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Influence of Fluid Charge Rate on Gravity Assisted Heat Pipe Performance at Low Temperatures

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ABSTRACT

The heat pipe is an effective heat transfer device. Thermosyphon is a closed, vacuumed, fluid contained and generally pipe shaped heat transfer device. It ensures high heat transfer even at very low temperature differences. Heat pipes have very wide utilization fields. Therefore, many studies have been carried out to improve the thermal performances of heat pipes. In this study, the effect of the fluid charge rate on the operation of the heat pipe was examined. To do this, a heat pipe model with a diameter of 25 mm was created. The heat pipe's evaporator, condenser and adiabatic lengths are 2m, 1m and 0.1m respectively. The heat pipe has been examined for various fluid charge rates and different heat source temperatures. The increase in fluid charge rate has been seen to have a negative impact on the operation of the heat pipe, particularly at low temperatures. In addition, at low temperatures the acetone performed better heat transfer than the methanol. At high temperatures, the methanol performed better heat transfer than the acetone.

Keywords: Fluid Charge Rate, Heat Pipe, Thermosyphon, Working Fluid.

Akışkan Şarj Oranının Düşük Sıcaklıklarda Yerçekimi Destekli Isı Borusu Performansına Etkisi

ÖZET

Isı borusu etkili bir ısı transfer cihazıdır. Isı borusu vakum prosesi uygulanmış her tarafı kapalı olan içerisinde kapiler yapıda fitil yerleştirilmiş ya da oluşturulmuş ve belirli miktarda

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iş akışkanı bulunan genellikle boru şeklinde bir sistemdir. Çok düşük sıcaklık farklılıklarında bile yüksek miktarda ısı transferi gerçekleştirir. Isı boruları çok geniş kullanım alanlarına sahiptir. Bu yüzden Isı boruların ısıl performansların iyileştirilmesi için ilgili birçok çalışma yapılmıştır. Bu çalışmada, akışkan şarj oranının ısı borusunun performansına etkisi araştırılmıştır. Bu amaçla çapı 25 mm olan bir ısı borusu modeli oluşturuldu. Isı borusunun evaporatör, kondenser ve adyabatik uzunlukları sırasıyla 2m, 1m ve 0.1m'dir. Çalışmada iş akışkanı olarak aseton ve metanol kullanılmıştır. Isı borusu farklı akışkan şarj oranları ve farklı ısı kaynağı sıcaklıkları için incelenmiştir. Akışkan şarj oranının artmasının, özellikle düşük ısı kaynağı sıcaklıklarında, ısı borusunun performansına olumsuz bir etkisi olduğu görülmüştür. Ayrıca, düşük ısı kaynaklarında aseton metanolden daha iyi ısı transferi gerçekleştirmiştir. Yüksek ısı kaynağında ise metanol asetondan daha yüksek oranda ısı transferi gerçekleştirmiştir.

Anahtar kelimeler: Akışkan Şarj Oranı, Isı Borusu, Termosifon, İş Akışkanı.

1. INTRODUCTION

Thermosyphon is a closed, vacuumed, certain amount of a fluid contained and generally pipe shaped heat transfer device. In some cases, capillary structured wick may be mounted into the thermosyphon to increase the heat transfer. It can transfer high amount of heat via the fluid’s phase change even with small differences of temperatures. Thermosyphon is a device that has three areas; evaporator, adiabatic and condenser. The structure of heat pipe is presented horizontally and vertically in Figure 1.

