90°)

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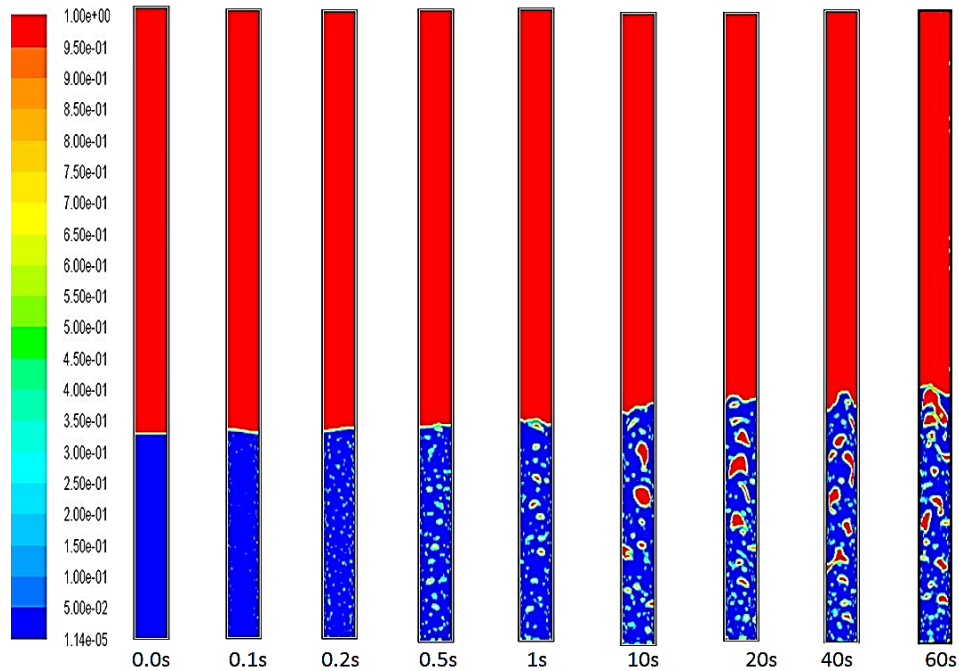
T Figure (5) shows the heat transfer process represented by temperature contours during simulation time at heat input 101W, fill ratio 65% and vertical orientation. Firstly, heat transfer from evaporator wall to the liquid due to constant heat flux, then, when the working fluid reaches its saturation temperature, it starts boiling and the phase change occurs. Therefore, vapour raises up to heat the upper part of the heat pipe and the temperature increases accordingly with time until reaching the steady state.

319

320

The variation of the vapour volume fraction with simulation time is illustrated in figure (6) in which the red colour refers to vapour phase (volume fraction=1) and the blue one refers to liquid phase (volume fraction=0). At the beginning, a very small bubble size is observed at time 0.1 second, then, bubbles size and number increase as simulation time increases due to increase in the temperature of the liquid reaching the boiling temperature and, hence, the steady state condition at time 60 seconds.

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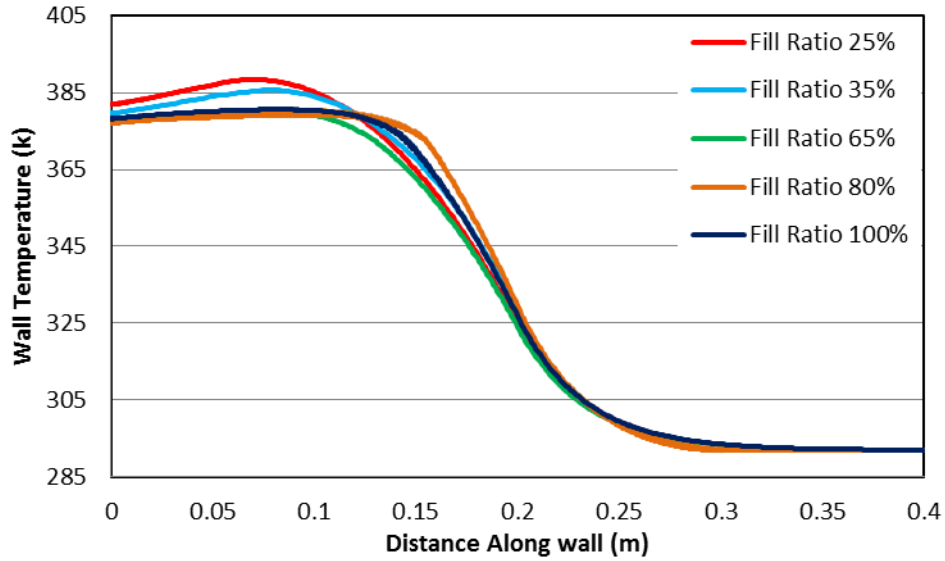
332

Fig.6. Vapour volume fraction contours at various simulation times (101W, 65 % and 90°)

333

## 334 **4.2 Fill Ratio Effect**

335 The influence of the volume of the charged liquid on the thermal performance of the (TPCT) is  
336 obtained by employing the CFD simulation. Therefore, the temperature distribution on the outer wall of  
337 the thermosiphon for fill ratios 25%, 35%, 65%, 80% and 100% is shown in figures (7.a), (7.b) and  
338 (7.c) at heat inputs of 39, 81 and 101W respectively. Figures (7.a, b and c) show similar trends in  
339 temperature distribution along the wall of thermosiphon at three heat inputs for each fill ratio. It is also  
340 observed that the effect of changing fill ratio and increasing heat input on temperature profile is more  
341 significant in the evaporator section than in condenser section. In addition, a lowest wall temperature  
342 distribution is seen at fill ratio 65% for all input energies. On the other hand, a high wall temperature  
343 occurs at the mid-distance of the evaporator wall at fill ratio 25% and 35% for all heat inputs. This wall  
344 temperature increases with increasing the heat input until reaching the highest value at heat input 101W  
345 and fill ratio 25%. For fill ratios 80% and 100%, a higher wall temperature in upper part of the  
346 evaporator is observed compared with other values of fill ratio for three heat inputs. This is due to  
347 higher liquid height in the evaporator which prevents large bubbles to reach liquid surface forming a  
348 vapour film on the inner wall of the evaporator and hence, increasing the wall evaporator temperature  
349 in that region. The effect of higher liquid height decreases with increasing the heat input in the case of  
350 80% fill ratio whereas increases in the case of 100%.

