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Study the Effect of Variable Condensers Length on Improving the Performance of a Heat Pipe Solar Collector

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ABSTRACT

This paper presents a study ofimproving the performance of solar collector heat pipe by increasing condenser length. The experimental rig consists of a heat pipe inserted in a water tank for heating. The dimensions of copper thermosyphon(heat pipe) are: inner diameter 14 mm, outer diameter 16 mm, fixed evaporator length 1.15 m and variable condenser lengths (0.150,0.200 and 0.300 m). The rig was built and tested outdoor in Baghdad with different inclination angles, namely:(15˚,30˚,45˚,60˚,75˚ and 90˚) anddifferent fillingratios, namely: (30,50,70, 90%)during aclear sky.Numerical investigation included solving fluid flow and heat transfer equations inside water tank by finite volume and energy equationsfor heat pipe by thermal resistance. The experimental results showthat, increasing condenser length by 25% will increase water tank temperature by 17% and increasesheat pipe vapor core temperature by 6% for a filling ratio of 30% and inclination angle 45˚. Experimental and numerical results show good agreement with a deviation of about 5%.

KEYWORDS: Solar collector; heat pipe; condenser length; tilt angle; filling ratio; CFD.

Symbol	Meaning of Symbol	Unit
A_s	south direction area	m^2
A_n	north direction area	m^2
A_e	east direction area	m^2
A_w	west direction area	m^2
A_t	top direction area	m^2
A_b	bottom direction area	m^2
C_s	constant	
C_n	constant	
C_e	constant	
C_w	constant	
C_t	constant	
C_b	constant	
C	constant	
C_p	specific heat at constant pressure	J/kg K
g	acceleration due to gravity	m/sec^2
h	Heat transfer coefficient	W/m^2 K
h_{air}	heat transfer coefficient of air	W/m^2 K

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h_w	heat transfer coefficient of water	W/m^2 K
i,j,k	nodes x-,y-,z- direction respectively	
k_c	thermal conductivity of copper	W/m K
k_w	thermal conductivity of water	W/m K
l	length	m
Pr	Prandtl number	
q_s	south direction heat transfer	W
q_n	north direction heat transfer	W
q_e	east direction heat transfer	W
q_w	west direction heat transfer	W
q_t	top direction heat transfer	W
q_b	bottom direction heat transfer	W
q_{solar}	solar flux	W/m^2
r,θ, z	cylindrical coordinates	m, degree, m
t	time	sec
T	temperature	K
u	velocity of x- component	m/sec
v	velocity of y- component	m/sec
w	velocity of z- component	m/sec
x,y,z	Cartesian coordinate	
Δv	volume of element	m^3
α	Thermal diffusivity	$[m^2/s]$
β	volume coefficient of expansion	1/K
θ	inclination angle	degree

INTRODUCTION

At the present time, applications of solar energy usingheat pipes are on a large scale, [1-4]as it has the advantage of ($\eta_{overall}$, 25-30%) over traditional collectors. Compared with traditional collectors, heat pipesolar collectors are better in that, they have good heat collection efficiency and better heat transfer. Furthermore, the rate of heat transfer is far greater than that of a solid finned conductor having the same size. Due to their physical properties, heat pipes are used in a variety of applications such as cooling of computers and electronic instruments, optical components, milling machines spindles, refrigeration and air conditioning systems waste heat recovery systems, heat exchangers, power generation systems and in solar energy production. Moreover, heat pipes are used in a variety of operations in such industries as chemistry in all its fields, metallurgy, transportation machinery and electronics and communication appliances.

Aljboory [5] made outdoor experiments using evacuated tube with heat pipe for solar a heating system.Copper tubes with 22 mm diameter was used without wick and used acetone, ethanol and methanol as fluids in the experiment with four filling ratios of 25%, 50%, 70% and 90%, and inclination angles of 45^{o}at a fill charge ratio of 50%. Experimental results showed an optimum inclination angle of 45^o at fill charge ratio 50%. Azad [6] made experiments on interconnected solar collector heat pipe. First version of the system was designed and constructed. With this system, heat pipe production cost was reduced and work conditions were improved, so that the system can be used as a transformer of heat flux compared with the traditional one. The heat transfer in the collector was examined by an analytical method developed to find out the efficiency of the collector, temperature of heat pipe, temperature of water outlet and cooling water absorption of useful heat. Noie*et al.* [7] studied a number of parameters such as filling ratio and aspect ratio to find out the performance of inclined closed thermosyphon with two phases operating at normal conditions. The working fluid was water and the experiment was conducted under aspect ratio ranging from 15, 20 and 60 and the fill charge ratios from 20% to 60%. A tube made of copper was used; it was 1m long,14 mm in inner diameter and 16 mm outer diameter.Results from experiment revealed that, the best heat performance took place at a ratio of charge of 45% and tilt angle 60°.Ong and Haider [8] dealt with the heat performance of heat pipe filled with R134a. The factors studied cover ratio of filling, difference in temperature between bath and condenser section and different mass flow rates. A tube made of copper was used; it was 780mm long (300mm- length of evaporator, 300mm- length of condenser) 25.5 mm inner diameter and 28.2 mm outer diameter. It was found that, the transfer of input heat flux increased with the increase in ratio of filling, difference in temperature between condenser section and bath and flow rate of coolant mass. Anjankar and Yarasu [9] studied the influence of heat load, the rate of coolant mass flow rate and length of condenser on heat performance of closed thermosyphon with two phases. A closed tube made of copper was used having 1m length (length of evaporator-300, length of condenser - 450mm/400mm/350mm) 26mm inner diameter and 32 mm external diameter. From the experimental results, it was shown that, the optimum performance was achieved at 500W heat input, 0.0027kg/s flow rate of coolant mass and 450mm condenser length. Rahimi*et al.* [10] looked into the influence of the evaporator surface modification and resurfacing of condenser on whole thermosyphonperformance. From the results, it was shown that; the condenser and evaporator were more hydrophilic, resulting in an increase in thermosyphon performance by 15.27% and a decrease in thermal resistance by 2.35. Mirshahi and Rahimi [11] made