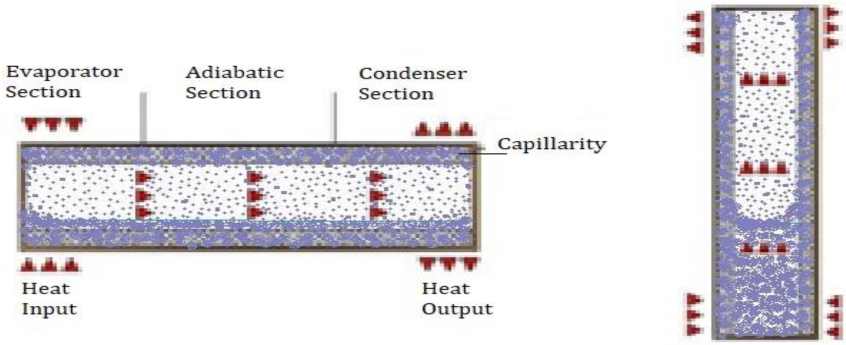

Figure 1. View of heat pipe in horizontal and vertical position

The fluid in the evaporator part takes the heat from the heat source and then starts to evaporate. The evaporated working fluid in the evaporator passes to the heat pipe's condenser by means of the pressure gradient between two sections. The fluid in the condenser section condenses by heat rejection and becomes liquid. Liquid fluid returns to the evaporator section due to the gravitation and capillary forces. This loop between evaporator and condenser goes on as long as there is a temperature difference between them. Because the operation of the heat pipe depends on the loop of the fluid, it is very important that the fluid return to the evaporator part after condensation. Capillary wick creates a capillary force which allows the fluid turn back to the evaporator. Heat pipe with capillary wick is used in places where there is no gravity force or the heat pipe is horizontally positioned. Because the fluid returns to the evaporator due to capillary wick, the evaporator can be located higher than the condenser. If the working fluid motion is provided by gravity, this device is called two-phase closed thermosyphon. Generally the thermosyphon type heat pipe has not capillary structure. The return back of the fluid from the condenser to the evaporator depends on gravity. So the condenser part of the thermosyphon must always be at a higher position than the evaporator. Otherwise the heat pipe will not work because the working fluid could not return from the condenser to the evaporator (Peterson, 1994)

The thermosyphon is considered as an isothermal device since heat is transferred at small temperature differences. High amount of heat can be transferred by the thermosyphon with the evaporation latent heat of the fluid at small temperature differences. Thermosyphon has several advantages such as lack of any mechanical parts and any external power to operate since it is a passive device so it possesses a wide use area (Joudi and Witvit, 1993; Ozsoy, 2005; Şimşak, 2009).

Thermosyphon can be produced in various forms according to the purpose of use, such as cylindrical, planar, and rotary. Thermosyphon is used in the cooling of electronic circuits and computer operating systems, heat recovery systems, air conditioning and ventilation systems, ice and snow protection systems, solar energy systems, space vehicles and many other applications (Faghri, 1995; Ozsoy, 2005).

The heat transfer potential of the thermosyphon depends on such factors as the shape, material, size, working fluid, and fluid charge rate of the thermosyphon. Various fluids can be used as heat carriers in thermosyphon depending on the working temperature and conditions. The working fluid must be able to carry the heat energy received from the evaporator to the condenser very quickly. So the working fluid must have certain properties (Faghri, 1995; ESDU 80017, 2005; Reay and Kew, 2006)

One of the most important properties of fluids used in thermosyphon is to be able to evaporate under the working conditions. However, when choosing the fluid, it is certainly not sufficient to take into account only the temperature range. Hence the fluid's density, evaporation latent heat, surface tension and viscosity must be considered when selecting the fluid. For this purpose the number of merits (Φ) containing all of these properties is defined. For a heat pipe and thermosyphon Φ_1 and Φ_2 merit numbers (Equation 1 and 2) are used, respectively. Furthermore, merit for nucleate boiling Φ_3 is defined in Equation 3. It is used as an important parameter in finding the inside thermal resistances of the thermosyphon (Peterson, 1994; ESDU 80017, 2005).

$$\Phi_1 = \frac{\sigma\, h_{fg}\, \rho_l}{\mu_l} \tag{1}$$

$$\Phi_2 = \left[\frac{h_{fg}\ \lambda_l^3 \rho_l^2}{\mu_l}\right]^{0.25} \tag{2}$$

$$\Phi_3 = 0.32\ \frac{\rho_l^{0.65}\, \lambda_l^{0.3}\ c_{pl}^{0.7}}{\rho_v^{0.25}\, h_{fg}^{\ 0.4}\, \mu_l^{0.1}} \left[\frac{p_v}{p_a}\right]^{0.23} \tag{3}$$

The working fluids and their operating ranges for low temperatures are given in Table 1 (ESDU 80017, 2005).