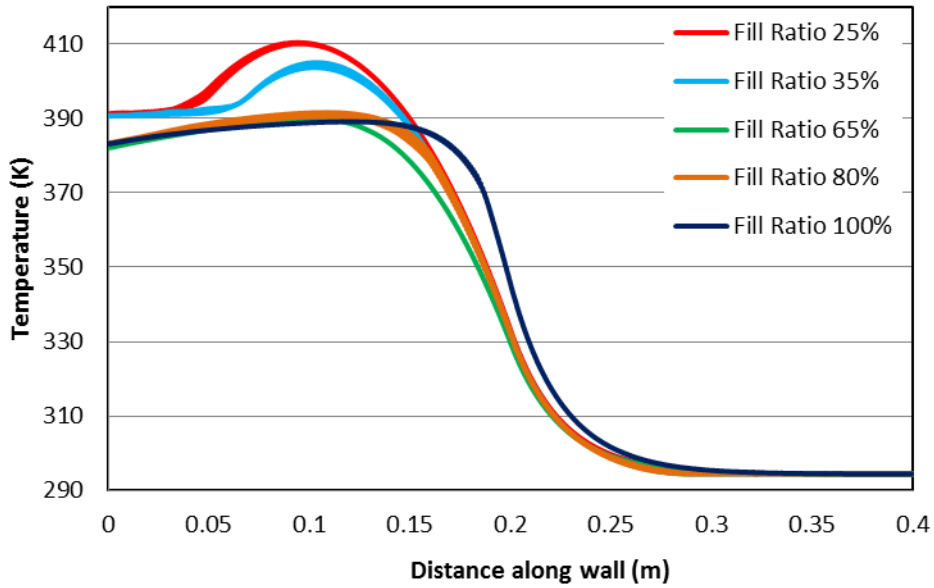


351

352 Fig.7.a Variation of temperature with the distance along the wall of the thermosiphon at heat input 39

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W for different fill ratios (Vertical orientation)



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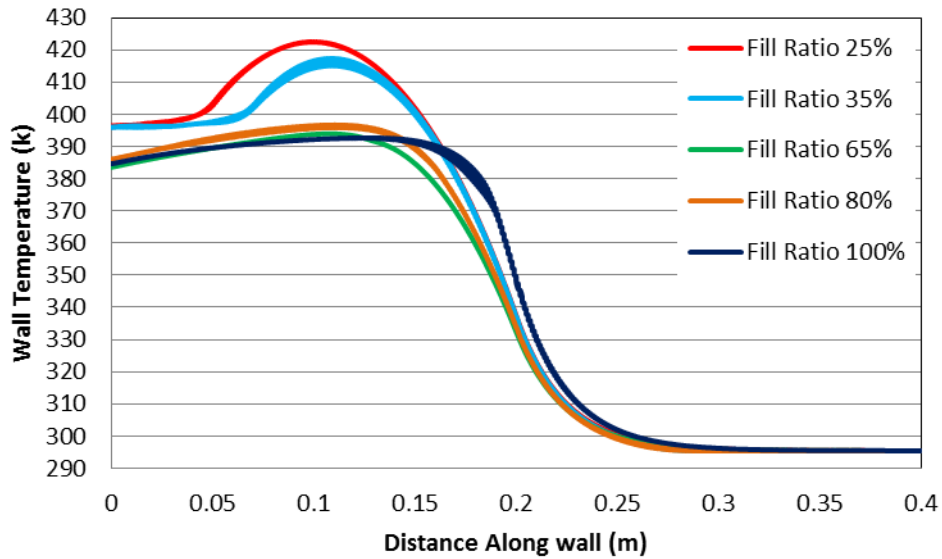
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356 Fig.7.b Variation of temperature with the distance along the wall of the thermosiphon at heat input 81

357

W for different fill ratios (Vertical orientation)

358



359

360 Fig.7.c Variation of temperature with the distance along the wall of the thermosiphon at heat input 101  
 361 W for different fill ratios (Vertical orientation)

362

363 Figure (8) presents the effect of the fill ratio on the average wall temperature of the evaporator for three  
 364 heat inputs. It is shown that the average evaporator wall temperature decreases from its maximum value  
 365 at fill ratio 25% to the minimum value at 65% then increases again to a certain value at fill ratios 80%  
 366 and 100% for input energies 81 and 101 W (similar trend was obtained by [9]). However, at heat input  
 367 39 W, there is a slight change in evaporator wall temperature between fill ratios 25% and 35% and after  
 368 fill ratio 80% the trend decreases slightly at fill ratio 100%. Therefore, the effect of fill ratio on  
 369 evaporator wall temperature is more clear at relatively high input energy (81 and 101 W) than that at low  
 370 energy (39 W).