experiments on the influence of large volume loads of heat and ratio of filling on evacuated thermosyphon heat performance. From the experimental results, it was shown that, the thermosyphon performance was greatly affected by fill charge ratio change, flux of heat and different additional volumes.Mohanad [12]studied the performance of copper thermosyphon (wickless) theoretically and experimentally. The dimensions of thermosyphonwere; outer diameter 12.7 mm, inner diameter 9.53 mm and the length 1967 mm (evaporator section was 1800mm, adiabatic section was 22 mm and condenser section was 145 mm). Five inclination angles were tested from 23° to 90°, five heat fluxes from 250 to 1253 W/m², three mass flow rates 0.0017, 0.0022 and 0.0031 kg/s and Fluorocarbon FC-72 was used as the working fluid in this work with a filling charge ratio of 70 %.Ahmad and Yousif[13]designed two systems of heat pipe and thermosyphon to investigate performance and to compare between them using different lengths of the evaporator. The paper studied the effect of parameters like (heat input, working fluid and filling ratio) on the thermal performance. The two devices were constructed from copper; the dimensions were0.35 m outer diameter,0.032 m inner diameter and a length of 1m. The two ends of devises were closed by end caps made from copper of a thickness of 0.003m and stainless steel was used as a wick in heat pipe.Anwarul*et al.*[14] discussed the performance of a gravity assist heat pipe made of stainless steel,inner diameter 12.5 mm and length 0.50 m.The working fluid used in the evaporator section was water. Different inclination angles from 0° to60° and various heat inputs from 25 to 40 W were used. The material of wick was of stainless steel. The temperature distribution along the outer surface of heat pipe was measured by 18 thermocouples type K. Heater Ni-Cr wire was used for heating of the evaporator and to avoid heat loss, the evaporator and adiabatic section were thermally insulated with glass wool. The condenser was inserted in water jacket. The results showed lower evaporator temperature for vertical heat pipes and higher evaporator temperature for a horizontal one and the evaporator temperature increased when inclination angle increased. Hammad and ALsuwaidi[15] investigated experimentally the performance of two phase closed thermosyphon used various geometry (rectangular, square and circular) with same length and hydraulic diameter made of copper material used methanol as working fluid. The results explain that the thermal resistance decreases with increases of the input power while heat transfer coefficient increases with increasesin input power and higher heat transfer coefficient occurs at power equal 500 W for square cross section. Ting-Ting *et al*. [20]determined theoretically and experimentally the performance of a solar water heating system with evacuated tube heat pipe collector. By using mathematical model with thermal and exergy analysis the collector performance was explain. The results showed that the maximum number of pipes was used to be 15. Also, relation was found between system performance and hot water consumption. Maximum efficiency reaches at the end of the day 5.4% and the maximum outlet temperature of collector was about 64 °C, which happen between 15:00 and 16:00 pm. Maximum relative error of 8.4% and the maximum standard error of 1.91% when comparison between theoretical and experimental results. Mustafa [21] study the effects of six various working fluids, hexane, ethanol, petroleum ether, acetone, chloroform and methanol on the energy and exergy performance in evacuated tube solar collectors with thermosyphon heat pipe under different range of air velocities as 2, 3 and 4 m/s. Six evacuated tube solar collectors with thermosyphon with the same properties and dimensions are designated for the air heating and examined under the outdoor conditions of Uşak, Turkey. The lowest energy and exergy efficiencies occur in the THPETC-Hexane under 2, 3 and 4 m/s, the highest energy efficiency occurs in the THPETC-Acetone for air velocity of 2 and 3 m/s and in the THPETC-Chloroform for air velocity of 4 m/s. The highest exergy efficiency occurs in the THPETC-Acetone for air velocity of 2 m/s and in the THPETC-Chloroform for air velocity of 3 and 4 m/s. Jahanbakhsh*et al.*[22]built heat pipes and examined using different working fluids of ethanol and water with different inclination angles. Tests were carried out to study the effects of some parameters like, the pipe evacuation, ethanol concentration and utilize of a wick inside the pipe to explainthe performance of the system. For ethanol the tests indicate that the best performance characteristics in transferring the heat at low heat flux, and that filling ratios of 50% and 75%. The system efficiency was found about 52%. Not improvement was foundfrom evacuation of a heat pipe and by using a wick of a heat pipe for performance, and optimum heat transfer coefficient for collector was occur at inclination angle of 35°. The results of solar testing for heat pipes, a thermo-syphon type improved, by using the evacuated glass tubes and results showed that using the evacuation of the glass tubes and had little effects on the improvement of the solar collector thermal performance.Carlos *et al*. [23]studied the use of solar energy to producing hot water. A solar water heating system (SWH) using a heat pipe with evacuated glass tube was built, and study's the effect of the water storage tank temperature TWT on the thermal performance of the system by calculationthe maximum quantity of withdrawable hot water. The results showed thatthe increases of the water storage tank temperaturereduction the net energy stored by the system between the water storage tank temperature of 40 °C and 80 °C at a solar radiation input of 8000 W/m^2.Jianshengetal. [24]studied two-dimensional single loop closed-loop pulsating heat pipe (CLPHP). The method of volume of fluid (VOF) is used to study thermal performance of CLPHP with various ratios of the evaporator length to that of the condenser. The action of wall adhesion and the effect of surface tension, the continuum surface force (CSF) model is approved. The range of input power (10 - 40 W) and range of filling ratio was used in tests (30% - 60%). The experimental and numerical results are compared with same condition. Increase of input power decreases the

thermal resistance and the start-up time of CLPHP. At the same input powerand filling ratio condition, decreasing the condenser length of within a suitable range would increase the start-up process and decrease the thermal resistance while low filling ratio caused the dry-out. Two kinds of start-up, sudden temperature increasing starts at lower input power and the smooth temperature increasing starts at higher input power are observed.

