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Experimental and Theoretical Study of the Performance of Solar Water Heaters with Gravity-Assisted Heat Pipes

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ABSTRACT :

The objective of this work is to study the performance of solar water heaters using heat pipes for heat transfer, from absorber plate to water tank, with two different types of working fluids (Ethanol and Acetone) and different numbers of heat pipes. Additionally, a theoretical investigation is conducted to predict the performance of the solar water heaters. These systems have been designed and fabricated with the same dimensions and materials. The hourly variation of the absorber plate temperature, the water storage tank temperature and solar radiation intensity are measured. Accordingly, the stored energy and the efficiency has been calculated. The results showed that, a good agreement has been achieved between the experimental and theoretical results. The operation point of the conventional system begins at the operating start while the operation point of the thermosyphon systems starts when the absorber plates temperatures reaches the boiling point of the working fluid. The maximum water temperatures gained of the forced convection system and thermosyphon system charged with ethanol and acetone were 66 °C, 67.8 °C, and 64.6 °C, respectively while those of theoretical calculations were around 69.2 °C, 69 °C and 69.3 °C, respectively. In addition, the maximum value of the actual efficiency of conventional system was ranged between 47~53% while that of theoretical calculation was between 50~53%. The experimental maximum efficiency of thermosyphon systems was around 55% at the case of installing 14 heat pipes charged with acetone and theoretical efficiency was 55% for same the case. The thermosyphon system gives better performance than forced convection system, because thermosyphon system does not consume electricity. Performance of the thermosyphon system using acetone as working fluid was better than the performance of the thermosyphon system using ethanol and the increase of the heat pipes number improves performance.

Keywords**:** Solar water heater, flat plate solar collector, forced convection water heater, heat pipes water heater, theoretical model.

ملخص البحث:

الهدف من هذا العمل هو دراسة أداء سخانات المياه بالطاقة الشمسية باستخدام الأنابيب الحراريه لنقل الحرارة من لوحة الامتصاص إلى خزان المياه ، مع نوعين مختلفين من سوائل التشغيل (إيثانول وأسيتون) وأعداد مختلفة من الأنابيب الحراريه. بالإضافة إلى ذلك ، يتم إجراء دراسة نظرية للتنبؤ بأداء سخانات المياه بالطاقة الشمسية. تم تصميم هذه الأنظمة وتصنيعها بنفس الأبعاد والمواد. يتم قياس التغير كل نصف ساعة في درجة حرارة لوحة الامتصاص ودرجة حرارة خزان تخزين المياه وكثافة الإشعاع الشمسي. وفقا لذلك ، تم حساب الطاقة المخزنة والكفاءة. أظهرت النتائج أنه تم التوصل إلى اتفاق جيد بين النتائج التجريبية والنظرية. تبدأ نقطة تشغيل النظام التقليدي عند بداية التشغيل بينما تبدأ نقطة التشغيل لأنظمة الانابيب الحراريه عندما تصل درجة حرارة لوحة الامتصاص إلى درجة غليان سائل العمل. كانت درجات الحرارة القصوى للمياه المكتسبة من النظام التقليدي ونظام الانابيب الحراريه المشحون بالإيثانول والأسيتون ٦٦ درجة مئوية و ٦٧.٨ درجة مئوية و ٦٤.٦ درجة مئوية على التوالي ، بينما كانت درجات الحرارة النظرية حوالي ٦٩.٢ درجة مئوية و ٦٩ درجة مئوية و ٦٩.٣ درجة مئوية على التوالي. بالإضافة إلى ذلك ، تراوحت القيمة القصوى للكفاءة الفعلية للنظام التقليدي بين ٤٧ ~ ٥٣ ٪ بينما كانت القيمة النظرية تتراوح بين ٥٠ إلى ٥٣ ٪. بلغت أقصى كفاءة تجريبية لأنظمة الانابيب الحراريه حوالي ٥٥٪ في حالة تركيب ١٤ أنبوبًا حراريًا مشحونًا بالأسيتون ، وكانت الكفاءة النظرية ٥٥٪ لنفس الحالة. وأستنتج أن أداء نظام الانابيب الحراريه كان أفضل من النظام الحمل الحراري القسري لأن نظام الانابيب الحراريه لا يستهلك الكهرباء، و كان أداء نظام الانابيب الحراريه باستخدام الأسيتون كمائع عامل أفضل من أداء نظام الانابيب الحراريه باستخدام الإيثانول وزيادة عدد الأنابيب الحراريه يحسن من الأداء.

1- INTRODUCTION

People are always looking for new sources of energy to cover their growing needs in their life applications. The classical types of energy as oil and natural gas will not be capable to supply this demand in a sustainable way. These sources of energy will be depleted, as also, the cost of their exploitation is high. Besides, the traditional fuels are main sources of environmental pollution. All these reasons have been made renewable energy as the best option at all. Solar energy is the most abundant of all renewable energy resources and the oldest energy source that human used through time. It is among the modern direct use for solar energy the most popular and its simplest is heating water.

The flat plate collector with heat pipes or thermosyphon system is one from different design models for water heater systems. The difference between heat pipes and thermosyphon is that the evaporator section is at higher level with respect to condenser in the case of heat pipe while it is at lower level in the case of thermosyphon. Therefore, no need of wick material in case of thermosyphon as the liquid moves from the condenser to the evaporator by gravity assist. A heat pipes system has several advantages as there working as thermocouples, it stops work when the storage tank temperature is higher than the evaporation temperature of the work fluid thus the system stops. In addition, it is suitable for all climatic conditions, they are the preferred choice for very cool areas where they can eliminate of freezing and interior corrosion by selecting a suitable working fluid. No electrical power is required for these systems.

