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Experimental study of thermal performance and pressure differences of different working fluids in two-phase closed thermosyphons usıng solar energy

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Experimental Study of Thermal Performance and Pressure Differences of Different Working Fluids in Two-phase Closed Thermosyphons Using Solar Energy

Araştırma Makalesi / Research Article

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ÖZ

Farklı çalışma akışkanları ile doldurulan yerçekimi destekli ısı borusunun basınç dağılımı ve termal performansının araştırılması için deneysel bir çalışma yapılmıştır. Farklı kaynama noktasına, yoğunluğa ve viskoziteye sahip olan metanol, su ve Mono-Etilen-Glikol (MEG), çalışma sıvıları olarak seçilmiştir. Üç ısı borusu için deneysel bir düzenek tasarlanmış ve buharlaştırıcı bölümünün ısıtılması güneş enerjisi ile sağlanmıştır. Isı borusunun basınç ve sıcaklık dağılımları için boru yüzeyineden gerekli ölçümler yapılmıştır. Yüksek kaynama noktasına sahip saf antifriz ile termosifon tipi ısı borularında yaygın olarak kullanılan su ve metanol çalışma akışkanı olarak seçilmiştir. Isı boruları, yüksek sıcaklıklar elde etmek için parabolik odaklı vakumlu cam tüp içine konulmuş ve 11 gün boyunca güneş enerjisi ile çalıştırılmıştır. Deney sonuçları için saat 06.30 ile 18.30 arasında alınan veriler dikkate alınmıştır. Deney sonuçları, en yüksek ve en düşük güneş ışınımına sahip (i) güneşli ve (ii) bulutlu günler seçilerek değerlendirilmiştir. (i) Güneşli günlerde, metanol, su ve antifriz için en yüksek depolama suyu sıcaklıkları 90.0°C, 83.5°C ve 86.7°C, ve basınç değerleri sırasıyla 6.7 bar, 1.6 bar ve 1.9 bar şeklinde gerçekleşmiştir. (ii) Bulutlu gün için ise, ilgili sıcaklık ve basınç değerleri; 43.9°C, 39.3°C, 29.0°C ve 2.0 bar, 0.2 bar ve -0.3 bar olmuştur.

Anahtar Kelimeler: Isı borusu, güneş enerjisi, ısıtma, performans.

Experimental Study of Thermal Performance and Pressure Differences of Different Working Fluids in Two-phase Closed Thermosyphons Using Solar Energy

ABSTRACT

An experimental study is carried out to investigate the pressure distribution and thermal performance of gravity assisted heat pipe charged with different working fluids. Methanol, water and Mono-Ethylene-Glycol (MEG) are chosen as working fluids which have different boiling point, density and viscosity. An experimental test apparatus is designed and produced including three heat pipes that heat input on the evaporator section are provided by solar energy. Measurements are conducted on the heat pipe surface for pressure and temperature variations. Pure antifreeze is chosen as working fluid, due to its high boiling point, along with water and methanol which are widely used in thermosyphon type heat pipes. Heat pipes are put into parabolic focused vacuumed glass tube and operated by solar energy for 11 days in order to achieve high temperatures. Data taken from 06.30 to 18.30 is taken into consideration for experimental results. Experimental results are evaluated for two days as (i) sunny and (ii) cloudy days by choosing days with the highest and lowest solar radiation. (i) As for the sunny day, the highest storage water temperatures for methanol, water and antifreeze were 90.0°C, 83.5°C and 86.7°C and the pressure values were 6.7 bar, 1.6 bar and 1.9 bar respectively. (ii) And for the cloudy day, the respective values of temperature and pressure were measured as; 43.9°C, 39.3°C, 29.0°C, and 2.0 bar, 0.2 bar and -0.3 bar.

Keywords: Heat pipe, solar energy, heating, performance.

1. INTRODUCTION

70% of the energy in the world is consumed as heat. It is important to shrink the carbon footprint by reducing the energy use and the irreversibilities due to heating and friction, because the energy costs are going up continuously. This reduction can be obtained by using heat exchangers or by heat pipes [1,2]. Heat pipe is a heat transfer device that allows a high heat transfer between the evaporator and condenser during the liquid-vapor

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phase change of the fluid [3]. A heat pipe can transfer the heat from one end to the other many times faster than a metal bar with the same material and size [4]. Heat pipes are used in electronic devices, in hospitals, hotels etc. in heating and ventilation (HVAC) systems, in space and nuclear technologies as a completely natural cooling apparatus and in some other industrial sectors [3].