A number of researches have investigated solar evacuated tubes made of glass and study the effect of evaporator length but there is a shortage in literature regarding study of condenser length.The aim of the present work is to study the effect of various condenser lengths on the thermal behavior of wickless heat pipe.This work includesexperimental and numerical studies.Numerical investigation simulate the whole complicated process, it consists of two parts;in first part, the energy equation inside heat pipe will be solved by thermal resistance with suitable heat transfer coefficients for boiling and condensing. In the second part, 3D Navier Stocks equationswith Cartesian Coordinates in water tank based on finite volume will be solved.The two parts will work simultaneouslyto determine the semi transient temperature history in heat pipe and water tank during the day.Two outdoor experimental rigswith different condenser lengthswill builtas shown in table 1. The tests were performed to see the effects of increasing condenser length on the heat pipe characteristics.

2. Experimental Apparatus and Test Procedure:

Two outdoor systemsof evacuated tube heat pipe solar water heating were built and tested at conditions of Baghdad;the first system, atube made of copper material was used with inner diameter 14 mm and outer diameter 16 mm, fixed evaporator length of 1.15 m and variable condenser length 15 and 20 cm.While, the second system was the same as the first system but with condenser length of 20 and 30 cm. Distilled water was used as working fluid with filling ratios (30, 50, 70 and 90%). The condenser was immersed in water storage tank while the evaporator was locatedin an evacuated glass tube. Each system containsthermosyphon, in evacuated glass tube, water storage tank and reflector mounted on a stand facing to the south.

In order to investigate the performance of a tilt two-phase closed thermosyphon, an outdoor experiment was designed and built as shown photographically in Fig.1.It consists of:

(1)Copper thermosyphon
(2)Water tank
(3)Thermocouples type K
(4)Data Acquisition
(5)Personal Computer
(6)Evacuated glass tube

Twenty thermocouples type (K) (Nickel Chrome, Nickel Aluminum) were used to measure the temperature, eight at the outer wall of heat pipe, five inside heat pipe (core), three inside the tank and two at inlet and outlet. The tips of these thermocouples were impeded in a drilled hole of 2 mm depth located at the outer surface of the heat pipe and fixed by using high conductivity epoxy. All the thermocouples were calibrated and connected to the Data Acquisition system type Lab – Jack.A pressure gauge was used to measure the pressure inside the thermosyphon. It was mounted on top.

The heat transfer coefficient for evaporator and condenser section can be calculated from the following equations[16]:

$$h_e = \frac{\mathrm{Q}}{2\,\pi r_o l_e (T_e - T_V)}$$

$$h_c = \frac{\mathrm{Q}}{2\,\pi r_o l_c (T_v - T_c)}$$

3. Numerical Model:

The numerical model consists of two parts; in the first part (subroutine) energy equation inside the heat pipe was solved. In the second part (main program),Navier Stokes equations in the water tank were solved based on finite volume. The aim of the two parts is to determine the temperature distribution inside heat pipe and the water tank heated by heat pipe. In the subroutine, the heat is transferred from evaporator to condenser in the heat pipe then to water storage tank by convection heat transfer. The numerical model for heat pipe is transient (one hour)because the phenomena are unsteady during the daywhile the numerical model in the water tank was assumed steadyto simplify the problem in water storage tank. The value of solar heat flux during the day was chosen,one value for each hour, and then the steady solution of this value was linked to the solution of water tank simultaneously.The same procedure was repeated for each hour, so semi-transient solution was obtained for the tank. At each hour, the resultant temperature solution was used as an initial temperature values for the next hour

3.1. Heat Pipe:

A 3-D cylindrical coordinate system was used in the numerical model. Thermal resistance method was used to calculate the temperature distribution in heat pipe.The energy balance equation can be written for the general node(shown in Fig. 2)[17, 18].

Energybalance:

$$\sum Q_{\text{in all sides}} = q_{\text{storage}}$$

$$q_{s-r} + q_{e-\theta} + q_{t-z} + q_{n-r} + q_{w-\theta} + q_{b-zt} + q_{solar} = \rho V Cp \frac{T_{(i,j,k)}^{p+1} - T_{(i,j,k)}^{p}}{\Delta t}$$

$$T_{(i,j,k)}^{p+1} = T_{(i,j,k)}^{p} + \frac{\alpha \Delta t}{V}\left[C_S T_{i-1,j,k}^{P} + C_w T_{i,j-1,k}^{P} + C_t T_{i,j,k+1}^{P} + C_n T_{i+1,j,k}^{P} + C_e T_{i,j+1,k}^{P} + C_b T_{i,j,k-1}^{P} - C\, T_{i,j,k}^{P} + q_{solar}\right]$$

$$\alpha = \frac{k}{\rho C_p}$$

$$A_s = \left(r_i - \frac{\Delta r}{2}\right)\Delta\theta\Delta z\,, \qquad A_e = \Delta r\,\Delta z\,, \qquad A_t = r_i \Delta r\,\Delta\theta$$

$$A_n = \left(r_i + \frac{\Delta r}{2}\right)\Delta\theta\Delta z\,, \qquad A_w = \Delta r\,\Delta z\,, \qquad A_b = r_i \Delta r\,\Delta\theta$$

Where:

$$C_S = \frac{A_s}{\Delta r}\,, \qquad C_e = \frac{A_e}{r_i\,\Delta\theta}\,, \qquad C_t = \frac{A_t}{\Delta z}$$

$$C_n = \frac{A_n}{\Delta r}\,, \qquad C_w = \frac{A_w}{r_i\,\Delta\theta}\,, \qquad C_b = \frac{A_b}{\Delta z}$$

$$C = C_s + C_n + C_e + C_w + C_t + C_b$$

The numerical solution has taken into account the effects of:

1- Solar flux to evaporator external wall surface.
2- Conduction through evaporator walls.
3- Convection heat transfer from internal evaporator wall to water inside heat pipe.
4- Evaporation inside heat pipe in evaporator side.
5- Heat transfer from evaporator core to condenser core.
6- Condensation ofvapor in condenser part.
7- Conduction through condenser wall.
8- Convection from condenser external wall to water inside water tank.
9- Conduction through heat pipe wall from evaporator to condenser parts.

Suitable values for heat transfer coefficients inside heat pipe for evaporation and condensation were used in the solution.