Table 1. Various working fluids

Fluids	**Operating temperature range (^oC)**
Ammonia (NH_3)	-77 ≥ T ≤ 90
Methanol (CH_3OH)	-97 ≥ T ≤ 180
Toluene ($C_6H_5CH_3$)	-94 ≥ T ≤ 260
Water (H_2O)	30 ≥ T ≤ 360
Acetone (CH_3COCH_3)	-95 ≥ T ≤ 180
Diphenyl ether/Diphenyl	12 ≥ T ≤ 400
O-Dichlorobenzene ($H_6H_4Cl_2$)	-17 ≥ T ≤ 340

The rate of fluid charge in the thermosyphon should be at an optimum value. The insufficient fluid in the thermosyphon or heat pipe results a dry out limit. The dry out limit appears when the fluid volume cannot entirely cover the inside area of the thermosyphon/heat pipe with the liquid film layer. Therefore, the fluid condensing in the condenser part evaporates again before reaching the evaporator section. This leads inside of the thermosyphon to dry out (Nguyen-chi and Groll, 1981.)

In addition, the fluid charge rate of thermosyphon should not be too much. Because, the working fluid may go to the condenser region in liquid form and adversely affect the performance of the thermosyphon. The fluid charge rate is calculated from Equation 4 (ESDU 81038, 2005).

$$F = \frac{V_l}{A\,L_e} \tag{4}$$

Bezrodynyi and Alekseenko, stated that the fluid charge rate should be more than 50% of the evaporator volume. Needed amount of the working fluid is given in Equation 5. This amount is sufficient to provide approximately a 0.3 mm liquid layer thickness on all over the inner surface of the thermosyphon (Bezrodnyı and Alekseenko, 1977).

$$V_l \geq 0.001\,D_i\,(L_e + L_a + L_c) \tag{5}$$

On the basis of the information accessible in the literature, it is suggested that, for thermosyphon, the working fluid fill rate needs to be in the range between 40% with 60% for vertical thermosyphon and 60% with 80% for inclined thermosyphon (ESDU 81038, 2005).

2. MATERIAL AND METHOD

In this study, the performance of a thermosyphon with a different temperature heat source according to the fluid charge rate was theoretically analyzed. For this reason, a thermosyphon model is formed as presented in Figure 2. The heat was applied to the evaporator section and then applied heat was rejected from the condenser section. There is a jacket heat exchanger in the condenser section of the thermosyphon. Water is sent at low temperature to draw heat from the condenser. The inlet temperature of the water entering the jacket was considered constant. The outlet temperature of the water from the jacket was calculated and the amount of carried heat was determined for different conditions.

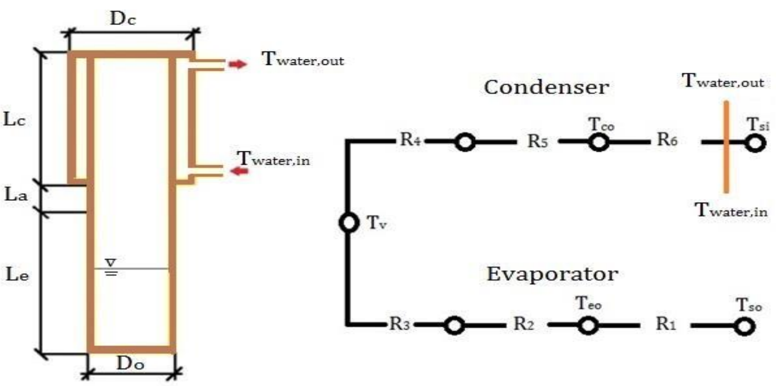

Figure 2. Thermosyphon model and thermal resistances

Thermal resistance is one of the important parameters affecting the operation of the thermosyphon (Figure 2). The thermal (R_1-R_6) resistances that occurred in the thermosyphon are given below (ESDU 81038, 2005):

R_1 is the resistance between the heat source and the evaporator outer surface, and R_6 is the resistance between the condenser outer surface and the heat sink. R_1 and R_6 are calculated from Equation 6 and Equation 7, respectively.

$$R_1 = \frac{1}{h_{eo}\, S_{eo}} \tag{6}$$

$$R_6 = \frac{1}{h_{co}\, S_{co}} \tag{7}$$

R_2 and R_5 are the resistances that occur in the thickness of the wall of the thermosyphon in the evaporator and condenser region, respectively. R_2 and R_5 are calculated from Equation 8 and Equation 9, respectively.