Many researches have been focused on studying the performance of heat pipes or thermosyphon solar collectors. Azad [1] conducted theoretical and experimental investigation on the performance of heat pipe solar collector. The heat pipe was made of copper and the evaporator section was finned with aluminum plates. The theoretical model was established to predict the outlet water temperature from heat exchanger, heat pipe temperature, and the thermal efficiency of solar collector. Tong et al. [2] presented a theoretical investigation of the thermal performance of evacuated heat pipe solar collector and compared to experimental results. A discrepancy of about 0-6% was observed due to the simple assumptions in the simulation process. The test efficiency was found to be in the range of 40%-60%, which is lower than the values predicted by modeling. The agreement between the measured and calculated storage temperatures was very good, which was just within 2.5%. Thaib et al. [3] studied the performance of solar water heating system used paraffin wax as working fluid. Experimental results indicated that using paraffin wax could able to increase efficiency of solar water heating systems. The maximum efficiency of a collector solar water heater was 36.6%. Taoufik et al. [4] presented a parametric study of a flat plate heat pipe collector with a single glass cover uses wick as an assist in the heat pipes. The parameters those affect the collector efficiency are identified, such as gap spacing between the absorber plate and the glazing cover and the emissivity of the absorber plate. Three different fluids were compared and the results showed that acetone performs better than methanol and ethanol. Hussein et al. [5] presented an experimental investigation for wickless heat pipe solar collector with three different cross sections, charged with three different distilled water filling ratios; 10%, 20% and 35%. Another group of researchers [6-8] studied improvements in the design of a flat plate solar collector with heat pipes, and discussed the efficiency with these improvements. Azad [9] presented an assessment of three types of heat pipe solar collectors installed in parallel and tested at the same working conditions. In the first type, the heat-pipes were connected at their ends with two headers and the absorber plate was constructed from fins. In the second type, each heat pipe was individual installed and bonded mechanically to the aluminum absorber plate. In the third type, the heat-pipes were connected with headers at their ends and bonded mechanically to the aluminum absorber plate. All of the three types' performance were satisfactory while the first type was produces better efficiency over all the reduced temperature parameter range. Ismail and Abogderah [10] presented a comparative study between a flat-plate solar collector with heat pipes and a conventional commercial solar collector. The experimental results showed that the efficiency of the heat pipe solar collector was lower than the conventional collector in the morning as the heat pipes were operating at a temperature less than their design temperature. It was higher when the heat pipes reach their operating temperatures. Rushi et al. [11] presented experimental investigation of the performance of a heat pipe solar water heater with different working fluids. The results were compared with those of a conventional flat plate solar water heating system. The efficiency of solar water heater with heat pipe is about 7% more than the conventional water heater. Ahmet and Adnan [12] presented an experimental investigation of single-phase and two-phase closed thermosyphon solar water heater systems. In the conventional single-phase system, water is used as working fluid while in the two-phase system R-134a is used as working fluid. Experimental results showed that, the efficiency of the two-phase systems is higher by about 42% than that of the classical system. Tong and Cho [13] studied experimentally and theoretically, the thermal performance of evacuated solar collectors with U-tubes and heat pipes. The comparison showed high accordance between the simulation and experimental results. The results showed that the efficiency of the heat pipe-type was higher than the U-tube-type during the sunny day, while the U-tube-type showed steadier and better thermal performance when it was cloudy.

Based on the previous review, little attention was paid in studying the performance of solar water heaters with heat pipes charged with different working fluids. The objective of this work is to study experimentally and theoretically the performance of solar water heaters using heat pipes for heat transfer, from absorber plate to water tank, with two different type of working fluid and different numbers of heat pipe.

2- Theoretical Model:

The performance of the solar water heater with thermosyphon as a means of heat transfer has been studied theoretically with taking into account the following assumptions:

- The temperature gradient in the longitudinal direction of the collector is neglected.
- All absorbed heat by evaporator is delivered to the condenser.
- The overall heat loss coefficient between the collector and the ambient has constant value .

The solar energy absorbed by the absorber plate is distributed to useful gain and thermal losses. Heat losses of a flat plate solar collector consists of the top heat losses through cover systems, bottom and edge heat losses through bottom and edge insulation of the collector as reported by Duffie and Beckman [14]. Based on [14], the governing relations for the losses are:

$$U_l = U_t + U_b + U_e \qquad (1)$$

$$U_t = \left[\frac{N_g}{\frac{C}{T_p}\left(\frac{T_p - T_a}{N + F}\right)^e} + \frac{1}{h_w} + \right] \frac{\sigma(Tp + Ta)(T_p^{\,2} + T_a^{\,2})}{\frac{1}{\varepsilon_p + 0.0059Nh_w} + \frac{2Ng + f - 1 + 0.133\varepsilon_p}{\varepsilon_g} - N} \qquad (2)$$

$$hw = 5.7 + 3.8V \qquad (3)$$

$$f = (1+0.089h_w-0.1166h_w\varepsilon_p)(1+0.07866N) \qquad (4)$$

$$C = 520(1-0.000051\beta^2) \qquad (5)$$

$$e = 0.43(1-100/T_p) \qquad (6)$$

$$U_b = \frac{K_b}{l_b} \qquad (7)$$

$$U_e = \frac{K_e}{l_e}\frac{2(L_c + W_c)H_c}{L_c W_c} \qquad (8)$$

The collector efficiency factor $\bar{F}$ is a constant for any collector design, includes bond and fluid resistances, as reported by [14] and it could be calculated by Eq. 9 as:

$$U_e = \frac{K_e}{l_e}\frac{2(L_c + W_c)H_c}{L_c W_c} \qquad (9)$$

$$\bar{F} = \frac{1}{W\left[\frac{1}{U_l\{D + (W - D)F\}} + \frac{1}{C_b} + \frac{1}{\pi D h_{f\,i}}\right]} \qquad (10)$$

$$F = \frac{\tanh[Z(W - D)/2]}{Z(W - D)/2} \qquad (11)$$

$$Z = \sqrt{U_l / K\delta} \qquad (12)$$

The effectiveness of a single heat pipe heat exchanger is a function of the number of transfer units (NTU), it is given by Azad [1].