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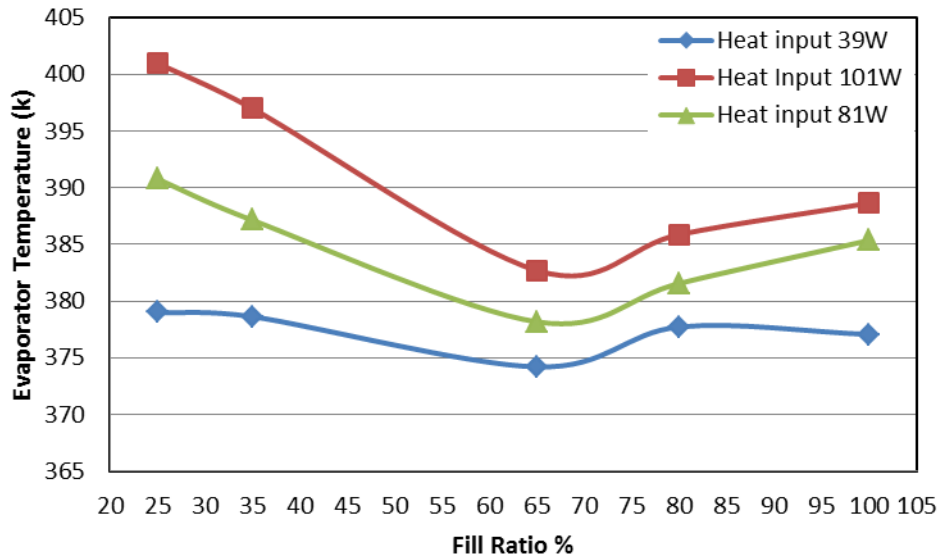


Fig.8. Variation of average wall temperature of evaporator with fill ratio at different heat inputs  
(Vertical orientation)

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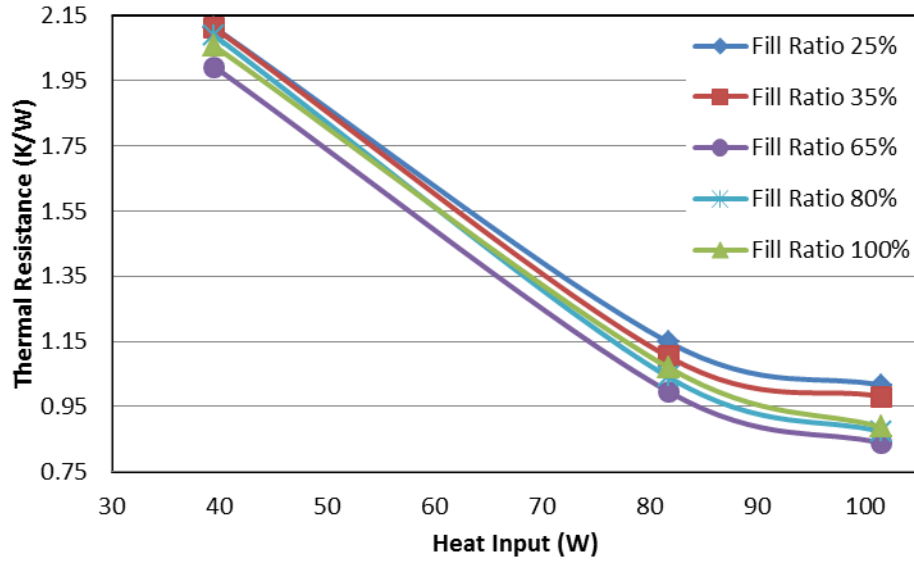
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Figure (9) shows the effect of heat input on thermal resistance for various fill ratios. It is seen that the thermal resistance decreases with increasing heat input for all fill ratios. A higher thermal resistance is observed at fill ratio 25% due to a small amount of working fluid whereas a lower value at 65% for all energy inputs (similar trend was obtained by [10]). However, a lower difference in thermal resistance between the fill ratios is seen at heat input of (39W), especially, between 25% and 35% compared with that at higher energy inputs (81 and 101W). This indicates that with low fill ratios and a heat input of 101W, the heat pipe reaches its heat transfer limit leading to high temperatures at the upper part of the evaporator as shown in figures 7.b and 7.c. In addition, the thermal resistance for fill ratio 80% is greater than that for 100% at input energy 39W compared with that at higher heat inputs (81 and 101W). Thus, the best fill ratio is 65% and this is a similar conclusion as those were concluded by [17] and [10]. The reason behind increasing the evaporator wall temperature and, hence, the thermal resistance at high fill ratios (80% and 100%) attribute to the fact that the thermal resistance of liquid film in the evaporator increases as liquid height increase (fill ratio) above the optimum value.



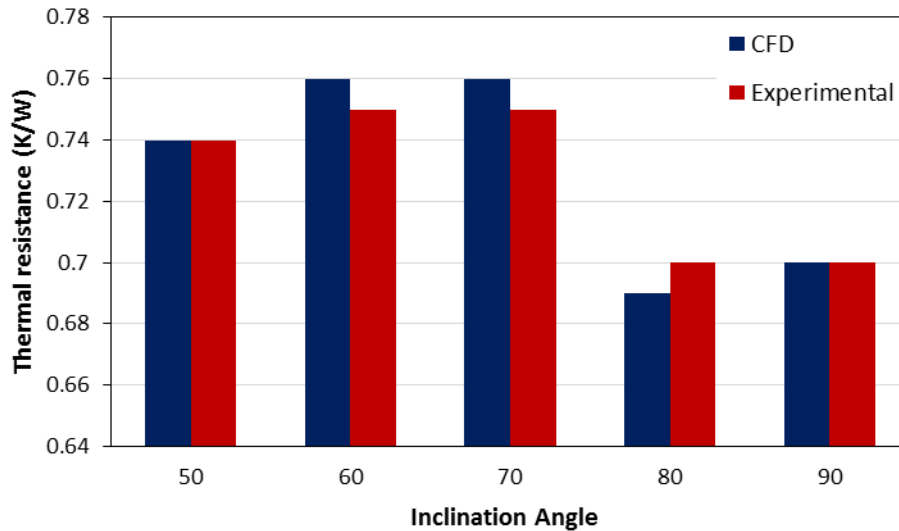
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389 Fig.9. Variation of thermal resistance with heat inputs at different fill ratios (Vertical orientation)