Heat pipes have advantages of high thermal conductivity and a high heat flux. A high temperature difference between the heat source and the heated place is not needed for an effective heat transfer with heat pipes [5]. Transfer of the working fluid from the evaporator to the condenser is driven by the vapor pressure and becomes in vapor phase. Fluid condensed by phase change in the condenser is transferred to the evaporator by an installed wick or by the gravity [6]. Heat pipes are the best solutions that can be chosen for the thermal control by their affordable costs and excellent heat transfer abilities. Various types of heat pipes are used in thermal engineering applications [7]. One of them is gravity driven heat pipes. This type of heat pipes are called two-phase closed thermosyphon (TPCT). There is no capillary structure in this kind of devices and the heat transfer is accomplished by transferring the latent heat of evaporation and condensation through a two-phase closed cycle [8]. In the TPCT type, vapor pressure drives the motion from the evaporator to condenser, while it returns back from condenser to evaporator by the gravity [9,10].

Schematic in Fig.1 shows the physical mechanism of the TPCT. Basically they have three main parts as evaporator section, condenser section and adiabatic section. Heat flow through the pipes defines the performance of TPCT [8]. In such kind of devices, liquid working fluid in the evaporator is heated by the heat at the outer surface of the evaporator. The liquid fluid reaches to the boiling point by this heat and goes to the condenser forced upward by the pressure. Working fluid in vapor phase touches to the pipe surface which is colder than the fluid and leaves its latent heat to the pipe. Working fluid begin to condense as it loses heat. Condensed fluid takes the shape of a film membrane due to the low vapor speed and goes downward to the evaporator by the gravity [11].

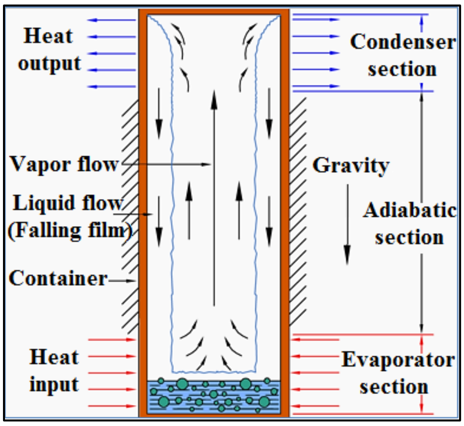


Figure 1. Physical mechanism of the TPCT [12].

There are many studies in the literature on TPCTs which are on the use of different working fluid, on filling rate, and on the tilt angle. Many of them use electrical energy and hot gases while some others use solar energy as the heat source [13]. Some of the studies are as follows;

Abreu and Colle, investigated the thermal behavior of TPCTs experimentally using electrical energy. They analyzed the effects of evaporator length, filling rate, cooling temperature and tilt angle on the performance [14].

W. Chun et al. compared wicked and thermosyphon type heat pipes with different working fluids. They used water, methanol, acetone and ethanol as the working fluid in the experiments and they also compared the collector surface areas and storage tank capacities [15]. Esen and Esen, used R134a, R407C, R410A as the working fluid in TPCT for sunny days using solar energy. They obtained the best performance with the heat pipe using R410A as the working fluid [16]. Nuntaphan et al. investigated the critical heat flux in the thermosiphon type heat pipe using triethylene glycol (TEG)-water solution [17]. Jahanbakhsh et al. experimentally investigated the effects of tilt angle and using water-ethanol solution in TPCT type heat pipes on the thermal performance with both electrical and solar energy [4].

Although the effect of various parameters such as inclination angle, filling ratio, different working fluid on the thermal performance of TPCT has been studied theoretically and experimentally in the literature, there are few papers examining the pressure variation in solar assisted heat pipes.

The main objective of the present study is to investigate not only the temperature values of the working fluid in the gravity assisted solar heat pipe but also the pressure values. Thus, we will have an idea of how much the working fluid evaporates and contributes to the heat transfer. For this aim an experimental apparatus was designed and built. Obtained results from experimental test were discussed in detail using graphical illustration. The evaporation time and amount for the working fluid is examined by measuring not only the temperature distribution but also the pressure distribution in the heat pipe.

2. MATERIALS AND METHOD

2.1. Experimental system

In the present study, the thermal performance and pressure change for various types of working fluids in two-phase closed thermosyphons were investigated experimentally by using heat input as solar energy. Methanol, water and Mono-Ethylene-Glycol (MEG) with different boiling point, liquid density and liquid viscosity were used as the working fluid. Table 1 gives some important physical properties related to the working fluid used. Evacuated glass tubes are commercially available with 1800 mm length and 37 mm inner diameter. Therefore the heat pipe is made of steel pipe with 28 mm of outer diameter, 1.5 mm of wall thickness and 2200 mm of length. 1800 mm of the pipe inside the glass tube is designed as the evaporator region, the next part of 200 mm length is adiabatic region and the rest 200 mm is the condenser region. In the condenser region, a heat exchanger with another steel pipe with 42.4 mm diameter, 3.6 mm thickness and 200 mm length as shown in Figure 2. The heat pipe is aimed to heat a total of 3.5 L water including the cubic storage tank and the heat exchanger. All three systems are identical in shape, size and material using only different working fluids.

Table 1. Physical properties of the working fluids at 20°C [18, 19].