$$(NTU)_c = \frac{A_c U_{c,o}}{\dot{m}_w\, Cp,w} \qquad (13)$$

In the condenser part, since cold fluid is crossing with the vapor flow in the heat pipe, the vapour inside a heat pipe is almost at constant temperature. Therefore, the effectiveness-NTU equation for this condition will be as follows:

$$\varepsilon = 1 - (1-\varepsilon)^{N_{hp}} \qquad (14)$$

The overall effectiveness E of heat pipes array can be written as:

$$E = 1 - (1-\varepsilon)^{N_{hp}} \qquad (15)$$

Heat transfer from the evaporator section to the condenser section may be mathematically modelled by a number of thermal resistances. The thermal resistance of the heat pipe is the sum of the individual resistances, including the resistance across the thickness of the pipe, the convective heat transfer resistance at the evaporator section, the liquid and vapor interface's resistance, the convective heat resistance at the condenser section and the resistance associated with the conduction process through the pipe wall of the condenser. These resistances could be obtained from the equations given by Tong et al. [2], respectively, as following:

$$R_{e,p} = \frac{\ln(D_o / D_i)}{k_p(2\pi L_e)} \qquad (16)$$

$$R_{e,i} = \frac{1}{h_e(\pi D_i L_e)} \qquad (17)$$

$$R_{e,iv} = \frac{2}{h_e(\pi D_i L_e)} \qquad (18)$$

$$R_{c,i} = \frac{1}{h_{i,c}(\pi D_i L_c)} \qquad (19)$$

$$R_{c,p} = \frac{\ln(D_o / D_i)}{k_p(2\pi L_c)} \qquad (20)$$

$$\sum R_{hp} = R_{e,p} + R_{e,i} + R_{e,iv} + R_{c,i} + R_{c,p} \qquad (21)$$

$$Uhp = \frac{1}{\sum R_{hp}} \qquad (22)$$

Where, the heat transfer coefficient between the fluid and the tube wall is given by:

$$h_{f,i} = 0.728[\frac{g\rho_l(\rho_l - \rho_v)k_l^3\lambda}{D_i\mu_l\Delta T_i}]^{0.25} \quad (23)$$

The absorbed plate temperature and stored water temperature are function of the overall effectiveness-NTU and could be calculated as:

$$T_p = T_a + \frac{S}{U_l}\frac{T_o - T_i}{NTU_{hp}} \quad (24)$$

$$T_o = T_i + \frac{E(T_p - T_i)A_{hp}U_{hp}}{A_cU_{c,o}} / (1 + \frac{A_{hp}U_{hp}}{A_cU_{c,o}}) \quad (25)$$

The number of transfer units for evaporator section of heat pipes array is equal to:

$$(NTU)_{hp} = N_{hp}(\frac{\bar{F}A_eU_l}{\dot{m}_w Cp,w})_{hp} \quad (26)$$

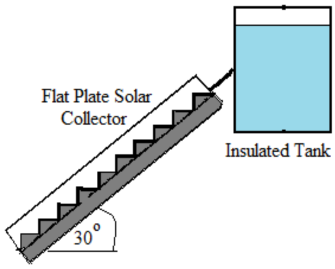
The efficiency of the thermal energy storage system is dependent on the incident solar energy and the amount of heat gained by the working fluid. It is the ratio between the sensible energy stored in the water tank and the incident solar energy on the glazing surface of flat plate collector during a certain period. It is defined as:

$$\eta_{storage} = \frac{\dot{Q}_u}{I_{FPC}} \quad (27)$$

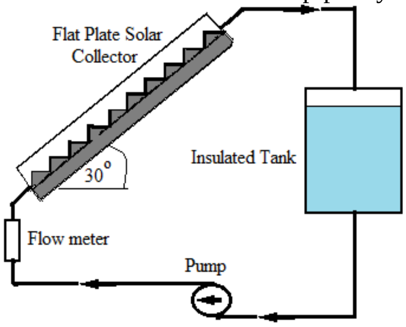
3- Experimental Setup:

An extensive experimental investigation has been performed under actual weather conditions in the solar energy laboratory, department of Mechanical Power Engineering, Faculty of Engineering, Menoufia University, Shebin El-Kom, Egypt at Latitude of 30.58°N and Longitude of 31.01°E. The experiments were conducted in two stages. In the first stage, two set of solar water heaters have been designed, fabricated and tested for comparing the performance of solar water heater with gravity assisted heat pipes system uses methanol and acetone as working fluid, with the performance of solar water heater with forced circulation works at two flow rates. The second stage of the experiment was studying the effect of increasing the number of heat pipes on the performance of the solar water heater with gravity assisted heat pipes system. Photograph of the two solar water heaters systems are shown in Fig. 1 and their diagrams are showing in Fig. 2.

Fig. 1. Photograph of the two solar water heaters systems



(a) solar water heater with heat pipes system



(b) solar water heater with forced circulation

Fig. 2. Schematic diagram of the two solar water heaters systems

The specifications of the two solar water heaters are summarized in Table (1). The flat plate with gravity assisted heat pipes system consists of four copper tubes forming the heat pipes. Charged with 20% of its volume by the working fluid under atmospheric pressure as reported by Hussien [5]. The first part of the heat pipe is combined together under pressure in the grooves of the copper flat plate absorbs. The solar radiation strikes this part is known as evaporator. The other side of the heat tube is the condenser. This part is inserted into the water tank for heating process.