3.2. Water Storage Tank:

For water storage tank, laminar3-D Cartesian Coordinates Navier Stokesequations (continuity, momentum and energy)weresolvedby finite volume (SIMPLE algorithm) using build in program (Fortran 90) and the graphs were plotted using Tecplot. The temperatures of the cylindrical heat pipe grids were transferred to the Cartesian water grids approximately by stair grids. Fig. 3 shows the stair grids of heat pipe inside the water tank. The surface area of the stair grids was adjusted to equal the surface area of the actual cylindrical grids of heat pipe.The 3d laminar Navier Stokes equations with natural convection are:

i- Continuity Equation:

$$\frac{\partial}{\partial x}(\rho u) + \frac{\partial V}{\partial y}(\rho v) + \frac{\partial W}{\partial z}(\rho w) = 0$$

ii-Momentum Equation:

u-Momentum (x-direction)

$$\frac{\partial}{\partial x}(uu) + \frac{\partial}{\partial y}(uv) + \frac{\partial}{\partial z}(uw) = -\frac{1}{\rho}\frac{\partial P}{\partial x} + \nu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right)$$

v-Momentum (y-direction)

$$\frac{\partial}{\partial x}(vu) + \frac{\partial}{\partial y}(vv) + \frac{\partial}{\partial z}(vw) = -\frac{1}{\rho}\frac{\partial P}{\partial y} + \nu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) + g\beta(T - T_w)$$

w-Momentum (z-direction)

$$\frac{\partial}{\partial x}(uw)+\frac{\partial}{\partial y}(vw)+\frac{\partial}{\partial z}(ww)=-\frac{1}{\rho}\frac{\partial P}{\partial z}+\nu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right)$$

iii-Energy Equation:

$$\frac{\partial}{\partial x}(uT)+\frac{\partial}{\partial y}(vT)+\frac{\partial}{\partial z}(wT)=\frac{k}{\rho C_p}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right)$$

RESULTS AND DISCUSSION

Figs. (4, 5 and 6) represent some of the numerical simulation results. Fig.4showsthe pattern of the flow field inside the water storage tank forFR 50%and inclination angle 45°. In a, where the plotted plane (z = 3.5 cm) passes the heat pipe, the water velocity vector is as expected, the highest velocities occur above the heat pipe,while, the lowest velocities occurred under heat pipe. This is due to the effect of natural convection, because the upper regions arefree to move.In b, where the plane (z = 4.5 cm) did not pass the heat pipe, the flow follows the same trend of a, but with less strength.

The temperature contoursin heat pipe with operating conditions FR (50%)and tilt angle 45°are shown in Fig. 5. The temperature variation across the entire heat pipe is dependent upon a number of parameters, namely; the coolant temperature, the convection coefficient between the outer wall of the condenser, the coolant fluid and the solar energy. Maximum temperature is located in evaporator section; while in condenser section, the temperature is less becausethe heat was rejected to heat water in storage tank.The temperature gradient between evaporator and condenser surfaces (axial direction) increases as the heat loadincreases. The temperature gradient in the radial direction was smaller and no temperature gradient was observed in θdirection.The Figure shows the behavior of heat pipe with different times during operation period from 8:00 am to 16:00 pm.

Fig. 6 shows the isothermalcontours in water storage tank (FR 50%, tilt angle 45°). The left lower corner of the tank was cut at 45° to facilitate assembly of heat pipe. The upper region was the highest temperature because it is near the heat pipe. The temperature of water storage tank increases with time because solar heat flux increases. The graphs were plotted at z=3.5 cm where the heat pipe exists. The higher temperature is in the top region.This is due to natural convection.

Fig. 7 shows the numerical results of the present work which have been compared with the experimental work. The relative error, for the evaporator-condenser temperature difference is 5-10%.

Fig. 8 shows the change of the mean water tank with respect to time for FR =0.5 and θ=45° for different lengths of condenser, namely: (20, 15 cm) with load condition 1 lit/hr. The temperatures in tank were measured by three thermocouples (top, bottom and middle point of tank).

The mean water tank temperature increased with time, reaching its optimum value at the end ofoperation period 16:00 p.m., and then reduced slightly. The temperature distribution in the storage tank is a function of the energy delivered, the energy losses and the mixing effect caused by natural convection.

Fig. 9 shows vapor core temperature for load condition with FR 0.5 and tilt angle 45° with two condenser lengths (150 and 200 mm). For evaporator section, the vapor core temperature is less than surface temperature but for condenser section, the behavior is opposite, the vapor core temperature is higher than the surface temperature. This difference between the core and surface temperatures for both evaporator and condenser sections increased with increased filling ratio and decreased with increased tilt angle. Another comparison is the difference between vapor core temperature in the evaporator and condenser sections. The comparison shows that vapor core temperature for evaporator section is higher than that in condenser section.

Fig. 10shows that, the trend of variation of the outer surface temperature with length in the present work is similar to that Mahdi [19],comparison with same design specifications and operation conditionsDI water was used as working fluid at 50% FR and tilts angle 45°. Mahdi working concerns a solar heat pipes with different parameter, with and without reflector, various evaporator lengths and different working fluids. His results showmaximum mean storage temperature occurs at larger evaporator length. The relative differences in the evaporator-condenser temperatures, between the present work and the work of Mahdi are 1.5% and 6%, respectively.

Fig. 11 shows the variation of surface temperature along the thermosyphon length with different solar heat flux and inclination angle 45° using water as the working fluid with various filling ratios 0.4 and 0.5, and initial tank temperature 25 °C. The x-axis shows the length of thermosyphon while the y-axis shows the temperature along thermosyphon the solar heat flux increases with time reached maximum value at 12:00 pm. and then decrease. In the evaporator section, the temperature distribution was near uniform and isothermal during operation of period.

Fig. 12 shows the experimental and numerical results of heat transfer coefficient change with respect to time during operation period for filling ratio 50 % and till angle 45°. Form the figure, the heat transfer coefficient increases as heat flux increases and reaches it maximum value at 12 PM. The main heat transfer

regime is natural convection, and the amount of vapor depends strongly on the amount of the working fluid, the heat pipe orientation and the input heat flux. The agreement between numerical and experimental results is good.