$$R_2 = \frac{ln\left(\frac{D_o}{D_i}\right)}{2\,\pi\, L_e\, \lambda_x} \tag{8}$$

$$R_5 = \frac{ln\,(\frac{D_o}{D_i})}{2\,\pi\, L_c\, \lambda_x} \tag{9}$$

R_3 is the internal resistance of the boiling and condensing fluid in the thermosyphon. This resistance is examined in two parts, including R_{3f} and R_{3p}. The equations of the resistances are given in Equation 10 and Equation 11, respectively. Initially R_{3p} and R_{3f} are calculated to establish R_3. If $R_{3p} > R_{3f}$ is $R_3 = R_{3p}$ is taken; otherwise calculate the mean value of R_3 by interpolation which is given Equation 12.

$$R_{3f} = \frac{C\, \dot{Q}^{0.33}}{{D_i}^{1.33}\, g^{0.33}\, L_e\, {\Phi_2}^{1.33}} \tag{10}$$

$$R_{3p} = \frac{1}{\Phi_3\, g^{0.2}\, \dot{Q}^{0.4} (\pi\, D_i\, L_e)^{0.6}} \tag{11}$$

$$R_3 = R_{3p}\, F + R_{3f}\, (1 - F) \tag{12}$$

R_4 is the resistance of the fluid in the condenser due to film condensation. R_4 is calculated from Equation 13.

$$R_4 = \frac{C\,\dot{Q}^{\,0.33}}{D_i^{\,1.33}\,g^{\,1/3}\,L_c\,\Phi_2^{\,1.33}} \tag{13}$$

The Reynolds number of the thermosyphon is calculated from Equation 14. If the Re number is greater than 1300, the correction factor is used for the R_4 resistance. This is shown in Equation 15.

$$Re = \frac{4\,\dot{Q}}{h_{fg}\mu_l\,\pi\,D_i} \tag{14}$$

$$R_4 = \frac{C\,\dot{Q}^{0.33}}{D^{1.33}\,g^{0.33}\,L_c\,\Phi_2^{1.33}}\,[191\,Re^{-0.73}] \tag{15}$$

The total maximum heat transfer rate of thermosyphon is calculated by Equation 16. The performance of thermosyphon relies on total thermal resistance and temperature difference.

$$\dot{Q} = \frac{\Delta T}{\Sigma R} \tag{16}$$

$$\Delta T = T_{so} - T_{si} - \Delta T_h \tag{17}$$

Hydrostatic pressure (P_p) of the liquid in the evaporator of vertically positioned thermosyphon influences saturation temperature of the working fluid. The pressure at the end of evaporator is determined by Equation 18. The hydrostatic temperature (T_p) is determined from the thermophysical properties of the working fluid according to the hydrostatic pressure. And the hydrostatic temperature difference is calculated from Equation 19 and used in Equation 17 (ESDU 81038, 2005).

$$P_p = P_v + \rho_l\,g\,L_e\,F\,sin\beta \tag{18}$$

$$\Delta T_h = \frac{T_p - T_v}{2}\,F \tag{19}$$

Since the evaporator part of the thermosyphon is heated by electricity there is no convection heat transfer (R_1 = 0). Therefore, the heat source temperature (T_{so}) and the evaporator surface temperature (T_{eo}) are equal to each other. The calculation of the Reynolds number of the water in the condenser outer section (in the water jacket) is given in Equation 20 (Çengel, 2011).

$$Re_{water} = \frac{V_{water}\,D_h}{\vartheta_{water}} \tag{20}$$

If Rewater is less than or equal to 2300, the flow is laminar. The Nusselt number for hydrodynamic and thermally undeveloped laminar flow is calculated from Equation 21 (Genceli, 2010).

$$Nu_{water} = 3.66 + \frac{0.0668\left(\frac{D_h}{L_c}\right) Re_{water}\, Pr_{water}}{1+0.04\left(\left(\frac{D_h}{L_c}\right) Re_{water}\, Pr_{water}\right)^{\frac{2}{3}}} \tag{21}$$

After the Nuwater number is determined, the convection heat transfer coefficient of the jacket (h_{co}) for the water is calculated by Equation 22. Then, the thermal resistance of the outside of the condenser (R_6) can be found (Çengel, 2011).

$$Nu_{water} = \frac{h_{co}\, D_h}{k_{water}} \tag{22}$$

Properties and variables of the thermosyphon are given in Table 2 and Table 3, respectively.