390 **4.3 Effect of Inclination Angle**

391 CFD simulation has been used to investigate the effect of inclination angle on the thermal performance  
 392 of the thermosiphon at angles of (10, 30, 50, 70 and 90°). Firstly, the numerical results were compared  
 393 with the experimental work of Abdullahi [20] in terms of thermal resistance to validate the CFD  
 394 solution. Fig.10 presents a comparison of variation of thermal resistance with inclination angle of  
 395 thermosiphon at heat input 109W between CFD modelling and experimental work [20]. CFD results  
 396 show a good agreement with experimental data with maximum deviation of (1.3%) and produce a  
 397 similar trend in which the lowest thermal resistance is obtained at angles of (80 and 90°) whereas the  
 398 highest at (70°).

399



400

401 Fig.10. Comparison of variation of thermal resistance with inclination angle between CFD result and  
 402 experimental work (109W and FR=65%)

403

404 Figure (11.a) presents the variation of vapour volume fraction during flow time for inclination angle of  
 405  $10^\circ$ , heat input 101W and fill ratio 65%. It is clear that the liquid in evaporator is not in contact at  
 406 certain parts of evaporator wall due to inclination leading to increase the wall evaporator temperature.  
 407 In addition, it is observed that the bubble size remains relatively small as time increases and this may  
 408 be attributed to the nearness of liquid surface to the bubble nucleation sites because of the inclination.  
 409 As a result, a vapour film forms on the upper part of the evaporator wall which leads to additional  
 410 increase in evaporator wall temperature. Fig.11.b shows the vapour volume fraction at simulation times  
 411 3 and 60 seconds for different fill ratios. Relatively small bubbles are observed for fill ratios 25% and  
 412 35% due to nearness of liquid surface from bubble sites. On the other hand, for fill ratios 80% and  
 413 100%, many large bubbles stuck on evaporator wall before they reach liquid surface due to high height  
 414 of liquid column resulting in higher evaporator temperature compared with fill ratio 65%.  
 415 Bubble dynamics and frequency can be greatly changed by changing surface wettability in terms of  
 416 contact angle [31]. This also depends on the type of fluid used where the contact angle is a function of

417 surface tension which changes from one fluid to another. Therefore, investigating of such point would  
418 be important to study the effect of these parameters on the thermal performance of thermosiphon heat  
419 pipe in future work.

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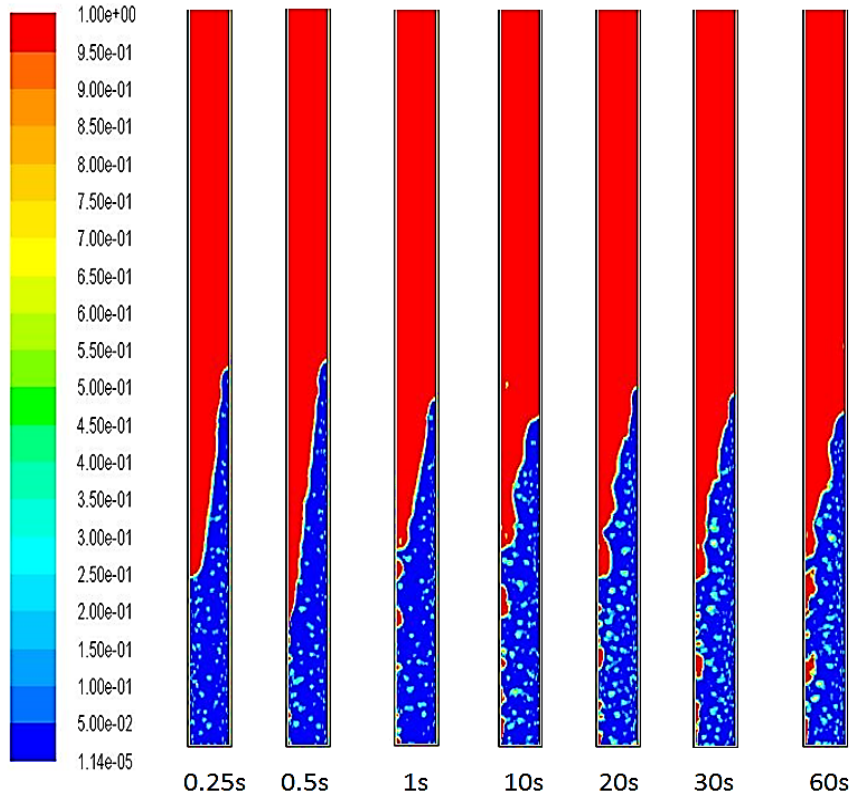
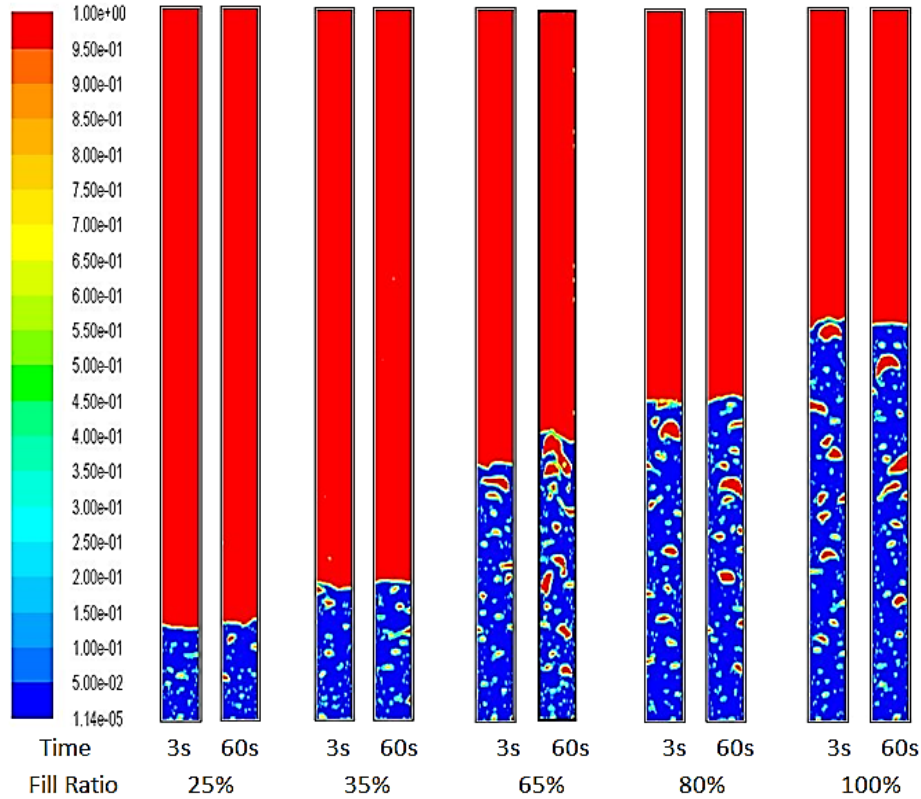