Conclusions:

1. The performance of heat pipe was improved by increasing condenser length.
2. Optimum fill charge ratio was 50% for solar application.
3. Higher water storage tank temperatures can be achieved with no load.
4. In evaporator section, the vapor core temperature was less than surface temperature, while in condenser section the behavior was opposite.
5. The difference between the core and surface temperatures for both evaporator and condenser sections increased with filing ratio and decreased with tilt angle.
6. For evaporator section of the heat pipe, the vapor core temperature is less than surface temperature but for condenser section, the behavior is opposite, the vapor core temperature is higher than the surface temperature. This difference between the core and surface temperatures for both evaporator and condenser sections increased with increased filling ratio and decreased with increased tilt angle.

Table 1: Heat Pipes Specifications

First System					
No.	Inner Diameter(mm)	Outer Diameter(mm)	Evaporator length(mm)	Condenser length(mm)	Total length(mm)
1	14	16	1150	200	1350
2	14	16	1150	300	1450
Second System					
3	14	16	1150	200	1350
4	14	16	1150	150	1450

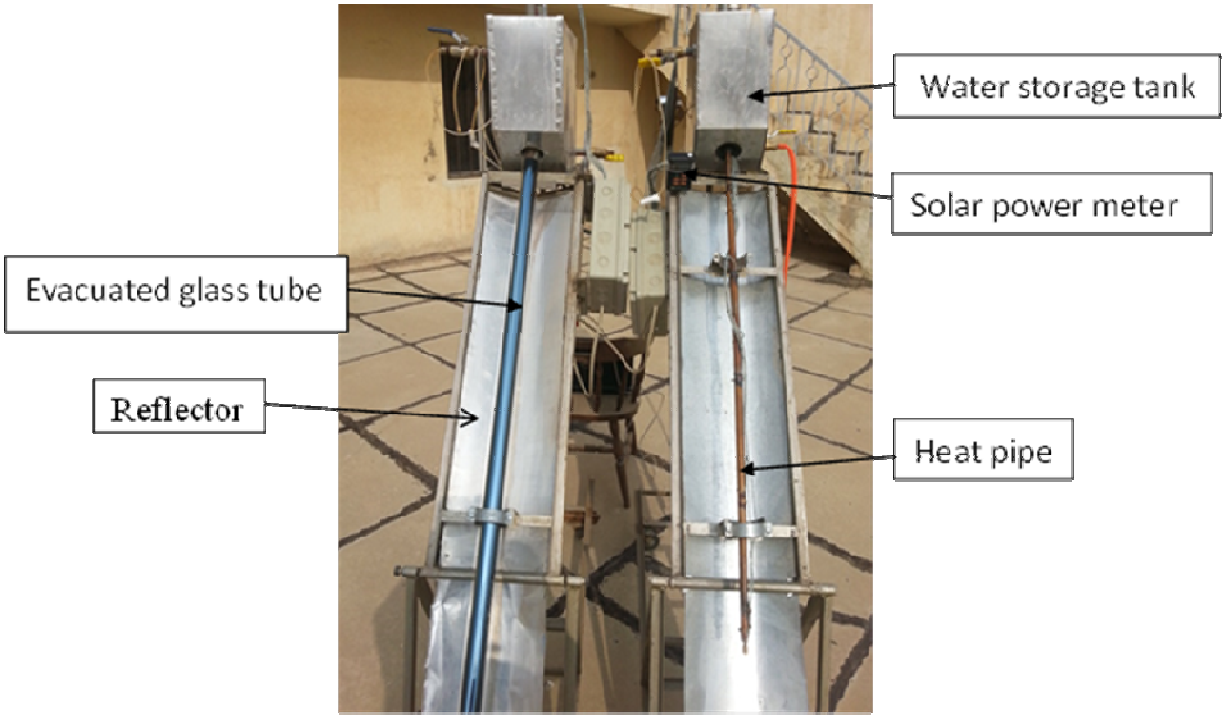


Fig. 1 Photographically of the experimental rig (in the right, the heat pipe before adding the evacuated tube while in the left, the evacuated was added).

