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Alammar, Ahmed A.; Al-mousawi, Fadhel N.; Al-dadah, Raya K.; Mahmoud, Saad M.; Hood, Richard

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Enhancing Thermal performance of a Two-phase Closed Thermosyphon With an Internal Surface Roughness

Ahmed A. Alammar ^{a,b}, Fadhel N. Al-Mousawi ^a, Raya K. Al-Dadah ^a, Saad M. Mahmoud ^a, Richard Hood ^a
 ^a Department of Mechanical Engineering, University of Birmingham
 Edgbaston, Birmingham, B15 2TT, <u>Ghulfus_eng@yahoo.com</u>
 ^b Training and Development Office, Ministry of Electricity, Baghdad, Iraq

7 8

9 Abstract

Enhancement of energy conversion devices has become an important task to reduce size and cost, and 10 11 design efficient systems. In this work, enhancement of heat transfer performance of a two-phase closed 12 thermosyphon has been investigated by making an internal surface roughness. Thus, a new advanced machining technique (Electrical Discharge Machining) is employed to modify the surface 13 14 characteristics of a TPCT. The experimental work has been carried out at two initial sub-atmospheric pressures (3 and 30 kPa), heat input range of (90-160 W) and a fill ratio of 50% using water as a working 15 fluid. The results of the new thermosyphon have been compared with a plain copper TPCT to consider 16 the enhancement in thermal performance resulting from resurfacing of the thermosyphon wall. The 17 18 results revealed that using internal wall roughness in TPCT can enhance its thermal performance by reducing the evaporator temperature, thereby the total thermal resistance decreasing by about 42% and 19 13% at initial pressures of 3 kPa and 30 kPa, respectively. On the other hand, the evaporator thermal 20 resistance decreases and the evaporator heat transfer coefficient increases by about 115% and 68% at 21 initial pressures of 3 kPa and 30 kPa, respectively. However, the condenser thermal performance 22 23 decreases using the resurfaced TPCT compared with plain thermosyphon.

Keywords: Two-phase closed thermosyphon; Surface roughness; Thermal performance enhancement;
Thermal resistance.

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NOMENCLATURE

D	Diameter of thermosyphon	m	Subscripts	
h	Heat transfer coefficient	W/m ² °C	av	Average
Ι	Current	Am	С	Condenser
Κ	Thermal conductivity	W/m °C	е	Evaporator
Q	heat input	W	i	Inside
\tilde{R}	Thermal resistance	°C/W	0	Outside
Т	Temperature	°C	t	total
V	Voltage	v		

29 1.Introduction

Energy demand has increased rapidly worldwide due to inefficient use and conversion of energy in 30 different applications. Therefore, reduction of losses and enhancing heat transfer processes in energy 31 systems have become an essential area of research in recent years (Jouhara et al. 2017). Heat pipe offer 32 33 an effective way to transfer thermal energy by utilising the latent heat of the working fluid by means of evaporation and condensation passively in a closed container. Due to their relatively low thermal 34 resistance, compact and employing a small quantity of the working fluid, they have widely used in 35 different applications such as solar thermal systems, heat exchangers and electronics cooling. Heat 36 pipes consist of two main sections: the evaporator where the heat is absorbed by the working 37 fluid; and the condenser in which heat is rejected. After the heat is added to the evaporator 38 section, the liquid reaches its saturation temperature and evaporates generating vapour. Due to 39 the difference in the vapour pressure between the evaporator and the condenser, it rises to the 40 condenser (with the assistance of the bouncy forces) where it condenses delivering its latent 41 heat to the coolant at the condenser. At that time, the vapour condenses due to a lower 42 temperature in the condenser and returns to the evaporator by gravity, if the heat pipe is 43 wickless (thermosiphon), or by capillary force, if a wick heat pipe is used. A special attention 44 has been paid to a two-phase closed thermosyphon (TPCT) due to its simplicity and cost-effectiveness 45 (Alammar et al. 2017). 46

Electrical Discharge Machining (EDM) is an advanced fully controlled technique that uses the electricspark to remove small pieces from a metal workpiece forming different shapes or surface roughness.

This performs by applying a high-frequency electrical current through an electrode which producing a very high-temperature resulting in erosion of a tiny piece of the metal. The electrode is controlled to erode a specified thickness of metal from the sample. Both the workpiece and electrode are submerged in a dielectric fluid for cooling purposes and removing the resulting eroded material (Johnson Waukesha).