Fig.11.a. Vapour volume fraction contours at various simulation times for inclination angle  $10^\circ$





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Fig.11.b. Vapour volume fraction contours at various simulation times for different fill ratios

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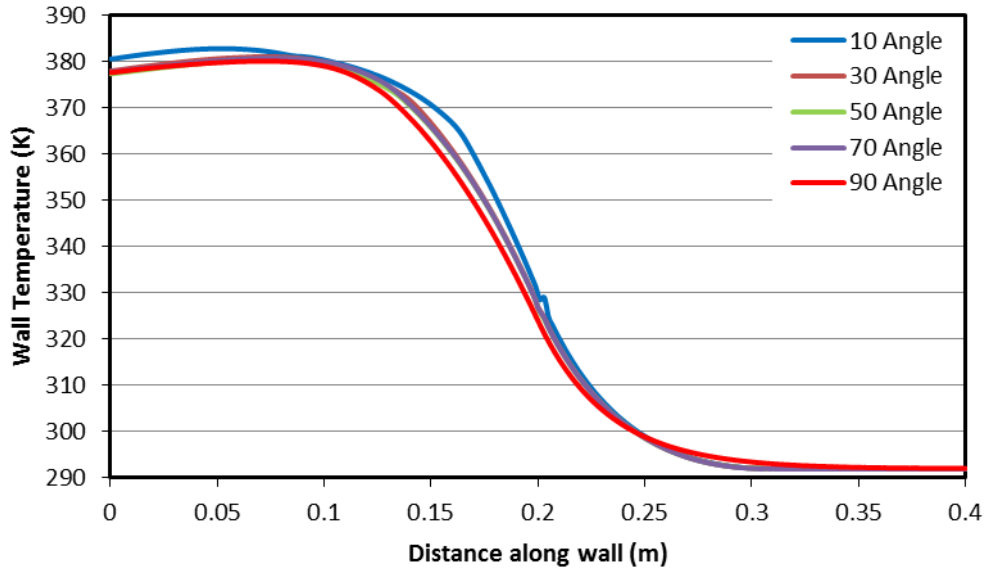
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Figures (12.a, 12.b and 12.c) illustrate the variation of the wall temperature of thermosiphon with the distance along the wall for three heat inputs (39, 81 and 101W) at five inclination angles (10, 30, 50, 70 and 90) and fill ratio of 65%. They show a similar trend for three input energies in which the highest and lowest wall temperature occur at angles of 10° and 90°, respectively. These higher temperatures at low inclination angles attribute to the fact that some of the upper part of the evaporator section is not in contact with liquid due to inclination. However, at the inclination angle of 10o and heat input 39 W, the wall temperature near 0.2 m (at the beginning of the condenser section) remains constant for a short distance and then decreases. This can be attributed to the existence of liquid at the lower part of the condenser as a result of inclination near the horizontal orientation (10 degree) leading to blockage of this part which prevents the temperature to decrease, after that, the wall temperature starts decreasing

448 again. This effect decreases as heat input increases (81W) due to increasing the evaporation rate which  
449 reduces the amount of liquid at that region allowing the wall temperature to decrease. It is also  
450 observed that the effect of inclination angle increases as the heat input increases.