Fluid	Boiling Point (1 atm) (°C)	Liquid Density (ρ) (kg/m^3)	Liquid Viscosity (N-s/m^2)
Methanol	64.7	791.5	0.587x10^{-3}
Water	100.0	1000.0	1.002x10^{-3}
Mono Ethylene Glycol (MEG)	197.6	1113.5	19.800x10^{-3}

Working fluid is charged into the heat pipe by two ways. (i) first the inside air is vacuumed and then the fluid is filled. (ii) Pipe is filled with working fluid and then the pipe is heated to the boiling point of the fluid, and the open end of the pipe is closed [2]. In this study, the first method is applied and one third of the 1800 mm heat pipe inside the glass tube is charged with 295 ml, 100% purity methanol, distilled water and 100% purity MEG. Fig.2 shows the locations of temperature and pressure measurements required for the thermal performance analyses.

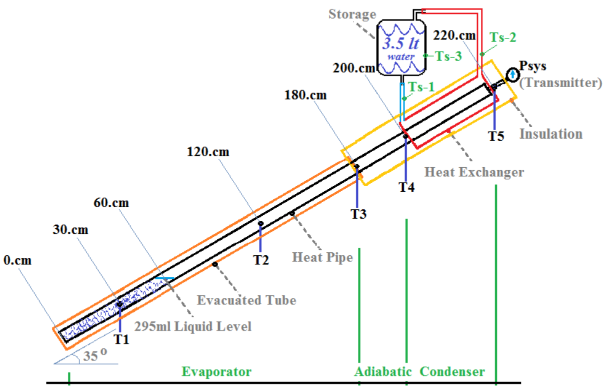
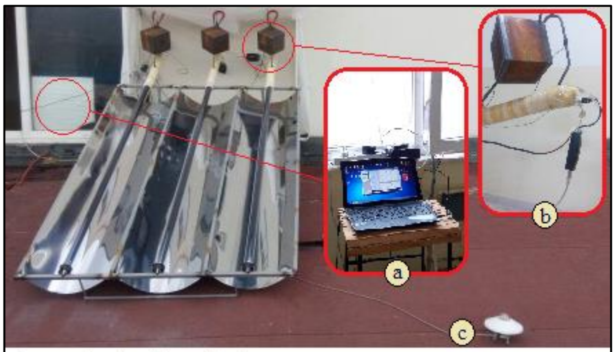


Figure 2. Schematic diagram of the system

All temperature measurements are conducted by K-type thermocouples placed at appropriate locations. DeltaOhm LP PYRA 02 pyranometer is used for measuring incident solar radiation and Keller PA-21Y pressure transmitter with ± 0.25 %FS linearity was used for pressure measurements. All experimental data is gathered at five-minute intervals and recorded by ORDEL UDL100 universal Data Logger with 0.2% accuracy.

Experimental setup in Fig.3 was designed and produced, and tested for 11 days at the same time and under the prevailing weather conditions of Samsun, on the north coast of Turkey with the exact location 41°14’N 36°26’E with sea level. Data measured from 05:30 to 19:30 is taken into consideration as the experimental results. Days with the maximum and minimum solar radiation incidents are chosen for the consideration of the experimental results for (i) sunny days and (ii) cloudy days.



a) computer for data collection process
b) data logger for temperatures and pressures c) pyranometer

Figure 3. Experimental setup

2.2. Theoretical calculations

The thermal resistance and efficiency calculations of heat pipes using the data obtained from the experiments are determined using the following equations:

The temperature difference between the condenser and the evaporator reveals the relationship between the working fluid in the heat pipe and boiling point, density and viscosity of the working fluid. ΔT is the difference between the mean evaporator temperature and the mean temperature of heat pipe condenser and can be calculated by:

$$\Delta T = (T1 + T2 + T3)/3 - (T4 + T5)/2 \tag{1}$$

The total amount of solar energy incident falling onto the heat pipe during the experiment can be calculated by heat pipe surface area and the amount of global radiation.

$$Q_{incident} = A\int_1^2 I dt \tag{2}$$

Where, A is vacuumed glass tube surface area (m^2), I is the global irradiance (W/m^2). The surface area of vacuumed tube used in the experiment is calculated as 0,1332 m^2.

The thermal resistance of the heat pipes according to working fluids can be calculated by;

$$R = \Delta T / Q_{incident} \tag{3}$$

The amount of heat transferred from heat pipe to storage can be calculated using temperature difference in the storage, amount of water and specific heat of water;

$$Q_{col} = m_w c_{p,w}(T_{w2} - T_{w1}) \quad (4)$$

where, m_w is the amount of storage water (kg), $c_{p,w}$ is the specific heat of water (kJ/kg^{o}C), T_{w1} is the initial water temperature according to the measurement range (^{o}C), T_{w2} is the last temperature (^{o}C).