Several research works have been carried out to investigate enhancing the thermal performance of heat 54 pipes using two different techniques. The first technique employs addition of nanoparticles to the 55 working fluid to increase its thermal conductivity and enhance heat pipe performance. Different studies 56 57 have investigated the effect of using various nanoparticles with water such as CuO nanoparticles (Yang et al. 2008; Liu et al. 2010; Manimaran et al. 2012; Cheedarala et al. 2016), Al2O3 nanoparticles (Noie 58 et al. 2009; Aly et al. 2017), silver nanoparticles (Paramatthanuwat et al. 2010; Ghanbarpour et al. 59 2015), iron oxide nanoparticles (Huminic et al. 2011; Huminic & Huminic 2013), graphene 60 61 nanoparticles (Sadeghinezhad et al. 2016) and multiwalled carbon nanotubes functionalized with ethyl-62 enediamine EDA-MWCNT nanoparticles (Shanbedi et al. 2012a). It was found that the best nanoparticles concentration which provided the highest thermal performance was 1.0wt% 63 (Yang et al. 2008; Shanbedi et al. 2012b), 0.1wt% (Sadeghinezhad et al. 2016), 0.06wt% 64 (Cheedarala et al. 2016) and 3wt% (Alv et al. 2017). Different studies showed that using 65 nanofluid increased the heat transfer coefficient by 46% (Yang et al. 2008) and 30.4% (Aly et 66 al. 2017), increased CHF by 30% (Yang et al. 2008) and 79% (Cheedarala et al. 2016), 67 increased thermal performance (Liu et al. 2010), by 14.7% (Noie et al. 2009), 70% 68 (Paramatthanuwat et al. 2010), 93% (Shanbedi et al. 2012b), 37.2% (Sadeghinezhad et al. 69 2016) and reduced the thermal resistance (Sureshkumar et al. 2013) by 62% (Manimaran et al. 70 2012), 48% (Sadeghinezhad et al. 2016) and 18.2% (Aly et al. 2017). Also, it was concluded 71 that some nanoparticles may deposit on the heat pipe wall making a coating resulting in an 72 increase of the surface wettability (Sadeghinezhad et al. 2016; Cheedarala et al. 2016). 73

74 On the other hand, some researchers have implemented different surface characteristics to enhance the thermal performance of heat pipes. (Han & Cho 2002) investigated the performance of a micro-grooved 75 76 thermosyphon heat pipe for different working fluids, number of grooves and operating temperatures. They found that the number of 60 grooves correspond to the highest condensation heat transfer 77 78 performance which was 2.5 times higher than that of a plain thermosyphon. Also, the condensation heat 79 transfer coefficients of grooved thermosyphons filled with methanol and ethanol were 1.5-2 and 1.3-1.5 times higher compared to the plain one, respectively, and water provides the highest heat transfer 80 81 rate. The thermal characteristics of two thermosyphon heat pipes with straight and helical grooves filled with water have been investigated by (Han & Cho 2005) for different inclinations, fill ratios and 82 operating temperatures. It is concluded that the fill ratio of 30% exhibits the highest heat flux. In 83 84 addition, angles of 25-30° and 40° provide the best thermal performance for helical and straight grooves, 85 respectively. (Jiao et al. 2005) studied theoretically and experimentally the effect of thin-film 86 evaporation in a groove heat pipe. They reported that the performance of the grooved heat pipe is highly affected by the thin film evaporation where the reduction in evaporator temperature is considerably 87 88 larger than in condenser temperature. Also, the thin film region is enlarged by the decrease in the contact 89 angle which increases the heat transfer performance. A similar mathematical study to (Jiao et al. 2005) has been carried out by (Jiao et al. 2007), but the thin fill region inside the groove was divided into three 90 different regions instead of one region. A numerical thermal model has been developed to predict the 91 92 thermal performance of a micro-grooved flat plate heat pipe and validated with an experimental study (Lefèvre et al. 2008). They found that the optimum dimensions of the rectangular groove are 0.36, 0.7 93 and 0.1 mm corresponding to groove width, height and fin width, respectively. These dimensions 94 95 provide a maximum heat flux and lowest thermal resistance. (Yong et al. 2010) investigated the 96 performance of a heat pipe with micro-grooves manufactured by Extrusion-ploughing process. The study reported that the heat transfer limit for the grooved heat pipe fabricated by the new technique is 97 98 larger than that for the normal grooved heat pipe, thus the low heat transfer limit for axially microgrooved heat pipe can be resolved. (Wong & Lin 2011) investigated the impact of surface wettability 99 on the performance of evaporator in a mesh wicked flat plate heat pipe with water, methanol and acetone 100 as working fluids. They concluded that the heat transfer limit decreases as the contact angle of the 101