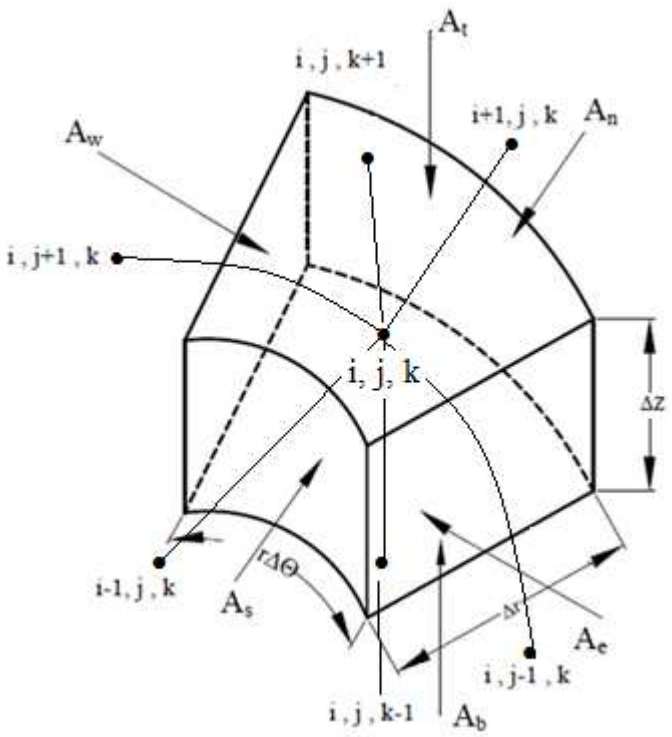


Fig. 2 General node for thermal resistance nodes

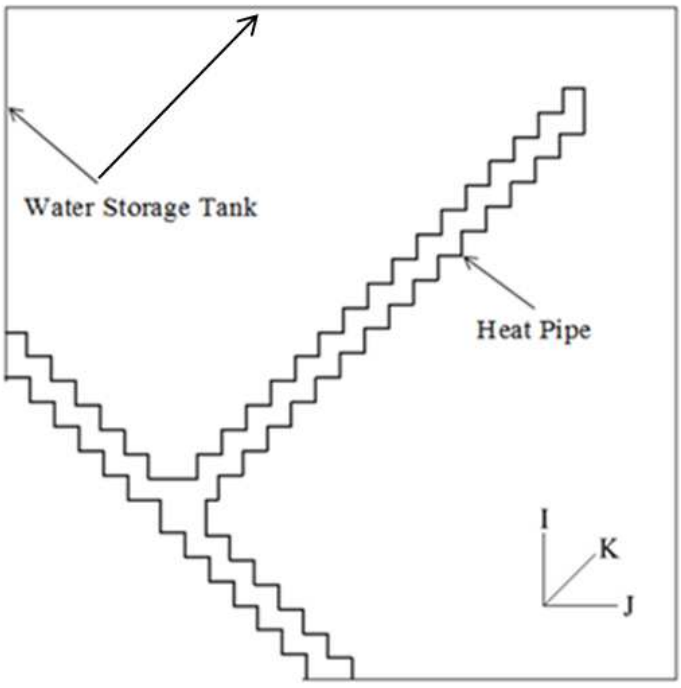
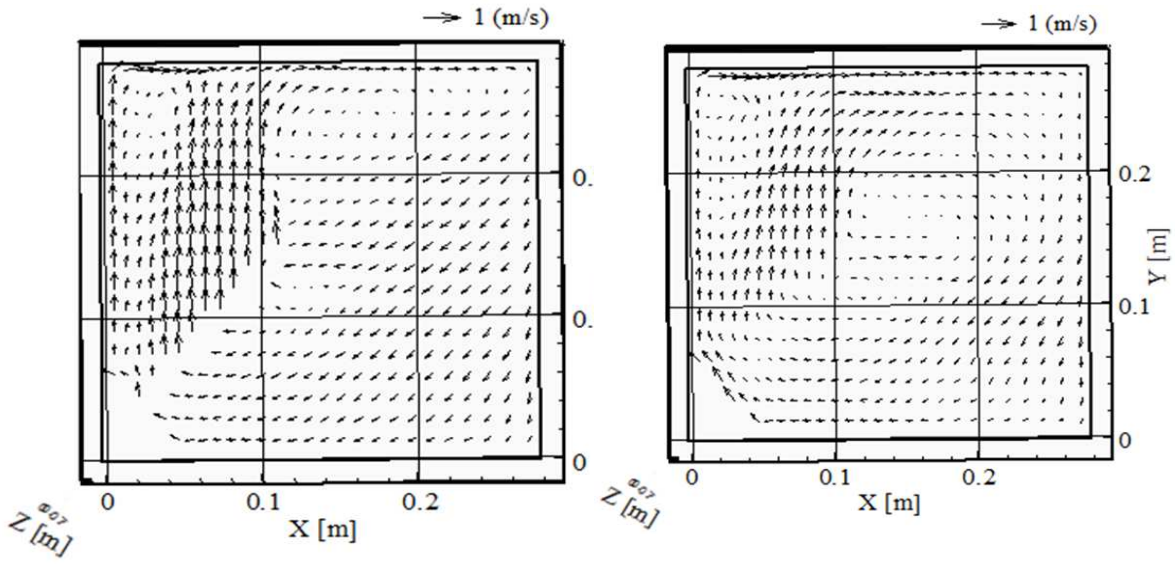


Fig. 3 Schematic plot of stair grids of the heat pipe inside water tank



(a) (b)

Fig. 4: Velocity vector in water storage tank, FR=50%, θ=45°, a)z=3.5 cm, b) z=4.5 cm.