The efficiency (η) can be calculated as the ratio of heat transferred from the heat pipe to the storage (Qcol) to the total amount of solar energy incident falling onto the heat pipe (Qincident);

$$\eta = Q_{col}/Q_{incident} \quad (5)$$

3. RESULTS AND DISCUSSION

Experiments started on June 26 and continued for 11 days until July 06. Data measured from 06:30 to 18:30 is used for comparing the thermal performance and pressure values of the heat pipes. Fig.4 gives the distribution of solar radiation for 11days. According to the mean solar radiation values the highest mean solar radiation was on June 29, and the lowest one was on July 02. Thermal performance and pressure values of the heat pipes according to the working fluid were investigated based on these two days which were chosen as Sunny (June 29) and Cloudy (July 02) days.

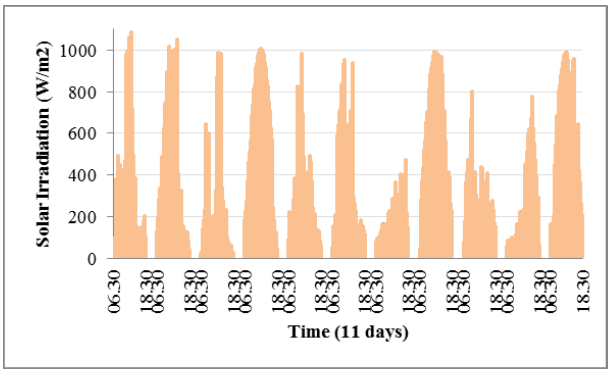


Figure 4. Distribution of solar radiation for 11 days

Fig.5 shows the pressure distribution in the heat pipes for the sunny day. The highest pressure value was obtained by the heat pipe using methanol as the working fluid which has a low boiling point. The peak pressure values for methanol, water and MEG are measured as; 6.7bar, 1.6bar and 1.9bar for sunny day.

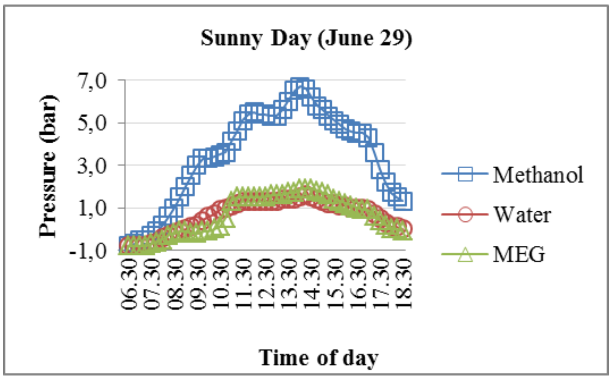


Figure 5. Distribution of pressure for sunny day

Figures 6, 7 and 8 show the temperature distributions of the measurement points of methanol, water and MEG heat pipes, respectively. T1, T2 and T3 are the temperature measurement points of the evaporator region of the heat pipe. As can be seen from figures 6, 7 and 8, the most homogeneous temperature distribution in the evaporator region is in the methanol-heat pipe. Because of the low boiling point, it can be said that the entire amount of methanol contributes to heat transfer. Since MEG has the highest boiling point, it is understood that the temperature difference between points T1, T2 and T3 is higher. The inlet and outlet temperature values of the condenser zone are defined by T4 and T5. It is understood that the heat transfer from T3 to T4 is good for all three heat pipes. It is seen that the T5 temperature is over 80^{o}C in the methanol-heat pipe, while it is below 80^{o}C in the water and MEG heat pipes.