102 copper surface with water increases, while it is unaffected by methanol and acetone. (Solomon et al. 2012) studied the effect of nanoparticles coating on the thermal performance of screen wicked heat 103 104 pipe. Results revealed that the heat transfer coefficient and thermal resistance of the evaporator 105 increases and reduces by 40%, respectively, while the thermal performance in the condenser section 106 decreases compared with an uncoated heat pipe. It is also reported that reduction of 19%, 15%, and 14% is achieved at heat loads of 100, 150 and 200 W respectively. Thermal characteristics of a 107 108 horizontal grooved heat pipe with different surface wettability for the three sections, evaporator, 109 adiabatic and condenser has been investigated by (Hu et al. 2013). The study revealed that significant decrease is achieved in the total thermal resistance due to the change to the surface characteristics to 110 hydrophilic, gradient wettability and normal surface for evaporator, adiabatic and condenser sections, 111 respectively. Also, more than 42% increase in the dry out limit of the grooved heat pipe is obtained. 112

(Rahimi et al. 2010) changed the surface characteristics of the evaporator and condenser to investigate 113 114 their influence on the thermal performance of a two-phase closed thermosyphon using water as a working fluid. The study showed that the thermosyphon efficiency can be increased by 15.27%, 115 116 whereas a decrease of 2.35 times in the thermal resistance is obtained compared with the plain TPCT. 117 Another surface modification study has been carried out by (Solomon et al. 2013) to test the heat transfer performance of an anodized Aluminium thermosyphon charged with acetone. It is found that a 118 119 maximum reduction in thermal resistance and increase in heat transfer coefficient of the TPCT 120 evaporator is 15% compared with non-anodized thermosyphon. In addition, a negligible effect of 121 anodized TPCT is observed on the condenser thermal performance. (Hsu et al. 2014) employed different surface characteristics in terms of contact angle in the evaporator and condenser sections to 122 investigate the thermal performance of a TPCT. Experimental results showed that when evaporator and 123 condenser are superhydrophilic and superhydrophobic, respectively, the highest performance of the 124 TPCT is obtained where the maximum reduction in the thermal resistance is 26.1% compared with plain 125 126 one. Also, the worst thermal performance of the thermosyphon is observed when the whole inside wall of the TPCT is superhydrophilic. The effect of internal helical microfin on the condensation heat 127 transfer performance in a TPCT has been investigated by (Wang et al. 2012). They reported that the 128

129 existence of the internal helical microfin provides a better thermal response and increases the heat transfer coefficient of condensation by 116.87% at high heat load. Also, A correlation for predicting the 130 condensation heat transfer coefficient of the TPCT was proposed. (Nair & Balaji 2015) investigated 131 numerically using Fluent and Matlab the effect of internal fins inside the condenser section on the 132 133 performance of a two-phase closed thermosyphon. They concluded that adding 8 fins in the condenser section increases the thermal conductivity of the TPCT by about 43%. It is also reported that additional 134 condensate mass of 22% and 32% can be produced using 8 and 12 fins, respectively, which would be 135 136 helpful to avoid the dry out during the operation of the thermosyphon. A similar study to (Nair & Balaji 2015) has been carried out experimentally by (Naresh & Balaji 2017), but for various fill ratios and two 137 working fluids, water and acetone. They concluded that at low heat load, reduction of 17% and 35.48% 138 139 is obtained in the temperature and thermal resistance of TPCT due additional condensate mass resulting 140 from inserting six internal fins in the condenser section. It is also reported that the optimum thermal 141 performance of the TPCT is achieved at a fill ratio of 50%. In addition, acetone exhibits higher performance at low heat loads, while water provides better performance at high heat inputs. 142

Many researchers have carried out numerous experimental investigations to enhance the thermal 143 144 performance and increase the heat transfer limit of heat pipes. This has been achieved by implementing 145 different means namely, using nanoparticles to improve the thermal characteristics of fluids or changing 146 the surface features of the wall using coatings or making micro-grooves. However, the preparation and using of nanofluids would be complex and occupied by instability and agglomeration of the 147 148 nanoparticles. In addition, surface coatings can be a difficult process, making additional conduction thermal resistance, time-consuming and expensive, whereas making micro-grooves may reduce the 149 150 boiling heat transfer limit of heat pipes.

In contrast, making a roughness on the internal wall of a TPCT implementing a new technique does not need any use of such additional coatings or materials. This would produce an effective energy conversion device that can be used in many applications. Therefore, the objective of this work is to enhance the thermal performance of thermosyphon heat pipe by making an internal wall roughness employing a new advanced machining technique named as Electrical Discharge Machining (EDM). To

achieve this goal, a copper tube was machined to make the wall roughness, manufactured and tested to compare its thermal performance with a plain copper thermosyphon at two different initial subatmospheric pressures (3 and 30 kPa) and various heat loads.