The conventional solar water heater system consists of 6 copper tubes (4 tubes risers and two tubes headers). They have been integrated together and well welded with a copper sheet. In addition, a flow meter and valves have been used to measure and control the water flow rate. A pump is used to circulate the water inside the system.

Table (1): Solar water heaters dimensions

Specification	**Heat pipe Collector**	**Convention Collector**
Size of Collectors	Width=0.5 m, length=0.7m, depth=0.1m	
Absorber areas	0.24 m^2	
Cover thickness	5 mm	
--Edge thickness	50 mm	
Tilt angle	30 Degree	
Tank capacity	14 l^3	
Header diameter	-	19.1 mm
Riser Tube diameter	-	12.7 mm
Condenser length	0.2 m	__
Evaporator Length	0.6 m	__
Flow rates	-	4,7.5,11*l/min*
Heat pipes number	4, 9, 14	__

4- Results and Discussions:

In order to assess the performance of the both solar water heaters systems, different operating parameters have been recorded. The measured quantities are solar intensity, absorber plate temperatures and storage tank temperature. Additionally, the amount of heat energy collected and the thermal efficiency of the two systems have been calculated from the measured data. The theoretical results were performed and compared with the experimental data to investigate the performance of both systems.

The incident solar radiation intensity on a slop surface at selected angle equals 30° with the local time was presented in Fig. 3. It is observed that, the solar intensity increases rapidly in the morning and attains its maximum value at the noon (about 12 O'clock), then it is decreased until approaching to small value at sunset. The maximum value of actual incident solar radiation was around 1100 W/m^2. Theoretical results indicate that, the variation of the incident solar intensity follow the well-known Gaussian distribution without any fluctuation, recording its maximum values between 950 and 1050 W/m^2.

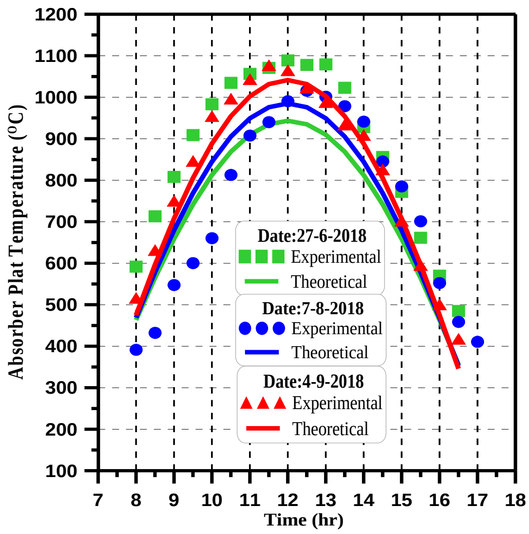


Fig. 3 Variation the incident solar intensity along the day.

Figure 4 presents the variations of the temperatures at absorber plate of the two heat pipes water heater systems, those use ethanol and acetone and their comparison with conventional system at flow rates 7.5 *l/min* and 11 *l/min*. The temperatures of absorber plate in the conventional system increases significantly from early morning with time until 1 PM. This is a direct effect of the increased solar radiation. After 1.00 PM, the absorber temperature increases and the heat losses become larger, causing a smaller gain in useful energy. This is attributed to the solar intensity decreasing. Therefore, the water begins to extract its heating energy from the plate itself. At this stage, the absorbed plate temperature is gradually decreased.

The heat pipes system measurement, with ethanol as a working fluid, is shown in Fig. 4.a. The ethanol begins to boil, under atmospheric pressure at 78 °C. It is observed that, the actual operating hours of the system start when the absorbed plate temperature reaches to 78 °C, and the heating continues until the surface temperature drops to lower from the boiling point of the ethanol. The same behavior also appears in the heat pipes system measurements with the acetone as a working fluid. The acetone begins to boil, under atmospheric pressure, at 56 °C. Figure 4b shows that, the actual operating hours of the system start when the absorbed plate temperature reaches to 56 °C. Thus, the operation period of the heat pipes system with acetone is higher than the operating period the heat pipes with ethanol. Generally, the operating hours of the heat pipes system are greater than the operating hours of the conventional system when measured in the same conditions.

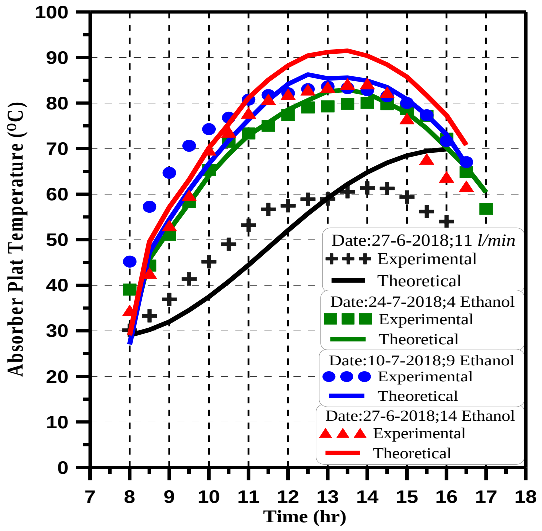


Fig.4a Variations of absorber plate temperatures of the heat pipes system uses ethanol.