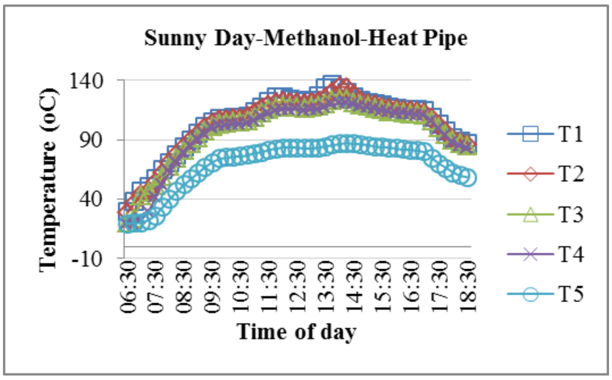


Figure 6. Distribution of methanol-heat pipe temperatures for sunny day

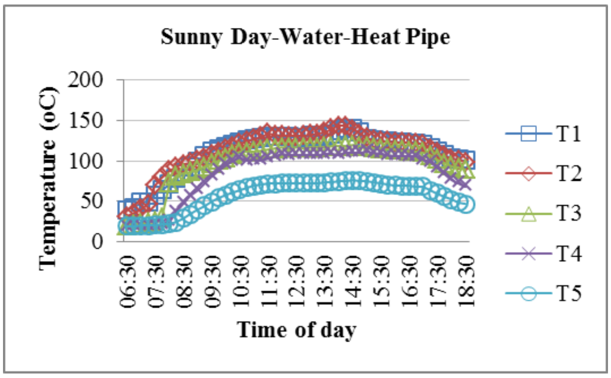


Figure 7. Distribution of water-heat pipe temperatures for sunny day

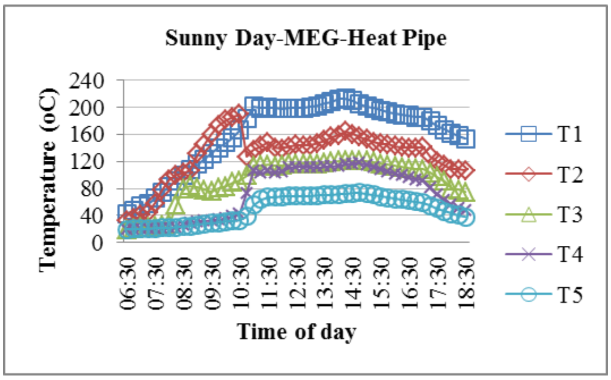


Figure 8. Distribution of MEG-heat pipe temperatures for sunny day

Figures 9, 10 and 11 show the distributions of temperature measurement points of storages for methanol, water and MEG, respectively. Ts-1 indicates the condenser inlet and storage outlet temperature point, and Ts-2 indicates the condenser outlet and storage inlet temperature point. The storage temperature measurement point is Ts-3. While the highest water temperature (Ts-3) in the reservoirs was 97.1ºC, 88.6ºC and 89.0ºC for methanol, water and MEG heat pipes, respectively, the average maximum temperatures in the reservoirs reached 90.0ºC, 84.6ºC and 86.7ºC.

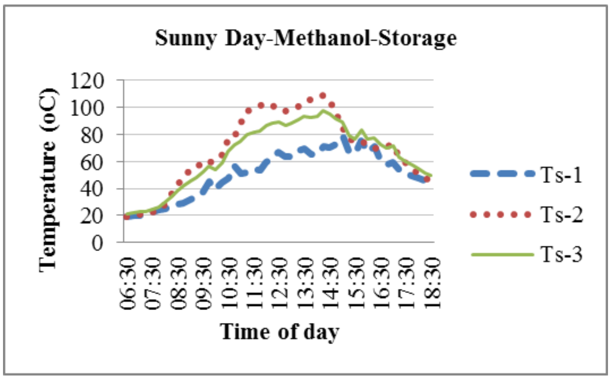


Figure 9. Distribution of methanol-storage temperatures for sunny day

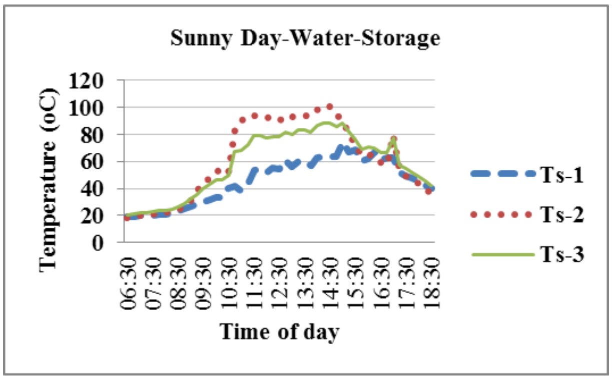


Figure 10. Distribution of water-storage temperatures for sunny day

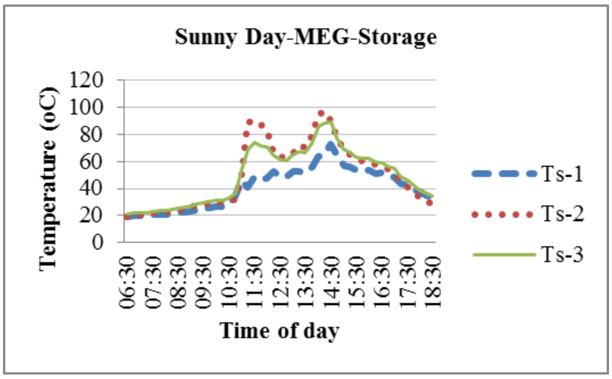


Figure 11. Distribution of MEG-storage temperatures for sunny day

The pressure distributions of the heat pipes for the cloudy day are given in Figure 12. High pressure values have not been reached since there is not enough solar radiation intensity. After then 14:00 o'clock, with the decrease of cloudiness, methanol with low boiling point has started to evaporate and pressure increase has occurred. The peak pressure values for methanol, water and MEG heat pipes were measured as 2.0bar, 0.2bar and -0.3bar, respectively.