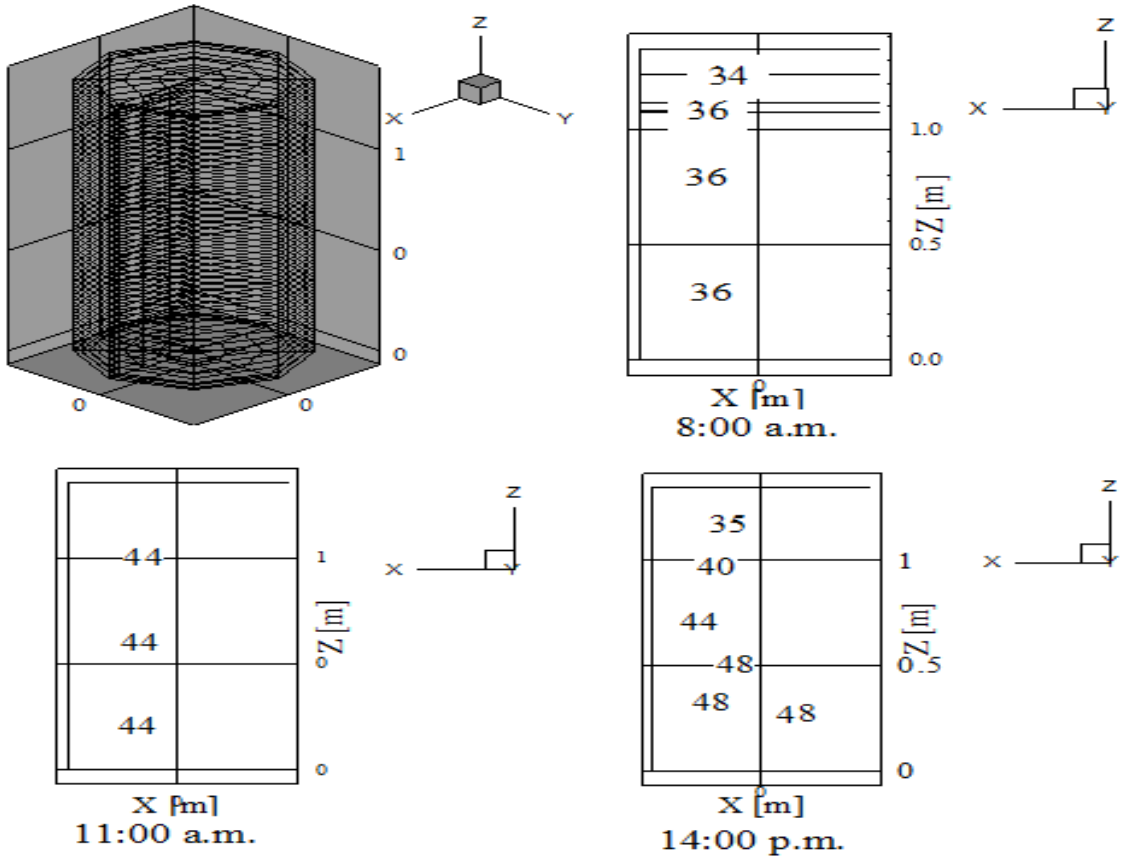


Fig. 5: Temperature contours for heat pipe FR=0.5, θ=45º

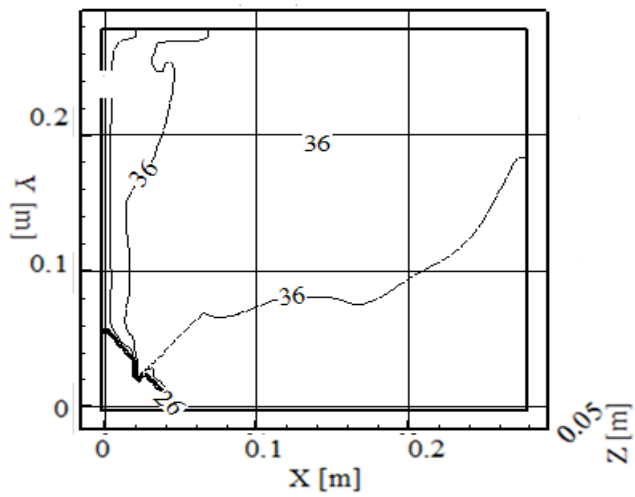


Fig. 6: Temperature contours inside water tank FR=50%, θ=45°

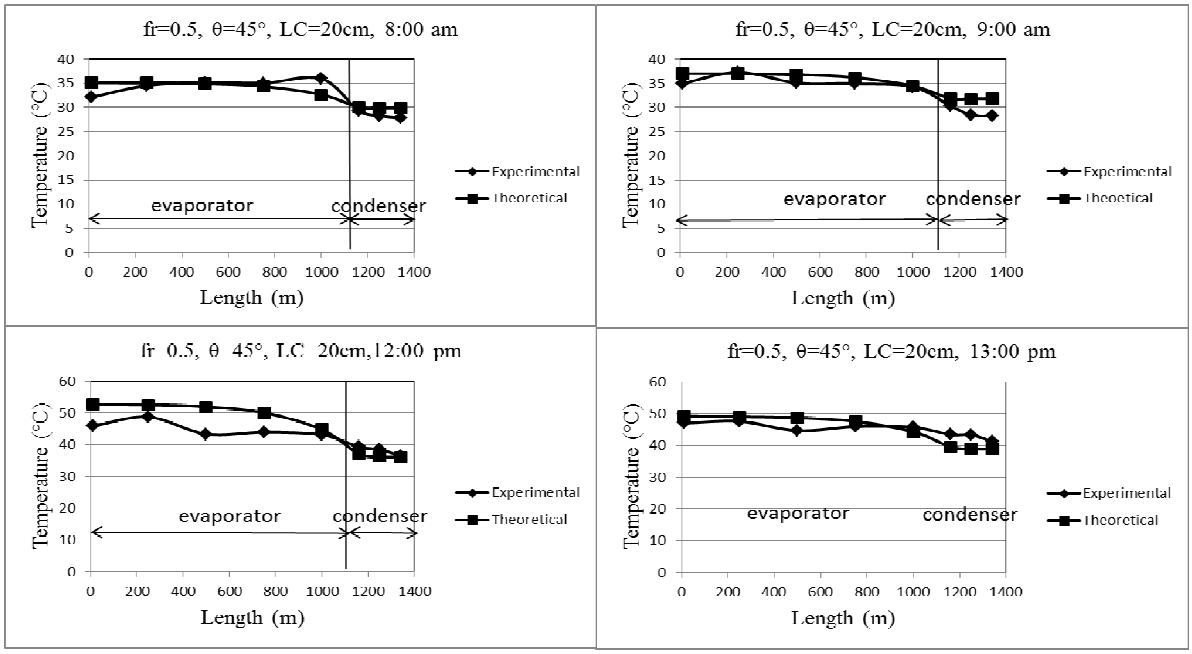


Fig. 7: Comparison between experimental and numerical results for outer wall heat pipe temperature with length