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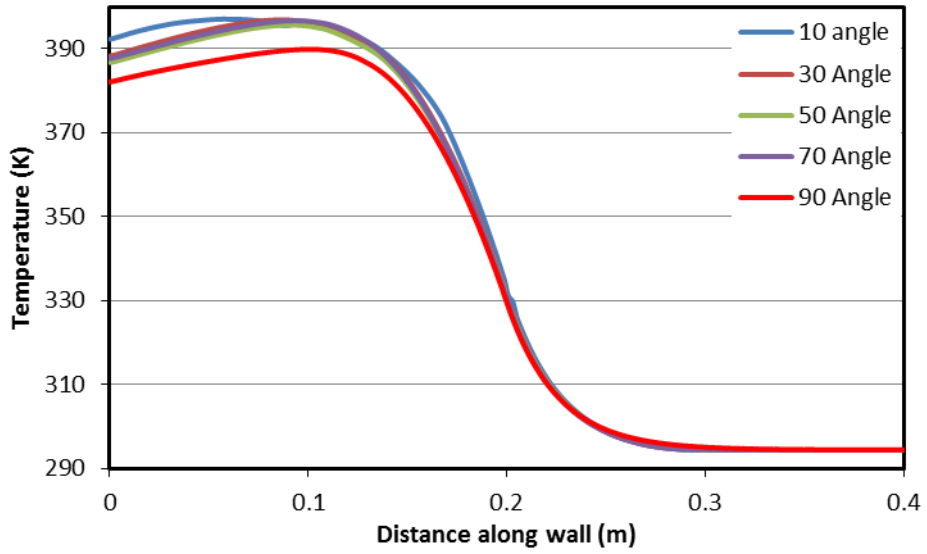
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454 Fig.12.a. Wall temperature distribution at heat input 39W and fill ratio 65% for different inclination  
455 angles

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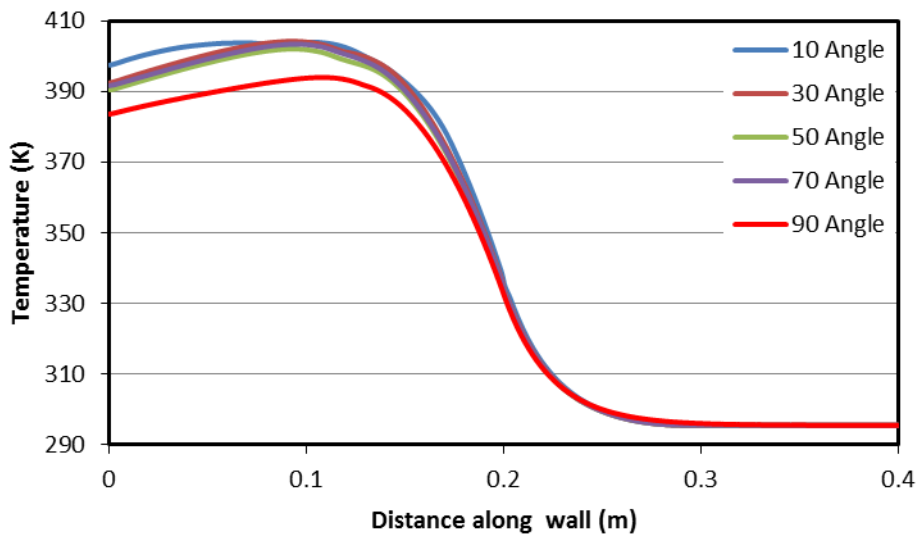
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459 Fig.12.b. Wall temperature distribution at heat input 81W and fill ratio 65% for different inclination

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angles

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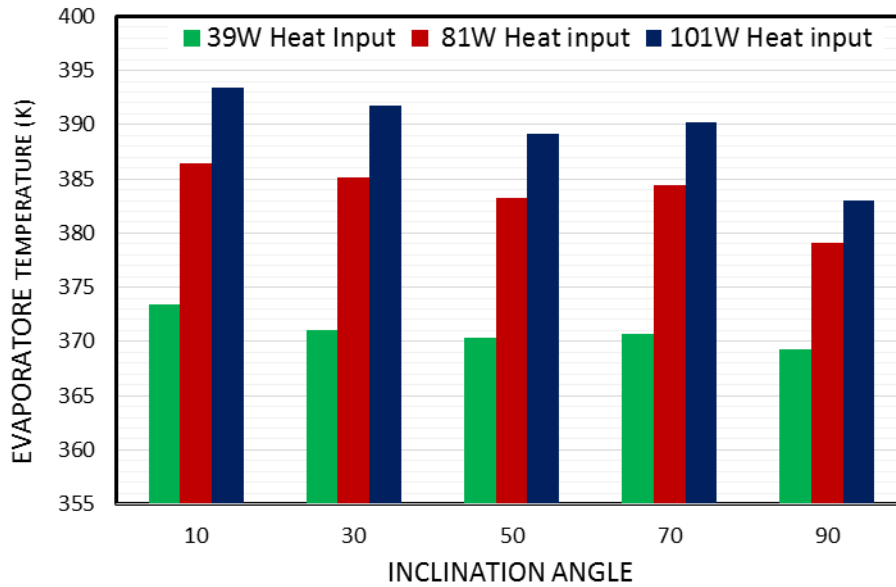
463 Fig.12.c. Wall temperature distribution at heat input 101W and fill ratio 65% for different inclination