2.Experimental work

160 **2.1. Manufacture of the rough surface**

Electrical Discharge Machining (EDM) or Spark Erosion Machining (SEM) was used to make a surface 161 roughness inside a tube with a 200 mm length, 12.7 mm outside diameter and 1.6 mm thickness. This 162 machine generates an electrical spark between a cutting wire (electrode) and a sample material. The 163 spark indicates the flowing of the electrical power through the wire. Thus, the material (workpiece) 164 starts melting due to the intensively produced heat which produces a very high temperature. The spark 165 is controlled and positioned cautiously in order to machine only the material surface. Deionized water 166 is always used as a dielectric medium for the spark in the case of the wire EDM. Water not only 167 functions as a coolant but also to remove the eroded material away from the surface. The wire diameter 168 169 is between 0.1-0.3 mm and is made either from brass or copper. Also, the electrode (wire) must not be in direct contact with the sample material and the workpiece must be electrically conductive. The 170 minimum eroded thickness is 0.00254 mm and the maximum is 0.051 mm per one pass (Johnson 171 172 Waukesha).

The resulting roughness was measured using Mitutoyo Surftest SJ-310 tester in terms of two 173 174 parameters. The first is *Ra* which represents the average distance between the peaks and valleys and the deviation from the mean line throughout the surface and along the length of the surface. The second is 175 176 Rz which represents the average of five sampling lengths by indicating the vertical distance between the highest peak and the deepest valley for each sampling length. The two roughness parameters Ra and 177 Rz are illustrated in Fig.1a and Fig.1b, respectively. The surface roughness was measured at five 178 179 different positions on the sample surface, Table 1 illustrates these values. Also, two actual zoomed 180 photos for rough and plain surfaces are presented in Fig.2a and Fig.2b, respectively. To report the wettability of the two surfaces, an optical tensiometer-contact angle meter was used to measure the 181

182 contact angle employing the sessile drop technique. The measured contact angles for the rough and183 plain surfaces are shown in Fig.3a and Fig.3b, respectively.



Fig.1 Sketch shows: (a)- Ra, arithmetical mean roughness and (b)- Rz, mean roughness depth

Table 1. Values of *Ra* and *Rz*

Rough surface		Plain surface		
Item	Ra (µm)	Rz (µm)	Ra (µm)	Rz (μm)
1	2.935	15.256	0.289	1.89
2	3.658	22.268	0.278	1.764
3	3.675	20.460	0.275	1.687
4	3.664	21.568	0.275	1.67
5	3.639	21.376	0.281	1.82





Fig.2a Rough copper surface



Fig.2b Plain copper surface



- 195
- 196

Fig.3 Measured contact angle for: (a) Rough and (b) plain copper surfaces

197

198 **2.2.** Test set up and procedure

An experimental apparatus was developed to investigate the effect of the surface roughness on the heattransfer performance of the TPCT at a range of heat inputs and two initial pressures.

201 After the roughness was made on the entire internal wall of the TPCT, the resulting rough tube and 202 another plain copper tube were employed to fabricate two thermosyphon heat pipes. The process starts 203 by rinsing the two tubes many times with the ethanol to remove any grease or other Contaminants, then 204 washing with deionised water to ensure that all ethanol was removed. After that, the two proposed 205 thermosyphons were evacuated to a desired pressure (3 kPa or 30 kPa) using a vacuum pump, then they 206 were charged with deionised water to fill the half of the evaporator (50%) using a syringe as shown in Fig.4a. The thermosyphon is 200 mm long and consists of two sections, the evaporator and condenser 207 208 with 100 mm length each, 12.7 mm outside diameter and 1.6 mm thickness. The condenser section is 209 surrounded by a brass water jacket of 16 mm inside diameter and 28 mm outside diameter to remove 210 the heat from the condenser using water as a cooling liquid. Eight type T surface thermocouples were fixed on the outer surface of the TPCT to measure the wall temperature, five thermocouples at the 211 evaporator and three at the condenser. In addition, two type T probe thermocouples were fitted in the 212 inlet and outlet of the water jacket to measure the inlet and outlet temperatures of the cooling water. 213 Before using the thermocouples, All the ten thermocouples were immersed in water at a constant 214 temperature to be calibrated with an RTD thermocouple where the maximum deviation from the RTD 215 216 reading was found to be ±0.4°C at steady state. Fig.4b illustrates the TPCT dimensions and the positions 217 of thermocouples.