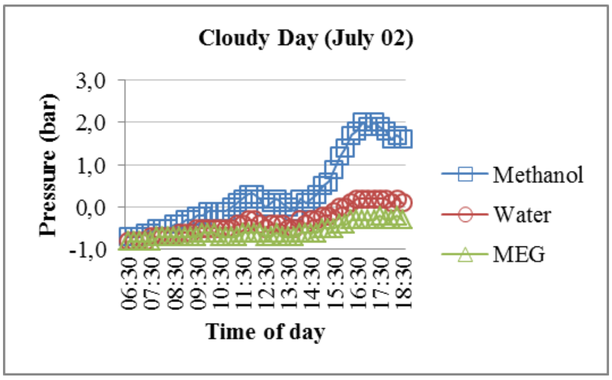


Figure 12. Distribution of pressure for cloudy day

Figures 13, 14 and 15 show the temperature distribution of the points (T1, T2, T3, T4 and T5) of methanol, water and MEG heat pipes, respectively. When Fig.13 is examined, it appears that evaporation in methanol immediately begins because it has a low boiling point. Despite the fact that the air does not turn on, the temperature at T5 has risen to 40ºC. With the opening of the air it is seen that this value has reached over 60ºC.

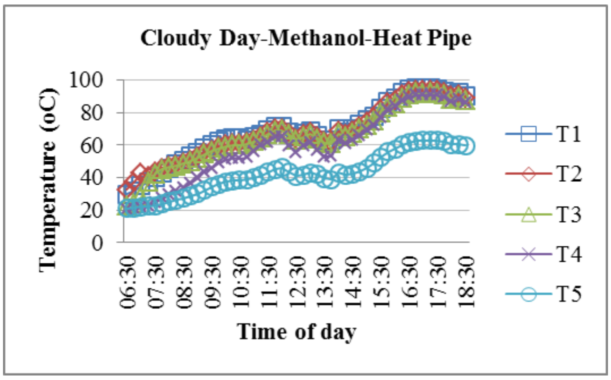


Figure 13. Distribution of methanol-heat pipe temperatures for cloudy day

In Fig.14, it is understood that the pressure in the water-heat pipe rises above 0 and the evaporation begins at the same time. However, due to the high boiling point (100ºC), it is clear from the temperature of T5 that it cannot make a full contribution to heat transfer. Partly with the increase in solar (after 14:00 o'clock), T5 temperature was reached about 50ºC.

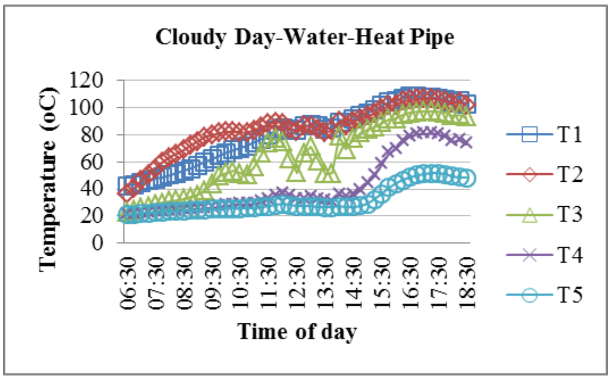


Figure 14. Distribution of water-heat pipe temperatures for cloudy day

It is understood from the pressure values and Fig.15 that there is almost no evaporation in the cloudy day-long experiment of the MEG-heat pipe. Therefore, there was no significant increase in T4 and T5 temperature values and were remained around 25-35ºC.

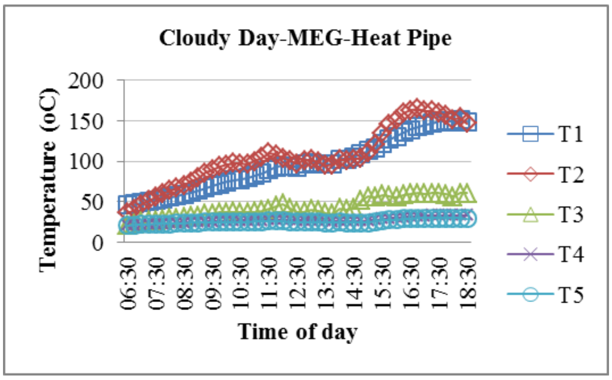


Figure 15. Distribution of MEG-heat pipe temperatures for cloudy day

Figures 16, 17 and 18 show the distributions of the storage temperatures (Ts-1, Ts-2 and Ts-3) for methanol, water and MEG, respectively. As seen by the temperature distribution of the heat pipe, the MEG-tank was nearly unheated. Although in water-storage temperature (Ts-3) increased above 40ºC, the best heating occurred at methanol-storage temperature (approx. 50ºC).