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angles

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466 The effect of inclination angle on the average wall temperature of the evaporator at input energies of  
467 39, 81, and 101W is illustrated in figure 13. It can be seen that the evaporator temperature increases as  
468 the inclination angle decrease toward the horizontal orientation for all heat inputs and this increase is  
469 higher when the heat input is higher. However, at angle of  $50^\circ$  the value of the evaporator temperature  
470 is less than that at angle  $70^\circ$  for all three cases, but it is still higher than the value at angle  $90^\circ$ .  
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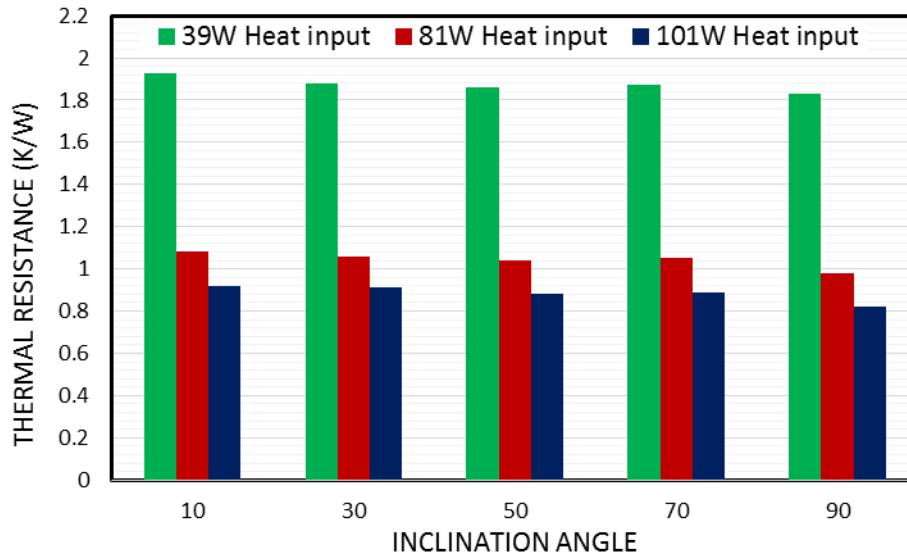


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473

474 Fig.13. variation of the evaporator wall temperature with inclination angle at heat inputs 39, 81, and  
475 101W (FR=65%)

476 Figure 14 shows the effect of inclination angle on the thermal resistance of the thermosiphon at heat  
477 inputs 39, 81, and 101W. The results show that the thermal resistance decreases as the inclination angle  
478 increases and the highest and lowest thermal resistance are at inclination angle  $10^\circ$  and  $90^\circ$ ,  
479 respectively, for all input energies. Therefore, the thermal performance of the (TPCT) is better at  
480 vertical orientation ( $90^\circ$ ) than that at other orientations (similar conclusions were reported by [10] and  
481 [15]).



482

483 Fig.14. variation of the thermal resistance with inclination angle at heat inputs 39, 81, and 101W (FR=  
484 65%)

485

## 486 5. CONCLUSIONS

487 The effect of five fill ratios of working fluid (25%, 35%, 65%, 80% and 100% of the evaporator  
488 volume) and five inclination angles (10, 30, 50, 70 and 90°) on the performance of the tow phase closed  
489 thermosiphon was investigated numerically by developing a new CFD simulation. A comparison  
490 between the CFD solution and a published experimental work was also carried out for different heat  
491 inputs 39, 81 and 101W and at fill ratio of 65%. It is concluded that:

492

493 1- Developed CFD simulation was successfully used to model the TPCT and investigate the effect of  
494 fill ratio and inclination angle on its thermal performance. This proved by comparing the wall  
495 temperature distribution and thermal resistance for three input energies at fill ratio 65% with published  
496 experimental data, and maximum deviations of 4.2% and 8.1% has been reported, respectively.

497 Regarding to inclination angle, a comparison in terms of thermal resistance for inclination angles of 50,

498 60, 70, 80 and 90° at heat input 109W and fill ratio 65% has been carried out with a maximum  
499 deviation of 1.3%.

500 2- Heat transfer limit is reached when the volume of the charged liquid is small at fill charge ratio of  
501 25% and 35%. This is observed when a considerable increase in evaporator wall temperature takes  
502 place, especially at higher energy input.

503 3- The lowest average evaporator wall temperature and thermal resistance take place at fill ratio of 65%  
504 and angle of 90° whereas the highest at 25% and 10° due to the effect of small fill ratio and inclination,  
505 respectively. This effect is higher as heat input increases.

506 4- The best fill ratio and inclination angle regarding to the thermal performance for this case were  
507 found to be 65% and 90°, respectively.

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#### 511 **REFERENCES**

- 512 [1] A. Faghri, "Heat pipe science and technology," Global Digital Press, 2015.  
513 [2] D. R. Reay, McGlen, and P. Kew, "Heat pipes: Theory, design and applications," Butterworth-  
514 Heinemann, 2014.  
515 [3] G. P. Peterson, "An Introduction to Heat Pipes Modelling, Testing, and Applications", John Wiley  
516 & Sons, Inc., 1994.  
517 [4] S. H. Noie, "Heat transfer characteristics of a two-phase closed thermosiphon," Appl. Therm. Eng.,  
518 vol. 25, no. 4, pp. 495–506, 2005.  
519 [5] B. Jiao et al, "Investigation on the effect of filling ratio on the steady-state heat transfer  
520 performance of a vertical two-phase closed thermosiphon," Appl. Therm. Eng., vol. 28, no. 11–12, pp.  
521 1417–1426, 2008.  
522 [6] H. Jouhara and A. J. Robinson, "Experimental investigation of small diameter two-phase closed  
523 thermosiphons charged with water, FC-84, FC-77 and FC-3283," Appl. Therm. Eng., vol. 30, no. 2–3,  
524 pp. 201–211, 2010.  
525 [7] A. K. Mozumder et al, "Performance of Heat Pipe for Different working fluids and fill ratios," J.  
526 Mech. Eng., vol. 41, no. 2, pp. 96–102, 2010.  
527 [8] T. Sukchana and C. Jaiboonma, "Effect of Filling Ratios and Adiabatic Length on Thermal  
528 Efficiency of Long Heat Pipe Filled with R-134a," Energy Procedia, vol. 34, pp. 298–306, 2013.  
529 [9] A. A. Chehade et al, and N. Abouzahab-Damaj, "Experimental investigation of thermosiphon loop  
530 thermal performance," Energy Convers. Manag., vol. 84, pp. 671–680, 2014.