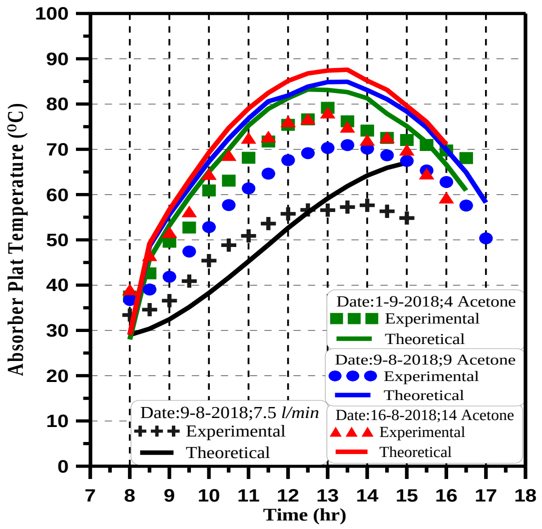


Fig.4b Variations of absorber plate temperatures of the heat pipes system uses acetone.

The theoretical model of three systems, indicate that the operating start point is at 8 AM. The theoretical plate temperature is lower than that of experimental data. This is due to the assumption of the initial conditions at the beginning of the operation, that the temperature of the absorbed plate is equal to the air temperature, but in fact they are higher than the ambient air temperature. Clearly, they are committed to the same degree of bending, a good agreement is obtained until noon time up to 1 PM. After that, there is an overestimation of the absorbed plate temperature. This is the match error between the experimental and theoretical results which may be attributed to neglecting the calculation of the plate increased thermal losses in theoretical model leading to increase the theoretical calculates.

Figure 5 demonstrates the comparisons between the experimental data and theoretical results of the storage tank water temperature for the conventional system at flow rate of 7.5 *l/min* and 11 *l/min* respectively. The temperature of water tank increases significantly from early morning with time until 1 PM, the reason is the direct effect of the absorbed radiation by plate. After 1.00 PM, the storage tank temperature is increasing slowly and reaches its maximum value at the end of working period. The obtained results agree well with the experimental data. Nevertheless, there is a little deviation. Such deviation may be due to the connecting hoses between the plate and the tank which have not been considered. Therefore, the heat loss from these hoses to the ambient has been neglected causing also an over estimation of the storage tank water temperature. There is an increase in the water tank temperature of the theoretically calculated along the operation period. However, the maximum temperature is around 65^{o}C at using flow rate 11 *l/min*, theoretical value was 69^{o}C.

The tank water temperature for the heat pipes systems are presented in Figs. 5.a and 5.b at use ethanol and acetone as working fluids, respectively. The tank water temperature for the case of installing 9 heat pipes was higher than 4 heat pipes and the case of installing 14 heat pipes was the highest. The percentage improvement in water temperature for the cases of 9 pipes and 14 pipes ethanol, relative to 4 pipes, were 20% and 30%, respectively. Also, the corresponding increase in the case of acetone as a working medium was 14% and 20% respectively. The maximum temperature attained in the experiments was around 67.8^{o}C. The theoretical results were well agreed with the experimental data. Nevertheless, it is observed that there is a little deviation at using the acetone. This deviation increases slightly if ethanol is used. Such deviation may be due to the delay of the beginning of the actual operating hours of the heat pipes system. The maximum stored water temperature was ranging between 54^{o}C~69.3^{o}C. The minimum water temperature was attained at using ethanol with installing 4 heat pipes and the maximum value was for ethanol with installing 14 heat pipes.

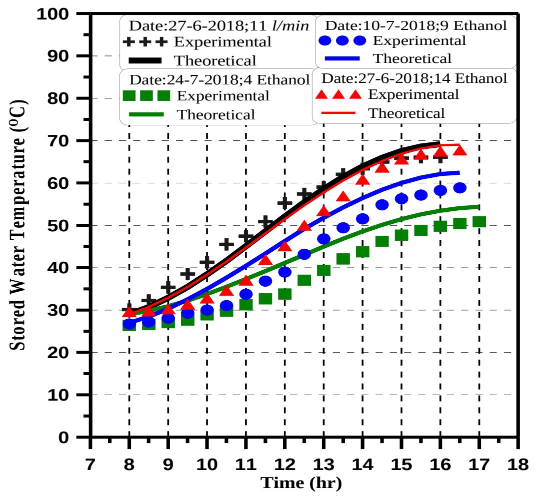


Fig. 5a Variations of the stored water temperatures of the heat pipes system uses ethanol.

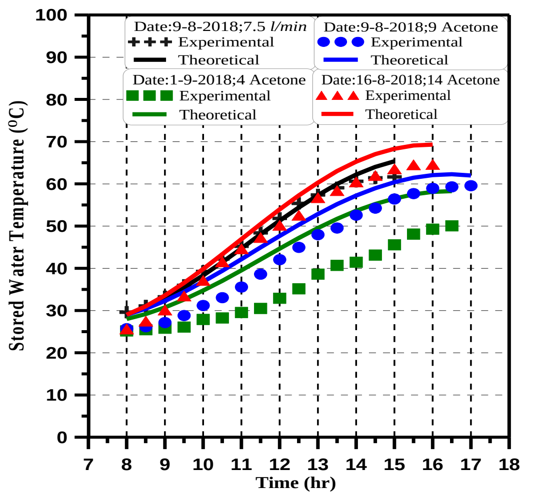


Fig. 5b Variations of the stored water temperatures of the heat pipes system uses acetone.

The amount of stored energy in the storage tank is illustrated in Fig. 6. The maximum stored energy of conventional system was about 2.1 MJ at flow rate 11 *l/min*. The stored energy of the heat pipes system increases with increasing the operating period. The percentage improvement in stored energy for the cases of 9 pipes and 14 pipes ethanol, relative to 4 pipes, were 44% and 74%, respectively. Also, the corresponding increase in the case of acetone as a working medium was 26% and 56% respectively. The maximum stored energy of heat pipes system was about 2.3 MJ with using acetone.