218

1-Heat pipe, 2-Electrical heater, 3-Thermocouples positions, 4- Water jacket, 5-Syringe, 6-Pressure gauge, 7-Three-way valve, 8-Vacuum pump, 9-Constant temperature water bath, 10- Flow meter 11-Computer, 12-Data logger, 13-Variable transformer, 14-Multimeter, 15-Power meter.

222

Fig.4: (a)- Test rig schematic diagram and (b)- Dimensions and thermocouples positions

223

224 An electrical heater with a maximum power of 160 W was used to supply the heat to the evaporator section where it was wrapped evenly to distribute the heat input equally on the evaporator surface. 225 Consequently, the value of the heat input applied to the evaporator wall can be changed by changing 226 the input voltage using a variable transformer. Also, a wattmeter and multimeter were used to measure 227 228 the heat load. Comparing the readings of the wattmeter, multimeter (volt and ampere) and the value of 229 the output heat, it is found that the maximum uncertainty in the input energy is about 3.2%. A high-230 temperature superwool blanket insulation of 50 mm thickness was used to reduce the thermal losses 231 from the evaporator wall of the TPCT, so, the heat losses were neglected. This was also proved by 232 comparing the heat output which was found to be more than 93% in all tests. Also, a rotameter was employed to measure the coolant mass flow rate at the condenser section with the uncertainty of 233 measuring the flow rate value of 2.8%. In addition, to ensure that all tests are performed at the same 234 inlet temperature of the cooling water, a constant temperature water bath was used to maintain the 235

coolant inlet temperature at the desired temperature. All thermocouples were connected to a data takerto send their temperature readings into a computer to be saved and analysed.

After the test rig was built, it was ready to examine the TPCT performance. Firstly, the water bath is set 238 at a desired cooling temperature $(20^{\circ}C)$. Then, the globe valve before the rotameter is opened to allow 239 the cooling water to circulate throughout the water jacket at the condenser section. Also, the rotameter 240 is adjusted to a specified flow rate of 0.0025 kg/s using the globe valve to be fixed for all tests. Before 241 power is supplied to the rope heater, enough time is provided to ensure that all thermocouples readings 242 reach approximately a value of 20°C which is another proof of thermocouples consistency and accuracy 243 244 in temperature measurement. Then, the power is supplied to the electrical heater by adjusting the variable transformer to a certain value which equivalent to the desired heat input needed to the 245 evaporator section. This heat input can be obtained by multiplying the voltage times the current as well 246 as the reading of the wattmeter. After all temperatures reach the steady state, the data is saved and the 247 248 power is switched off. Some runs were repeated three times to prove the repeatability and accuracy of the test facility and the procedure used. The measured quantities are the heat load, operating pressure, 249 250 coolant mass flow rate, inlet and outlet temperatures of the cooling water and wall temperatures of the 251 evaporator and condenser sections.

- 252
- 253

254 2.3. Data reduction

Parameters such as evaporator and condenser thermal resistances, total thermal resistance and the
evaporator heat transfer coefficient need to be determined to obtain and compare the heat transfer
characteristics of the plain and modified TPCTs.

258 The evaporator and condenser thermal resistances can be obtained from the following equations:

259
$$R_e = \frac{T_{e,av} - T_{sat}}{Q}$$
(1)

260
$$R_c = \frac{T_{c,av} - T_{sat}}{Q}$$
(2)

261 Where R_e and R_c are the evaporator and condenser thermal resistances, respectively, T_{sat} is the

saturation temperature which corresponds to operating pressure at each heat input, and Q is the heat

263 input calculated from:

264
$$Q = IV$$
(3)

265 Where *I* and *V* are the circuit current and voltage, respectively.

266 $T_{e,av}$ and $T_{c,av}$ are the average wall temperatures of the evaporator and condenser, respectively and can

..(5)

be obtained as follow:

268
$$T_{e,av} = \frac{T_0 + T_2 + T_4 + T_6 + T_8}{5}$$
(4)

- 269 $T_{c,av} = \frac{T_{13} + T_{16} + T_{19}}{3}$
- 270 Therefore, the total thermal resistance of the TPCT can be calculated from:

271
$$R_t = \frac{T_{e,av} - T_{c,av}}{Q}$$
(6)

- 272 Where R_t is the total thermal resistance of the thermosyphon.
- 273 The evaporator heat transfer coefficient can be obtained from the following equation:

274
$$h_e = \frac{Q}{\pi D_i L_e (T_{i,av} - T_{sat})}$$
(7)

Where h_e is the evaporator heat transfer coefficient, D_i and L are the inside diameter and length of the evaporator and $T_{i,av}$ is the inside surface average temperature of the evaporator and can be determined from:

278
$$T_{i,av} = T_{e,av} - \frac{Q}{2\pi LK} ln \left(\frac{D_o}{D_i}\right) \quad \dots \dots \dots \dots \dots (8)$$

279 Where D_o is the outside diameter of the evaporator and K is the solid thermal conductivity.

280 **3- Results and discussion**

281 **3.1.** Temperature distribution

A TPCT with internal wall roughness made using the EDM technique was tested and compared with a smooth TPCT to investigate the enhancement in the heat transfer at a range of heat loads and two different initial pressures.