Table 2. Properties of the thermosyphon

Thermosyphon material	Stainless steel
Adiabatic section length, L_a	0.1 m
Evaporator section length, L_e	2 m
Condenser section length, L_c	1m
External diameter of the thermosyphon, D_o	25 mm
Thickness of the thermosyphon, t_x	2 mm
The jacket's inlet diameter, D_c	35 mm
Cooling water flow rate, $\dot{V}$	20 lt/h
Water inlet temperature to the jacket, $T_{water,in}$	15 ^{o}C

Table 3. Variables of the thermosyphon

Working fluids	Acetone, Methanol
Heat source temperatures (^{o}C)	25, 30, 50
Filling rate (%)	15, 30, 45, 60, 75, 90

3. RESULTS

The thermal performance of the thermosyphon that uses acetone or methanol as working fluid, at different charge rates and different heat source temperatures is presented in Figure 3 and Table 4.

As shown in Figure 3 and Table 4, when the heat source temperature is 25 ^oC, acetone and methanol transferred approximately the same amount of heat at low fluid charges. However, as the fluid charge rate increases, the heat transfer rate of both fluids reduces. In addition, the thermosyphon containing methanol transfers less heat than the thermosyphon containing acetone at high fluid charge rates. As the heat source temperature goes up to 30 °C, thermosyphon with acetone and methanol fluids transfer nearly the same amount of heat at low fluid charge rates (up to 50%). But at higher charge rates the amount of heat carried by both thermosyphon is reduced. The thermosyphon using acetone as working fluid performs better than the thermosyphon using methanol at higher charge rates.

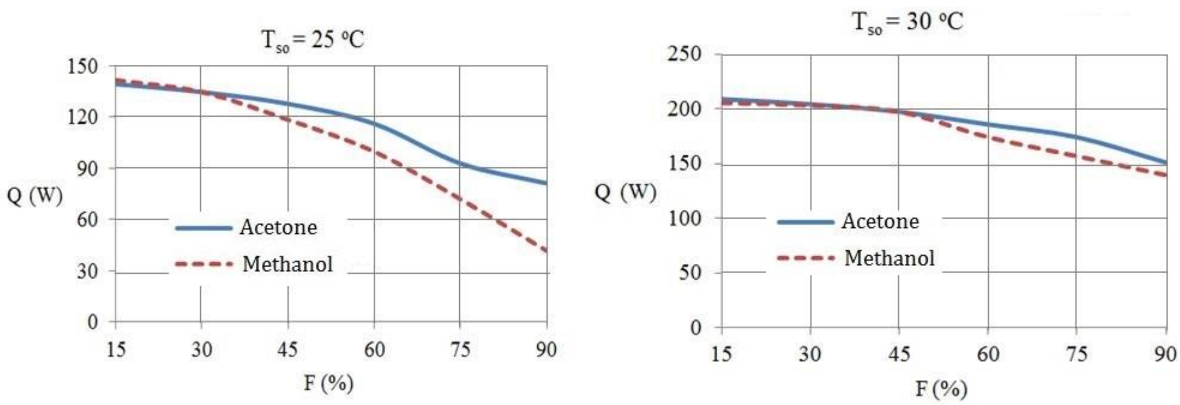

Figure 3. Effect of fluid charge rate on thermosyphon performance

Table 4. Effect of the fluid charge rate on the performance of the thermosyphon

Heat source temperature	Working fluid	Heat flux of thermosyphon (W)					
		%15	**%30**	**%45**	**%60**	**%75**	**%90**
25 ^oC	Acetone	139	134	127	116	92	81
25 ^oC	Methanol	141	134	118	99	71	41
30 ^oC	Acetone	208	204	197	185	174	150
30 ^oC	Methanol	205	203	197	174	156	139

Table 5. Saturation temperature of the working fluid at the different fluid charge rates

Heat source temperature	**Working Fluid**	**Saturation temperature of working fluid (°C)**					
		%15	**%30**	**%45**	**%60**	**%75**	**%90**
30 °C	Acetone	30.21	31.35	32.32	33.01	33.71	33.84
30 °C	Methanol	30.35	32.26	33.84	34.55	35.56	36.56