It could be observed that, the theoretical results have well agreed with the experimental values of stored energy. Where the tank water temperature is the main parameter in stored energy accounts. Thus, it is clearly the stored energy behavior follows the water tank temperature behavior of each case. The maximum stored energy of conventional system was around 2.35 MJ at flow rate 11 *l/min*. Also, the stored energy of the heat pipes system was showed, the value maximum was ranging between 2.34~2.37 MJ at installation 14 heat pipes.

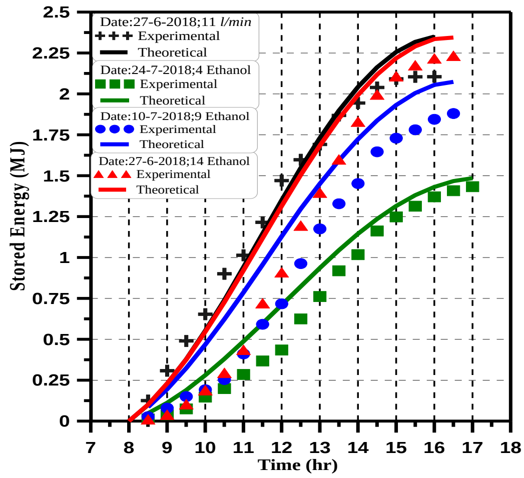


Fig. 6a Variations of stored energy of the heat pipes system uses ethanol.

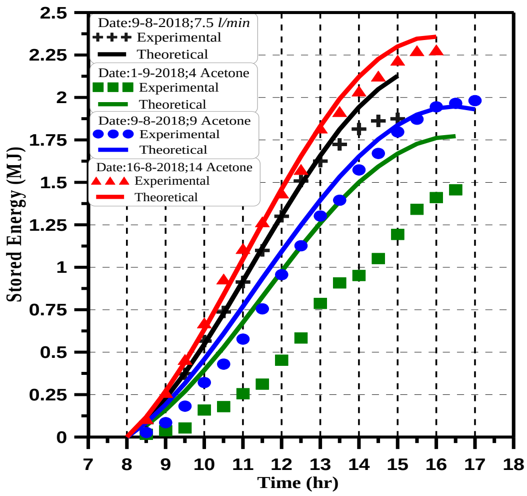


Fig. 6b Variations of stored energy of the heat pipes system uses acetone.

The variation of the hourly efficiency for sample at the measured day is shown in Fig. 7. For conventional system, it is observed that the efficiency has higher values at the beginning of the storage operation and increases rapidly until it reaches its maximum value near it before the noon time. A gradually decrease is followed till the end of the process. The main effective parameter on the system efficiency is the heat gained by water. At the beginning, such heat is small due to the heat absorbed by the capacity of the system but its increasing rate becomes more dramatic than the solar radiation with time until noon. The theoretical and practical water temperatures were very agreed and close, but the difference was that the calculated solar radiation was less than the actual solar density at the measurement days, as it is observed from Fig. 3. This is the main reason for the disparity between theoretical and practical efficiency. The maximum values of the theoretical and actual efficiency are at the beginning of operation, because this is the period at which the water temperature is close to the ambient air temperature, as reported by Struckmann [15]. Maximum value of theoretical efficiency was in the range of 50~53% while the actual efficiency was between 47~53%. Generally, it is observed that, there is a good agreement between the theoretically calculated efficiency and the actual value derived from practical measurements.

In the same way, it can be seen that the efficiency of the heat pipes systems increases gradually with the increase of solar radiation. Due to the delay of the operation temperature of the heat pipes system, a decrease in efficiency before peak time is noticed. However, in the afternoon time, the efficiency of the heat pipes systems is higher than that of the conventional system. It could be also noted that, whether theoretical or actual efficiency follows the same behavior as the conventional system efficiency, where the maximum value is in the first hours of operation. But the actual operating hours of the heat pipes systems are a slightly delayed and may start at noon. This explain why the actual efficiency is less than theoretical efficiency before noon time. However, there is good agreement between the theoretical and actual efficiency. The maximum theoretical efficiency was around 55% at installation 14 heat pipes of acetone, which the experimental efficiency was 53% for same the case.

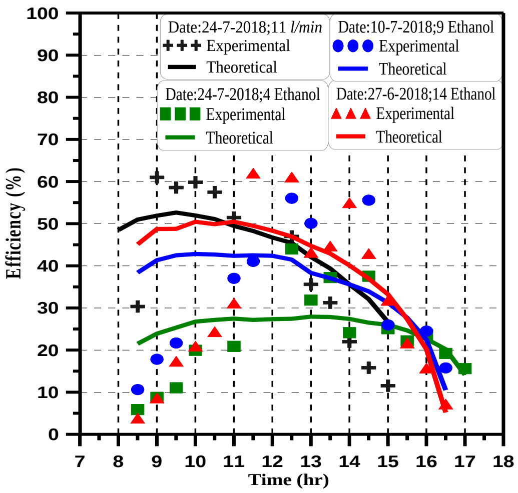


Fig. 7a Variations of hourly efficiency of the heat pipes system uses ethanol.