285 Variation of the wall temperature of the plain and rough thermosyphons with distance along the wall at a heat load of 100 W is shown in Fig.5a and Fig.5b for initial pressures of 3 and 30 kPa, respectively. 286 287 Fig. 5a shows that a significant reduction in the evaporator wall temperature is achieved for the TPCT with roughness compared to the plain TPCT. This can be explained by the increase in the nucleation 288 sites density (as confirmed by Fig. 2a), thereby increasing the frequency of bubbles generation 289 (Solomon et al. 2013) resulting from a rough surface, which transfers heat efficiently from the TPCT 290 291 wall reducing noticeably the wall temperature. Another reason causing the decrease in the evaporator wall temperature is the hydrophilic characteristics of the modified wall [25, 27] which make the surface 292 wetted with liquid instead of vapour as illustrated in Fig.3a. However, in the condenser section, it is 293 observed that the condenser wall temperature of the plain TPCT is higher than that for the TPCT with 294 295 roughness, but the difference is much lower compared with the evaporator. This also may be attributed to the wettability feature of the rough surface which provides opposite effect on the condensation heat 296 transfer in the condenser. This results in increasing the condensate film thickness which leads to 297 298 additional heat transfer resistance, thereby lower condenser wall temperature. Fig.5b presents a similar trend as Fig.5a in the evaporator section for both plain and modified TPCTs. However, a lower 299 300 difference in evaporator temperature is obtained between the two thermosyphons due to the higher 301 pressure. The reason behind that may be attributed to the activation of small surface cavities of the plain 302 TPCT when the pressure increases (Khodabandeh & Palm 2002) which reduces the wall temperature 303 of the plain TPCT. On the other hand, most cavities of the rough surface are already activated, so the increase in pressure produces relatively less temperature reduction compared with the plain TPCT, but 304 305 the evaporator wall temperature of the rough TPCT is still lower than that of the plain TPCT due to the

roughness effect. Also, a different trend of the condenser wall temperature is observed at a pressure of
30 kPa compared with that at 3 kPa for both TPCTs. The reason will be explained in the discussion of
Fig.7a and b.



Fig.5 Comparison of thermosyphon wall temperature between plain and rough TPCT at heat load 100 W and
 initial pressures: (a)-3 kPa and (b)-30 kPa

312

Figs.6a and b also show the temperature distribution along the wall of the two TPCTs at 3 and 30 kPa, 313 respectively, but at a heat input of 160 W. It is observed that the difference in the evaporator wall 314 temperature between the plain and modified thermosyphons is higher compared with that at a heat load 315 316 of 100 W. This could be explained as: before reaching the critical heat flux, when the heat load 317 increases, the heat transfer mechanism is enhanced due to the generation of more bubbles transferring further heat from the heating surface to the fluid, thereby further reduces the evaporator wall 318 319 temperature. On the other hand, approximately the same difference in the condenser wall temperature 320 as in the case of 100 W is obtained when the pressure is 3 kPa (Fig.6a). However, when the pressure is 30 kPa (Fig.6b), a higher difference in the wall temperature of the condenser is noticed between the two 321 322 TPTCs compared with that at 100 W, especially at the upper part of the rough thermosyphon. This can be explained that the rate at which the vapour is generated at 160 W is higher than that at 100 W in both 323 324 plain and rough TPCTs. Therefore, the rate of the condensate removal is smaller than the rate of droplets growth, which leads to thickening the condensate film thus reducing the condenser wall temperature 325 326 (Attinger et al. 2014). This effect is higher in the case of the rough condenser due to the wettable

327 characteristics of the rough surface compared with the smooth surface, so that the difference at 160 W



328 is higher than that at 100 W.