- 531 [10] R. Manimaran et al, "An Investigation of Thermal Performance of Heat Pipe Using Di-water," *Sci.*  
532 *Technol.*, vol. 2, no. 4, pp. 77–80, 2012.
- 533 [11] E. Sadeghinezhad et al, "Experimental investigation of the effect of graphene nanofluids on heat  
534 pipe thermal performance," *Appl. Therm. Eng.*, vol. 100, pp. 775–787, 2016.
- 535 [12] M. Ghanbarpour et al, "Thermal performance of inclined screen mesh heat pipes using silver  
536 nanofluids," *Int. Commun. Heat Mass Transf.*, vol. 67, pp. 14–20, 2015.
- 537 [13] Z. Xue and W. Qu, "Experimental study on effect of inclination angles to ammonia pulsating heat  
538 pipe," *Chinese J. Aeronaut.*, vol. 27, no. 5, pp. 1122–1127, 2014.
- 539 [14] R. Senthilkumar et al, "Effect of inclination angle in heat pipe performance using copper  
540 nanofluid," *Procedia Eng.*, vol. 38, pp. 3715–3721, 2012.
- 541 [15] M. Nazarimanesh et al, "Experimental study on the effects of inclination situation of the sintered  
542 heat pipe on its thermal performance," *Exp. Therm. Fluid Sci.*, vol. 68, pp. 625–633, 2015.
- 543 [16] B. Fadhl et al, "Numerical modelling of the temperature distribution in a two-phase closed  
544 thermosiphon," *Appl. Therm. Eng.*, vol. 60, no. 1–2, pp. 122–131, 2013.
- 545 [17] A. Alizadehdakhel et al, "CFD modelling of flow and heat transfer in a thermosiphon," *Int.*  
546 *Commun. Heat Mass Transf.*, vol. 37, no. 3, pp. 312–318, 2010.
- 547 [18] G. Huminic and A. Huminic, "Numerical study on heat transfer characteristics of thermosyphon  
548 heat pipes using nanofluids," *Energy Convers. Manag.*, vol. 76, pp. 393–399, 2013.
- 549 [19] B. Fadhl et al, "CFD modelling of a two-phase closed thermosiphon charged with R134a and  
550 R404a," *Appl. Therm. Eng.*, vol. 78, pp. 482–490, 2015.
- 551 [20] B. Abdullahi, "Development and Optimization of heat pipe based Compound Parabolic Collector,"  
552 Thesis submitted to University of Birmingham, 2015.
- 553 [21] Y. Kim et al, "Effects of mass transfer time relaxation parameters on condensation in a  
554 thermosyphon," *J. Mech. Sci. Technol.*, vol. 29, no. 12, pp. 5497–5505, 2015.
- 555 [22] S. C. K. De Schepper, G. J. Heynderickx, and G. B. Marin, "Modeling the evaporation of a  
556 hydrocarbon feedstock in the convection section of a steam cracker," *Comput. Chem. Eng.*, vol.  
557 33, no. 1, pp. 122–132, 2009.
- 558 [23] Z. Yang et al, "Numerical and experimental investigation of two phase flow during boiling in a  
559 coiled tube," *Int. J. Heat Mass Transf.*, vol. 51, no. 5–6, pp. 1003–1016, 2008.
- 560 [24] C. Fang et al, "Volume of Fluid Simulation of Boiling Two-Phase Flow in a Vapour-Venting  
561 Microchannel," *Front. Heat Mass Transf.*, vol. 1, no. 1, 2010.
- 562 [25] S. C. K. De Schepper et al, "CFD modelling of all gas-liquid and vapour-liquid flow regimes  
563 predicted by the Baker chart," *Chem. Eng. J.*, vol. 138, no. 1–3, pp. 349–357, 2008.
- 564 [26] H. Ganapathy et al, "Volume of fluid-based numerical modeling of condensation heat transfer and  
565 fluid flow characteristics in microchannels," *Int. J. Heat Mass Transf.*, vol. 65, pp. 62–72, 2013.
- 566 [27] S. Ghorai and K. D. P. Nigam, "CFD modeling of flow profiles and interfacial phenomena in two-  
567 phase flow in pipes," *Chem. Eng. Process. Process Intensif.*, vol. 45, no. 1, pp. 55–65, 2006.
- 568 [28] ANSYS FLUENT, Theory Guide (Release 15.0). "Multiphase Flows," ANSYS, Inc.,  
569 November 2013, pp. 465–600 (chapter 17).
- 570 [29] J. Brackbill, D. Kothe, and C. Zemach, "A continuum method for modelling surface tension," *J.*  
571 *Comput. Phys.*, vol. 100, pp. 335–354, 1992.
- 572 [30] ANSYS FLUENT, User's Guide (Release 15.0). "Modelling Multiphase Flows," ANSYS, Inc.,  
573 November 2013, pp. 1243–1387 (chapter 25).
- 574 [31] R. Ahmadi and T. Okawa, "Influence of surface wettability on bubble behaviour and void  
575 evolution in subcooled flow boiling," *International Journal of Thermal Sciences.*, vol. 97, pp. 114–125,  
576 2015.
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- 578