Table 6. Water outlet temperature from the condenser jacket at the different fluid charge rates

Heat source temperature	**Working Fluid**	**Water outlet temperature (°C)**					
		%15	**%30**	**%45**	**%60**	**%75**	**%90**
30 °C	Acetone	24.00	23.80	23.50	23.00	22.50	21.50
30 °C	Methanol	23.85	23.75	23.50	22.50	21.75	21.00

As the fluid charge rate increase, the hydrostatic pressure goes up along with the saturation temperature of the fluid (Figure 4, Figure 5 and Table 5). If the saturation temperature of the fluid at the evaporator is higher than the heat source temperature, the heat pipe does not work. Because the heat source temperature is not enough to vaporize the working fluid inside the evaporator. The fluid charge rate, especially at low heat source temperatures, is an important factor affecting the operation of the thermosyphon.

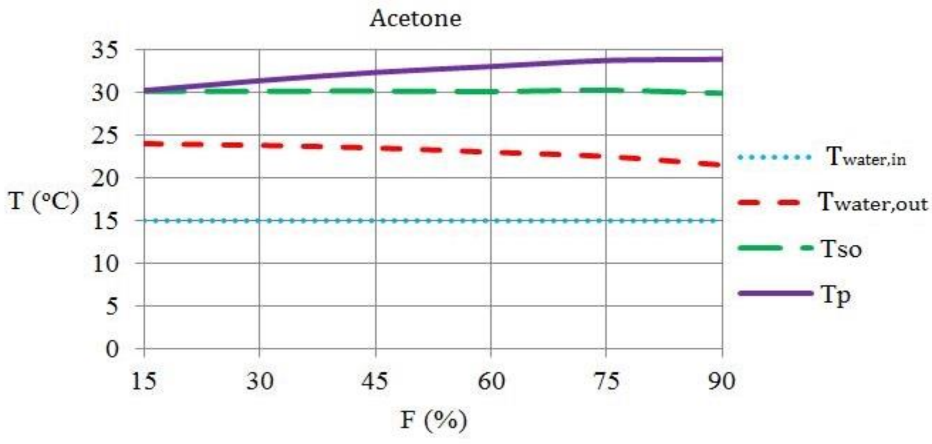

Figure 4. Influence of the fluid charge rate on $T_{water,out}$ and T_p

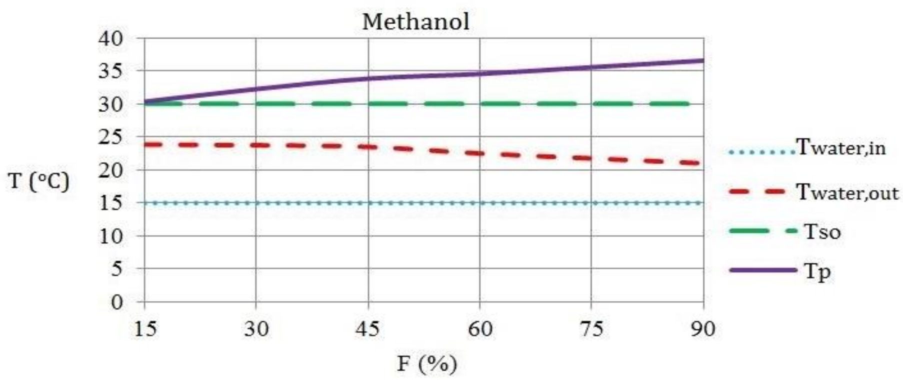

Figure 5. Influence of the fluid charge rate on $T_{water,out}$ and T_p

It is seen that the saturation temperature of methanol is higher than that of acetone as shown in Table 5. Therefore, at lower heat source temperatures, the thermosyphon containing acetone performs better than the thermosyphon containing methanol (Table 4 and Table 5).

At higher heat source temperatures, methanol performed better than acetone (Figure 6 and Table 7). At high heat source temperature, the effect of the fluid charge rate on the thermal performance of the thermosyphon is less than the effect at low heat source temperature (Figure 3 and Figure 6).