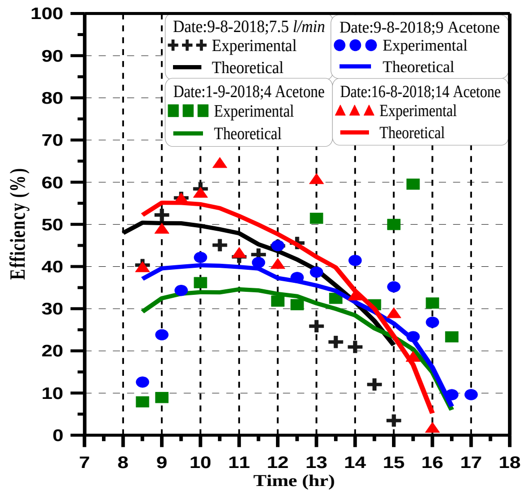


Fig. 7b Variations of hourly efficiency of the heat pipes system uses acetone.

Figure 8 shows the effect of flow rate on the accumulated stored energy and the efficiency of forced circulation system. It is seen that, the performance of the convention system uses flow rate 11 *l/min* was better than 4 *l/min* and 7.5 *l/min*.

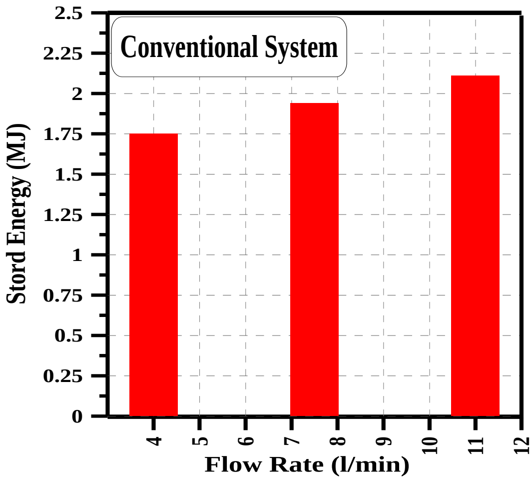


Fig. 8a. The effect of the flow rate on the stored of conventional system.

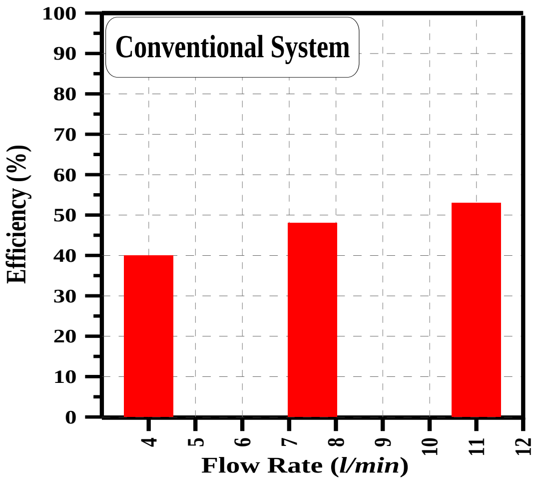


Fig. 8b. The effect of the flow rate on the hourly efficiency of conventional system.

The effect of using ethanol and acetone as the working fluids and heat pipes number on the accumulated stored energy and the efficiency of the heat pipes system are illustrated in Figs. 9a, 9b respectively. It is observed that, the performance of the heat pipes system uses acetone is higher than that uses ethanol. This is because the acetone boils at 56°C which increases the operating times of the system, thus lead to increasing the accumulated stored energy efficiency. As well as number change of heat pipes on the absorbed plate lead to a substantial improvement in the stored energy and the efficiency. It can be noted that, the stored energy at installation 9 heat pipes was higher than 4 heat pipes, also at installation 14 heat pipes was the highest.

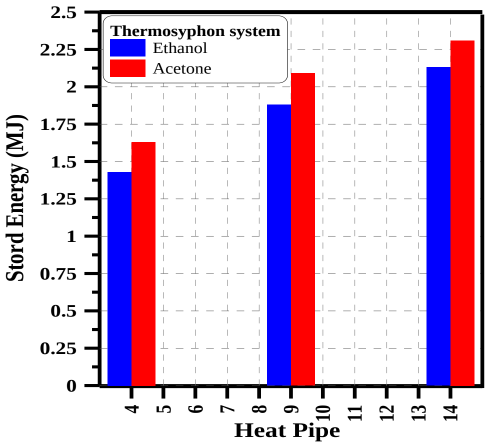


Fig. 9a. The effect of the number of heat pipes and working fluid on the stored energy of the heat pipes system.

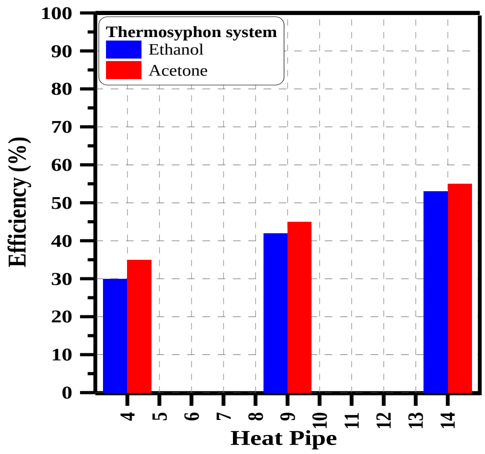


Fig. 9b. The effect of the number of heat pipes and working fluid on the hourly efficiency of the heat pipes system.

The efficiency of the heat pipes system uses ethanol and that uses acetone are presented in Fig. 10. In the figure, a comparing of the efficiency with that of the heat pipes system uses distilled water as working fluid that carried by Nada et al [6] is shown. Figure 10.a shows the rapprochement of climatic conditions of measuring days for the heat pipes systems. The effect of the working fluid boiling point on efficiency of heat pipes was noted in Fig. 10.b. The efficiency of the system utilizes acetone was better than that of the system utilizes ethanol and distilled water. In addition, the efficiency of heat pipes uses distilled water was the lowest due to the rise of distilled water boiling point. The boiling point of acetone is the main parameter in the low efficiency afternoon. This is the reason that the temperature of water in the storage tank approaches the boiling point of acetone which leads to reduce the amount of heat transferred and the efficiency. The efficiency of the system utilized ethanol and distiller water are high in the afternoon. This because the temperature of water in the storage tank does not approaching their boiling point which lead to increase the amount of heat transferred, thus rises the efficiency.