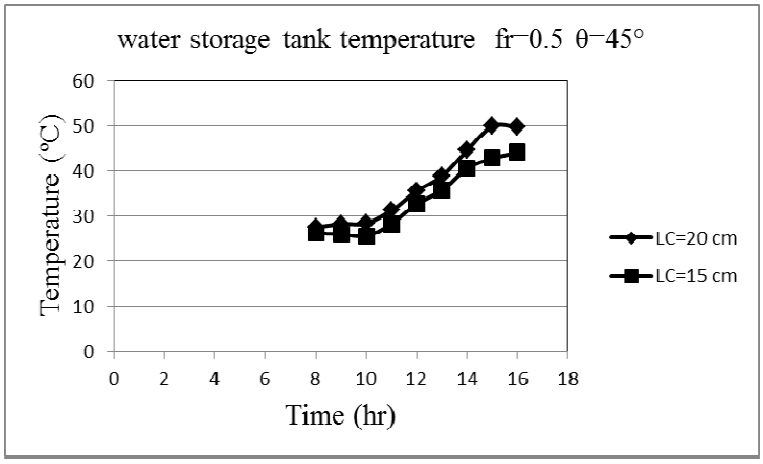


Fig. 8: Variation of water storage tank and outlet temperature with time

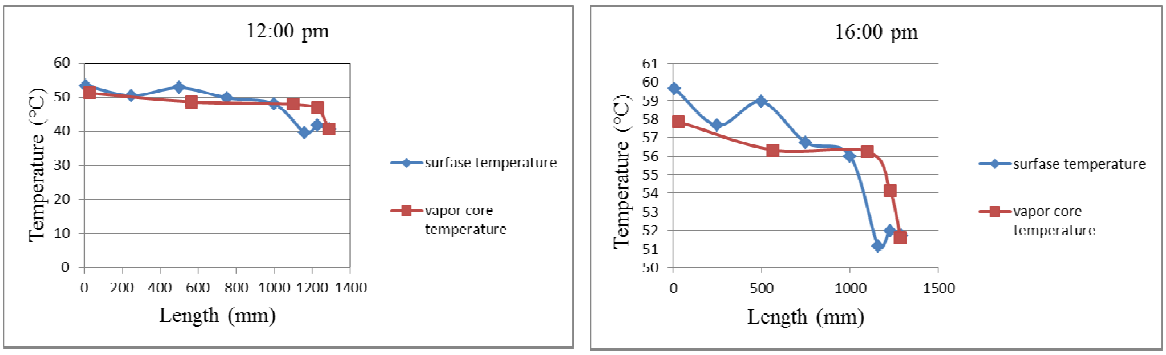


Fig. 9: Variation of vapor core temperature along the heat pipe during operation period

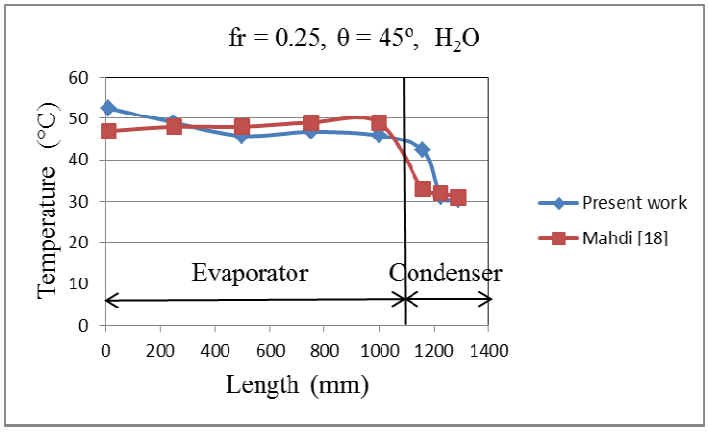


Fig. 10: Comparison of outer surface temperature along thermosyphon with other worker

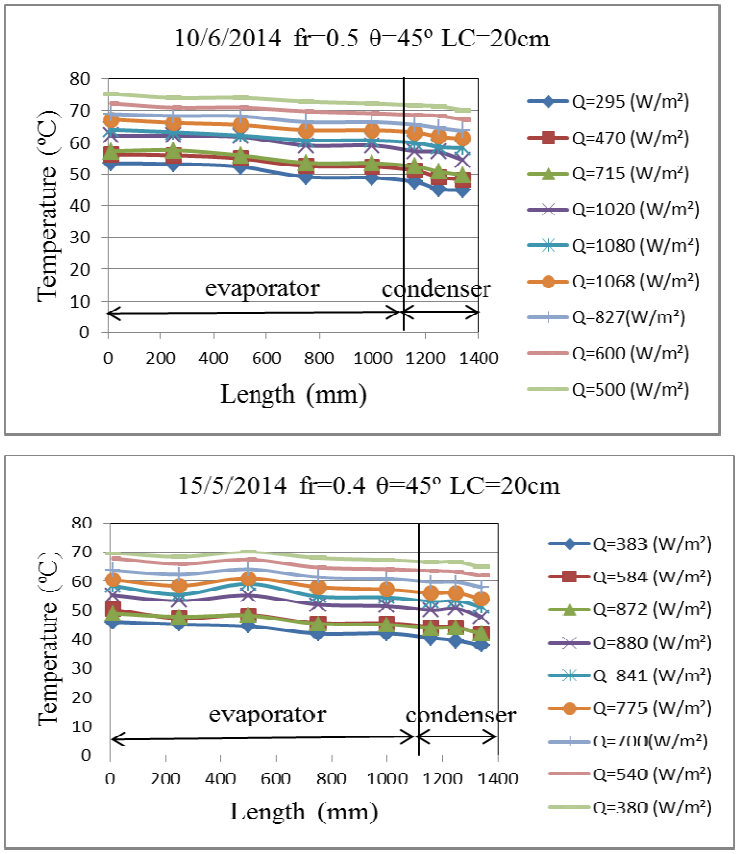


Fig. 11: Variation of outer surface temperature of heat pipe with length with different FR

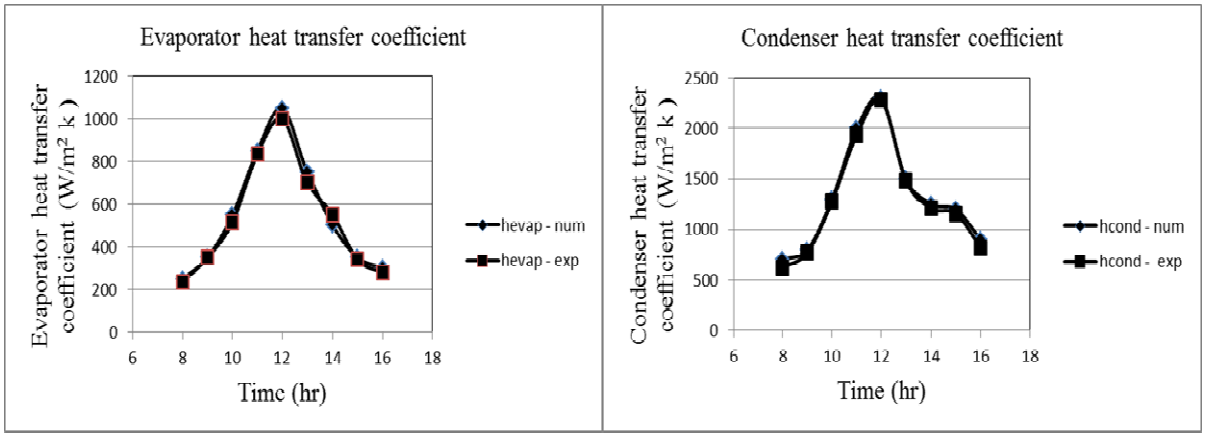


Fig. 12: Evaporator and condenser heat transfer coefficient, FR=50%, θ=60°

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