Fig.6 Comparison of thermosyphon wall temperature between plain and rough TPCT at heat load 160 W and
 initial pressures: (a)-3 kPa and (b)-30 kPa

332

Effect of two initial pressures of 3 and 30 kPa on the wall temperature distribution is presented in Fig.7a 333 334 and Fig.7b for rough and plain thermosyphons, respectively, at a heat input of 160 W. It can be seen that for both TPCTs, using the pressure of 3 kPa provides a lower evaporator wall temperature compared 335 with 30 kPa due to corresponding low saturation temperature which leads to earlier evaporation start-336 up, thereby a lower evaporator wall temperature (Yang et al. 2008), (Lee et al. 2014). On the other 337 338 hand, a higher condenser wall temperature is obtained employing 3 kPa at the middle and upper parts of the condenser (T16 and T19), while it is lower at the lower part (T13). This may result from the rising 339 of the saturated vapour to the upper part of the condenser and the small condensate film thickness, 340 resulting in a low thermal resistance, thereby higher heat transfer coefficient between the hot vapour 341 and the wall leading to a higher condenser wall temperature at the upper part compared with the lower 342 343 part (Alizadehdakhel et al. 2010). In contrast, in the case of 30 kPa, the upper part of the condenser wall exhibits a lower wall temperature compared to the lower part for both TPCTs. This may be attributed 344 to the presence of non-condensable gases in the case of 30 kPa which blocks the upper part of the 345 346 condenser preventing the hot vapour to reach this part and deteriorating the heat transfer mechanism leading to a lower condenser wall temperature compared with the lower part at 3 kPa. Thus, a smaller 347 348 condensate quantity is produced making the wall temperature of the lower part of the condenser (T13)

- 349 for both TPCTs at 30 kPa higher than that at 3 kPa. In addition, the difference in the wall temperature
- between the two pressures is higher in the case of modified TPCT compared with plain one for the same 350
- 351 reasons explained in the discussion of Fig.5a-b and Fig.6a-b.
- 352



- 354 Fig.7 Comparison of thermosyphon wall temperature between initial pressures 3 and 30 kPa at heat load 160 W 355 and: (a)-TPCT with roughness and (b)-Plain TPCT
- 356

3.2. Thermal performance of the Thermosyphon 357

Variation of evaporator thermal resistance (R_e) with the heat load for the plain and rough TPCTs are 358 shown in Fig.8a and Fig.8b at two different initial pressures of 3 and 30 kPa, respectively. They show 359 that a considerable decrease in the evaporator thermal resistance is achieved when the rough 360 thermosyphon is used compared with the plain one for both pressures. It is found that the reduction in 361 the evaporator thermal resistance varies with the heat load from about 51-68% and from 68-115% for 362 pressures of 30 and 3 kPa, respectively (30.4% (Aly et al. 2017), 40% (Solomon et al. 2012), 15.01% 363 (Solomon et al. 2013)). This reduction in R_e may result from the presence of the roughness in the 364 evaporator wall which creates additional nucleation sites leading to generate more bubbles, thereby 365 more heat is released from the evaporator internal surface. Also, the rough surface increases the wall 366 wettability by decreasing the contact angle making the liquid in continuous contact with the evaporator 367 wall removing the vapour away from the wall surface. In addition, it is observed that the R_e for the plain 368 369 TPCT increases at a heat input of 140 W, while for the TPCT with roughness, it increases at 150 W indicating an increase in the CHF for the rough TPCT. 370



Fig.8 Comparison of evaporator thermal resistance versus heat input between plain and rough TPCTs at initial
 pressures: (a)-3 kPa and (b)-30 kPa

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However, Fig.9a and Fig.9b show that the condenser thermal resistance (R_c) increases when the 375 376 modified TPCT is employed compared with the plain one which worsens the heat transfer performance in the condenser section ((Solomon et al. 2012) also reported higher R_c for coated TPCT and (Solomon 377 378 et al. 2013) reported no reduction in R_c for anodised TPCT). This may be attributed to the fact that the high surface wettability produced from the rough surface can form a liquid film on the condenser wall 379 380 which prevents the vapour to be in direct contact with the condenser inner wall resulting in additional thermal resistance. The maximum increase in the R_c is about 22% compared with plain TPCT. It is also 381 seen from Fig.9a 3 kPa initial pressure that R_c of the rough and plain TPCTs decreases steadily with the 382 heat load, while Fig.9b for initial pressure of 30 kPa shows that R_c of both TPCTs decrease sharply with 383 the heat load. This may be explained by a larger amount of vapour generated at the low pressure 384 compared with the high pressure. This increases the liquid film thickness, thereby the condenser thermal 385 386 resistance reducing the effect of heat input on the thermal resistance at the low pressure. The film 387 thickness on the rough wettable condenser wall is higher (at 3 kPa), so that a higher difference is noticed between the two thermal resistances at a pressure of 3 kPa (Fig.9a) compared with that at 30 kPa 388 (Fig.9b) and they both decreases with the input energy. 389



Fig.9 Comparison of condenser thermal resistance versus heat input between plain and rough TPCTs at initial
 pressures: (a)-3 kPa and (b)-30 kPa.