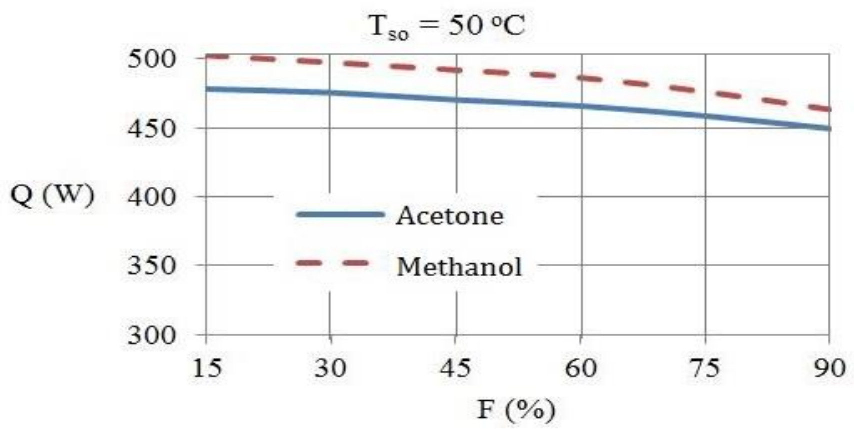

Figure 6. Effect of the fluid charge rate on thermosyphon performance

As can be seen in Figure 7, the water outlet temperature does not change much as the fluid charge rate increases. In addition, the saturation temperature of the fluid cannot exceed the heat source temperature. Consequently, at high heat source temperatures, the effect of the fluid charge rate is small. A similar situation can be seen in Figure 8 for methanol.

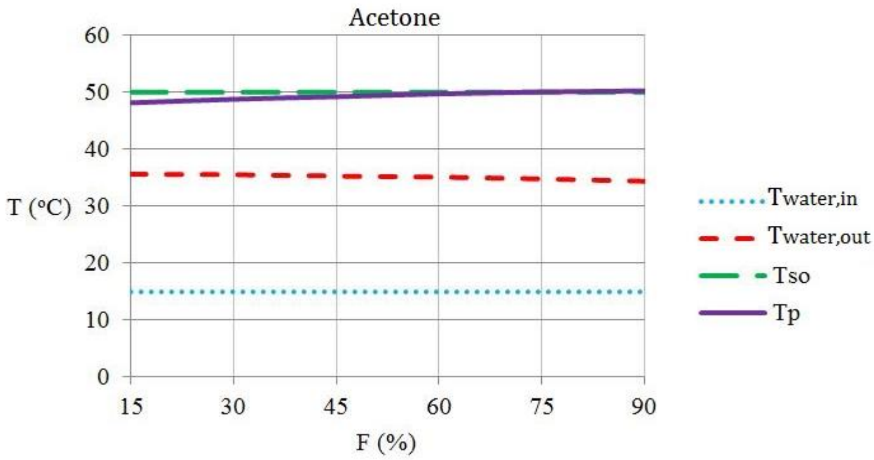

Figure 7. Influence of the fluid charge rate on $T_{water,out}$ and T_p

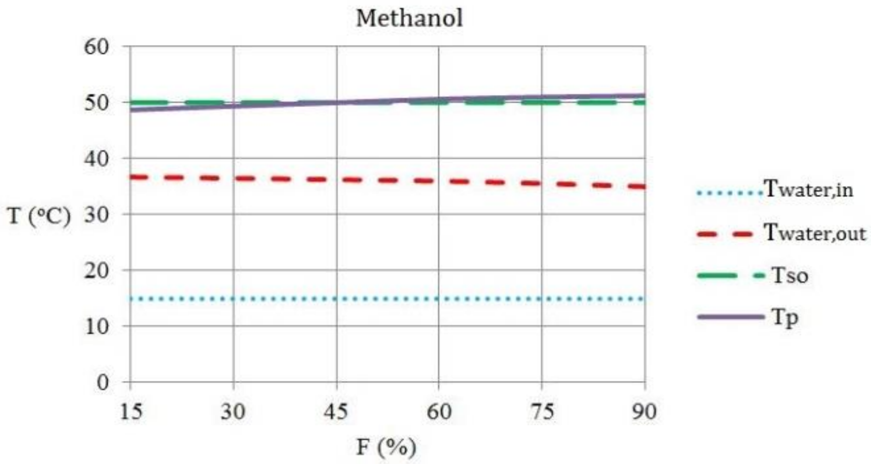

Figure 8. Influence of the fluid charge rate on $T_{water,out}$ and T_p

Table 7. Effect of the fluid charge rate on the thermal performance of the thermosyphon