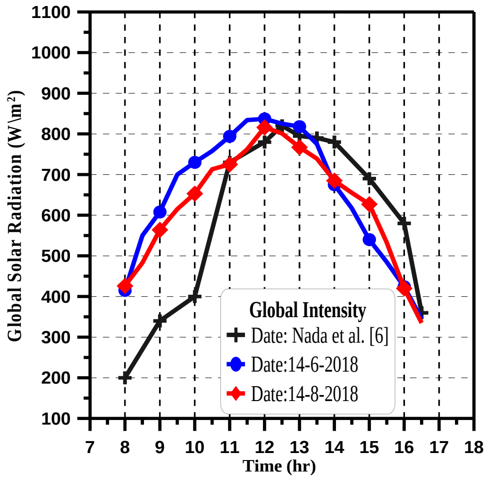


Fig.10a. The global intensity of the experimental data and date carried by Nada et al [6].

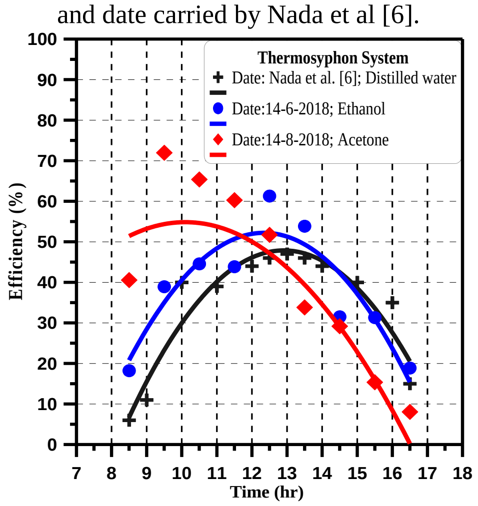


Fig. 10b. The hourly efficiency of the experimental data and date carried by Nada et al [6].

Conclusions:

This study of the conventional solar water heater and that with heat pipes led to the following conclusions:

1- The maximum obtained experimental water temperatures of the conventional system and the heat pipes system uses ethanol or acetone were 66°C, 67.8°C, and 64.6°C, respectively. For theoretical calculations were 69.2°C, 69°C and 69.3°C, respectively.
2- The maximum accumulated stored energy of the forced convection system and the heat pipes system charged with ethanol and acetone was 2.11 MJ, 2.23 MJ and 2.3 MJ.
3- The efficiency of the conventional system is high at the beginning of the operational. Maximum value of the actual efficiency was in range 47~53% and theoretical efficiency was between 50~53%.
4- The efficiency of the heat pipes systems are highest in the afternoon. The experimental efficiency maximum was around 55% at installing 14 heat pipes, which the theoretical efficiency was 55% for same the case.
5- The heat pipes system remains better because it does not consume electricity.
6- The performance of the heat pipes system uses acetone as working fluid was better than the heat pipes system uses ethanol.
7- The percentage improvement in water temperature for the cases of 9 pipes and 14 pipes ethanol, relative to 4 pipes, were 20% and 30%, respectively. Also, the corresponding increase in the case of acetone was 14% and 20% respectively.
8- The percentage improvement in stored energy for the cases of 9 pipes and 14 pipes ethanol, relative to 4 pipes, were 44% and 74%, respectively. Also, the corresponding increase in the case of acetone as a working medium was 26% and 56% respectively.

Nomenclature:

(Units are as stated here, unless noted otherwise in the text of the study)

A: Area, m^2.
C_b: Thermal conductance between plate and tubes, W/K.
C_p: Specific heat capacity, J/kg.K.
D: Diameter of tube, m.
E: Overall heat transfer effectiveness.
F: Standard fin efficiency.
F_R: Collector efficiency.
$\bar{F}$: Heat removal factor.
G: Gravity acceleration, m/s^2.
H: Height, m.
h: Convective heat transfer coefficient, $W/m^2.K$.
I_T: Total solar radiation incident on tilted surface, W/m^2.
k: Thermal conductivity, W/m.k.
L: Length, m.
l: Thickness, *m.*
m: Mass flow rate of water, kg/s.
N: Number.
NTU: Number of transfer units for heat pipes.
Q_u: Useful heat gain rate, W.
R: Thermal resistance m^2 K/W.
S: Absorbed solar radiation per unit area, W/m^2.
T: Temperature, °C
t: Time, s.
t_p: Tube thickness, m.
U: Heat transfer coefficient, $W/m^2.K$.
V: Wind Speed, m/s.
W: Width, m.

Greek symbols

β: Tilted surface angle Degree.
ε_1: Effectiveness.
μ: Dynamic viscosity, Pa.s.
λ: Latent heat, J/kg.
η: Efficiency.
ρ: Density, kg/m^3.
σ: Stefan-Boltzmann constant, $W/m^2.K^4$

Subscripts

a: Ambient.
c: Collector, Condenser.
fi: Fluid inlet tubs.
hp: Heat pipe.
ins: Insulation.
lv: Liquid-vapor interfaces.
p: Plate.
t: Top.
w: Water, Wind.
b: Back.
e: Evaporator, Edge.
g: Glass cover.
i: Inlet, Inside.
l: Liquid.
o: Outlet, Outside.
s: Surface.
v: vapour.

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