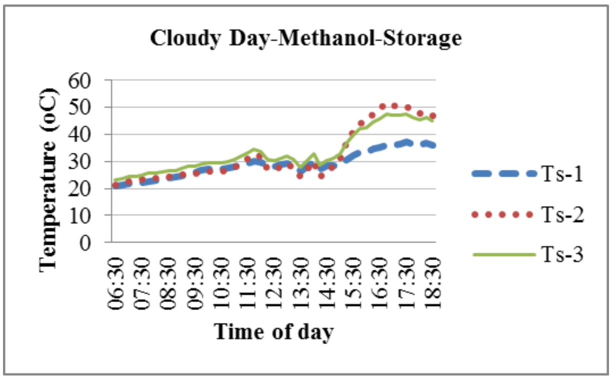


Figure 16. Distribution of methanol-storage temperatures for cloudy day

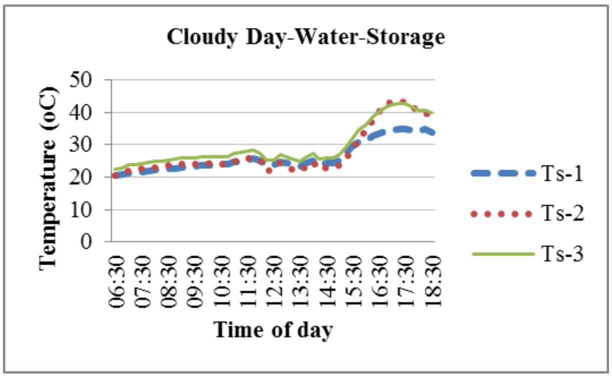


Figure 17. Distribution of water-storage temperatures for cloudy day

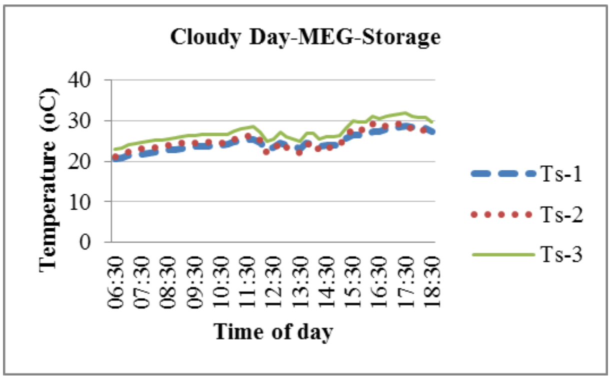


Figure 18. Distribution of MEG-storage temperatures for cloudy day

It is not the right approach to do about efficiency and thermal resistance calculations for cloudy days. Therefore, efficiency and thermal resistance calculations have been done only for sunny days. Figure 19 shows the efficiency distribution of heat pipes. As can be understood from the graph, the efficiency decreases as the difference between the heat pipe temperatures (T1, T2, T3, T4 and T5) and the storage temperatures (Ts-1, Ts-2 and Ts-3) increases. The highest storage temperatures were reached at between 14:10 and 14:20. The efficiency for methanol, water and MEG calculated 0.37, 0.33 and 0.35, respectively.

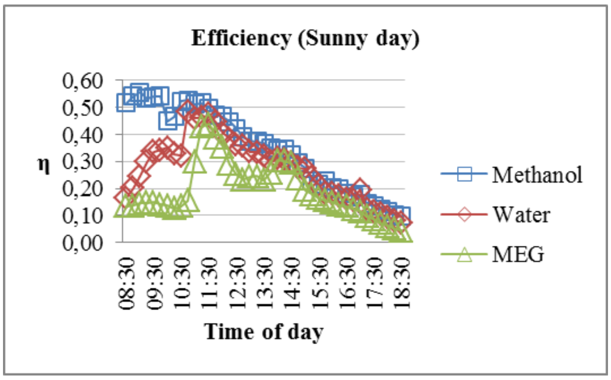


Figure 19. Distribution of efficiency of heat pipes for sunny day

As the temperature difference between the evaporator region (T1, T2 and T3) and the condenser region (T4 and T5) increases, the thermal resistance increases. Therefore, a low and stable distribution of the thermal resistance is an indication that the heat pipe fluid works well. Fig.20 shows that the best working fluid is methanol. It is seen that the fluid that reaches the latest stabilize and has the highest thermal resistance is MEG.

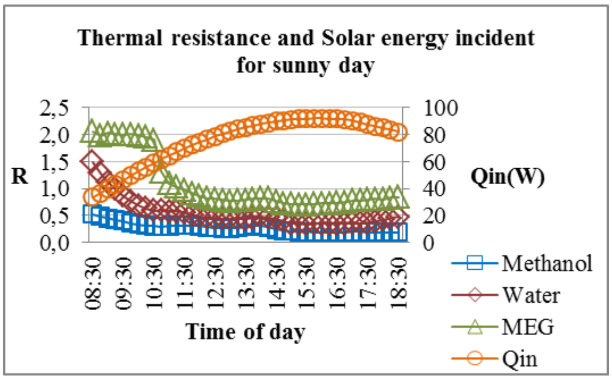


Figure 20. Distribution of thermal resistance of heat pipes for sunny day

4. CONCLUSION

The aim of this study is to compare the thermal performance and pressure changes of heat pipes by solar energy. For this, methanol, water and MEG were chosen as working fluids which have different boiling point, density and viscosity. Identical heat pipes were manufactured and tested for 11 days at the same conditions inside parabolic trough collector to reach high temperatures. Temperature and pressure changes of the working fluid were investigated and compared for both sunny and cloudy days and their thermal performance has been examined.