Heat source temperature	**Working fluids**	**Heat flux of thermosyphon (W)**					
		%15	**%30**	**%45**	**%60**	**%75**	**%90**
50 ^oC	Acetone	478	476	471	466	459	450
50 ^oC	Methanol	502	497	492	486	476	463

Table 8. Water outlet temperature at different working fluid charge rates charge

Heat source temperature	**Working Fluid**	**Water outlet temperature (^oC)**					
		%15	**%30**	**%45**	**%60**	**%75**	**%90**
50 ^oC	Acetone	35.64	35.53	35.31	35.11	34.80	34.40
50 ^oC	Methanol	36.70	36.48	36.24	36.00	35.56	35.00

4. CONCLUSIONS

Consequently, the fluid charge rate for both fluids was found to be detrimental to the performance of the thermosyphon. Fluid’s fill rate is more influential on the thermal performance of the thermosyphon, especially at low temperatures. The reason for this is that the saturation temperature of the fluid rises above the heat source temperature. In this study, it was investigated how the thermal performance of the thermosyphon with acetone or methanol changes at different fluid charge rates at low and high heat source temperatures. The thermosyphon containing acetone at lower heat source temperatures perform better than the thermosyphon containing methanol. Thermosyphon containing methanol performed better than thermosyphon containing acetone at high heat source temperatures.

DECLARATION

This study is an extension of "Investigating the Effect of Fluid Charging Rate on the Thermal Performance of Two-Phase Closed Thermosyphon" presented in the 1st International Advanced Research and Engineering Congress 2017 (IAREC 2017).

NOMENCLATURE

A	Cross-sectional area of the thermosyphon, [m^2]
C	Equation constant, [-]
C_{pl}	Liquid evaporation latent heat, [J/kg K]
D_c	Heat exchanger diameter in condenser, [m]
D_h	Hydraulic diameter, [m]
D_i	Inner diameter of the thermosyphon, [m]
D_o	Thermosyphon's outer diameter, [m]
F	Filling rate, [%]
g	Gravity acceleration, [m/s^2]
h_{co}	Heat transfer coefficient of condenser exterior surface, [W/m^{2}K]
h_{eo}	Heat transfer coefficient of evaporator exterior surface [W/m^{2}K]

h_{fg}	Specific latent heat of vaporization, [J/kg]
k_{water}	The thermal conductivity of water, [W/mK]
L_a	Adiabatic section length, [m]
L_c	Condenser length, [m]
L_e	Evaporator length, [m]
Nu_{water}	Nusselt number of water, [-]
Pa	Atmospheric pressure, [Pa]
P_p	Hydrostatic pressure, [Pa]
Pr_{water}	Prandl number of water, [-]
P_v	Steam pressure, [Pa]
Q	Heat flux, [W]
Re	Reynolds number for thermosyphon, [-]
Re_{water}	Reynolds number of water, [-]
S_{co}	Condenser outer surface area, [m^2]
S_{eo}	Evaporator outer surface area, [m^2]
T_{co}	Condenser outer surface temperature, [^oC]
T_{eo}	Evaporator outer temperature, [^oC]
T_{si}	Heat sink temperature, [^oC]
T_{so}	Heat source temperature, [^oC]
T_p	Saturation temperature, [^oC]
T_v	Steam temperature, [^oC]
$T_{water,in}$	Water inlet temperature, [^oC]
$T_{water,out}$	Water out temperature [^oC]

t_x	Wall thickness of the thermosyphon material, [m]
V	Volumetric flow rate, [lt/h]
V_l	Volume of fluid, [m^3]
V_{water}	Velocity of water, [m/s]
β	Angle of the heat pipe with horizontal plane, [^{o}]
μ_l	Liquid dynamic viscosity, [N s/m^2]
ΔT_h	Hydrostatic temperature difference, [^{o}C]
ρ_l	Liquid density, [kg/m^3]
ρ_v	Steam density, [kg/m^3]
λ_l	Liquid thermal conductivity, [W/m K]
λ_x	Heat pipe material thermal conductivity, [W/mK]
σ	Surface tension, [N/m]
Φ	Merit number, [-]
ϑ_{water}	Kinematic viscosity of water, [m^2/s]

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