390

Despite the increase in condenser thermal resistance for the rough TPCT, a noticeable decrease in the 394 total thermal resistance (R_t) of the rough TPCT is shown in Fig.10a and Fig.10b at 3 and 30 kPa, 395 respectively, due to the high reduction in the evaporator thermal resistance. The reduction in the R_t 396 varies with the input energy from about 9-13% and 28-42% compared with the plain TPCT at 30 and 3 397 398 kPa respectively (18.2% (Aly et al. 2017), 19% (Solomon et al. 2012), 125% (Rahimi et al. 2010), 15% (Solomon et al. 2013), 26.1% (Hsu et al. 2014), 35.48% (Naresh & Balaji 2017)). In addition, Fig.10a 399 (3 kPa) shows a same trend as the R_e in Fig.8a, and almost a same rate of decrease in the R_t for both 400 TPCTs with the heat load is observed at a pressure of 30 kPa (Fig.10b). 401





405

pressures: (a)-3 kPa and (b)-30 kPa.



408 51-68% at 3 and 30 kPa, respectively (40% (Solomon et al. 2012), 50-100% for methanol and 30-50% 409 for ethanol (Han & Cho 2002), maximum of 116.87% (Wang et al. 2012)). In addition, at a pressure of 410 3 kPa (Fig.11a), h_e generally increases as the heat load increases for the both TPCTs. However, the rate 411 of increase in h_e is higher for the modified TPCT compared with the plain one and it becomes 412 approximately constant after a heat load of 130 W for the plain TPCT. Therefore, the difference in h_e 413 between the two TPCTs increases as the input energy increases. This is also true at a pressure of 30 kPa 414 (Fig.11b), but with a lower difference in h_e and a lower rate of increase for the rough TPCT.



Fig.11 Comparison of evaporator heat transfer coefficient versus heat input between plain and rough TPCTs at
 initial pressures: (a)-3 kPa and (b)-30 kPa.

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415

419 **4- Conclusions**

Thermal performance of a TPCT with an internal surface roughness produced using a new 420 technique of EDM was tested to investigate the enhancement of heat transfer characteristics. 421 This was carried out by comparing the modified TPCT with a plain TPCT at various heat loads 422 and two different initial pressures (sub-atmospheric pressures). It is concluded that a significant 423 decrease in the evaporator wall temperature is achieved using the resurfaced thermosyphon at 424 both initial pressures 3 and 30 kPa. It is also seen that the reduction increases as the input 425 energy increases. In addition, less reduction is obtained at a pressure of 30 kPa compared with 426 3 kPa and the difference in $T_{e,av}$ between the two pressures for the rough TPCT is higher than 427 428 that for the plain. Accordingly, a considerable decrease in the evaporator thermal resistance

and enhancement in the evaporator heat transfer coefficient of 115% and 68% are obtained at 429 3 and 30 kPa, respectively. However, the condenser wall temperature for the rough TPCT is 430 noticed to be lower than that for the plain one. Likewise, the thermal resistance of the condenser 431 section for the rough TPCT is higher, but the difference in the condenser much lower than that 432 at the evaporator. Thus, the total thermal resistance for modified TPCT is decreased by about 433 42% at a pressure of 3 kPa, whereas it is reduced by 13% at 30 kPa compared with the plain 434 TPCT despite the increase in the condenser thermal resistance. More enhancement in the 435 performance of the TPCT may be achieved if another proved enhanced surface is employed in 436 437 the condenser rather than the rough surface or using a nanofluid such as Ti/H2O which was proved to enhance the h_c by 2-3 times (Baojin et al. 2009) with the rough TPCT. This may need 438 to be investigated by a further research study and can be included as a future question: how can 439 enhance the heat transfer characteristics in the condenser to achieve more enhancement in the 440 thermal performance of the TPCT? 441

Therefore, making a roughness in the internal wall surface of the TPCT using EDM provides
a simple and inexpensive technique to enhance the heat transfer performance of the TPCT. This
would offer an efficient energy conversion and heat removal device for different systems in
many applications.

446

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-Heat pipe performance is enhanced by making a wall roughness using a new technique.

- -A significant reduction in evaporator thermal resistance of 115% is obtained.
- -A significant increase in evaporator heat transfer coefficient of 115% is achieve.
- -A considerable reduction of 42% in the total thermal resistance is obtained.