Examining the pressure values along with temperatures made it possible to understand the heat transfer between the measurement points better. The temperature difference between the bottom (T1) and the top (T5) of the heat pipe were reached 49.1oC, 63.8oC and 139.4oC for methanol, water and MEG, respectively. So we can say that, in terms of temperature, pressure and heat transfer, methanol is the most and MEG is the least active fluids.

According to the pressure and temperature values, the best result for both cloudy and sunny days was obtained with methanol. Methanol may be preferred in solar water heating systems where high temperatures are not desired.

Pressure measurement in heat pipes gave us an idea of the amount of fluid that evaporates. In special work requiring high temperature, the amount of MEG in the heat pipe can be reduced and more evaporation can be achieved. Thus, much higher temperature values can be achieved in the condenser region.

REFERENCES

[1] Leriche M., Harmand S., Lippert M., Desmet B., “An experimental and analytical study of a variable conductance heat pipe: Application to vehicle thermal management”, ***Applied Thermal Engineering***, 38: 48-57, (2012).

[2] Chan C.W., Siqueiros E., Ling-Chin J., Royapoor M., Roskilly A.P., "Heat utilization technologies: A critical review of heat pipes", ***Renewable and Sustainable Energy Reviews***, 50: 615-627, (2015).

[3] Zohuri B., Heat Pipe Design and Technology, CRC Press (2011).

[4] Jahanbakhsh A., Haghgou H.R., Alizadeh S., "Experimental analysis of a heat pipe operated solar collector using water-ethanol solution as the working fluid", ***Solar Energy***, 118: 267-275, (2015).

[5] Yang X., Yan Y.Y., Mullen D., "Recent developments of lightweight, high performance heat pipes", ***Applied Thermal Engineering***, 33-34: 1-14, (2012).

[6] Vasiliev L., Vasiliev Jr L., "Sorption heat pipe-a new thermal control device for space and ground application", ***International Journal of Heat and Mass Transfer***, 48: 2464-2472, (2005).

[7] Vasiliev L.L., Kakac S., "Heat pipe and solid sorption transformation, fundamentals and practical applications", ***Taylor&Francis Group***, LLC (2013).

[8] Jafari D., Franco A., Filippeschi S., Di Marco P., "Two-phase closed thermosyphons: A review of studies and solar applications", ***Renewable and Sustainable Energy Reviews***, 53: 575-593, (2016).

[9] Franco A., Filippeschi S., "Closed loop two-phase thermosyphon of small dimensions: are view of the experimental results", ***Microgravity Science and Technology***, 24: 165-179, (2012).

[10] Filippeschi S., "Comparison between miniature periodic two-phase thermosyphons and miniature LHP applied to electronic cooling equipment", ***Applied Thermal Engineering***, 31: 795-802, (2011).

[11] Reay D.A., Kew P., "Heat pipes (5th edition)", Oxford, UK, Butterworth-Heinemann, (2006).

[12] Faghri A., "Heat Pipe Science and Technology", ***Taylor & Francis***, Washington, DC, (1995).

[13] Gedik E., Yılmaz M., Kurt H., "Experimental investigation on the thermal performance of heat recovery system with gravity assisted heat pipe charged with R134a and R410a", ***Applied Thermal Engineering***, 99: 334-342, (2016).

[14] Abreu S.L., Colle S., "An experimental study of two-phase closed thermosyphons for compact solar domestic hot-water systems", ***Solar Energy***, 76: 141-145, (2004).

[15] Chun W., Kang Y.H., Kwak H.Y., Lee Y.S., "An experimental study of the utilization of heat pipes for solar water heaters", ***Applied Thermal Engineering***, 19: 807-817, (1999).

[16] Esen M., Esen H., "Experimental investigation of a two-phase closed thermosyphon solar water heater", ***Solar Energy***, 79: 459-468, (2005).

[17] Nuntaphan A., Tiansuwan J., Kiatsiriroat T., "Enhancement of heat transport in thermosyphon air preheater at high temperature with binary working fluid: A case study of TEG–water", ***Applied Thermal Engineering***, 22: 251-266, (2002).

[18] Yue H., Zhao Y., Ma X., Gong J., "Ethylene glycol: properties, synthesis, and applications", ***Chemical Society Reviews***, 41: 4218-4244, (2012).

[19] Peyghambarzadeh S.M., Shahpouri S., Aslanzadeh N., Rahimnejad M., "Thermal performance of different working fluids in a dual diameter circular heat pipe", ***Ain Shams Engineering Journal***, 4: 855-861, (2013).