INVESTIGATION OF HEAT PIPES AS A MEANS OF HEAT TRANSFER DEVICE FOR SOLAR THERMAL APPLICATIONS

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Abstract

Experimental investigation of an axial transient thermal performance of heat pipes has been carried out. An experimental apparatus was designed and constructed to produce heat pipes. Two types of heat pipes, with wick and wickless, in lengths of 1m, 1.3m and 1.6m were produced. The inclination angle of the heat pipe with respect to the horizontal ground level was varied in angles of 7°, 8°, 9° and 10°. The influence of the parameters has been observed for various configurations of the experiment to provide vital information on the performance of the heat pipes. The set up includes a hot water storage to model the heat source. Copper heat pipe with internal and external diameters of 11.7mm and 12.7mm respectively and cold water storage were used to model the heat load. Thermocouples were attached at different points to measure temperature. The initial temperature of the hot storage was 80°C. Axial temperature gradient of the heat pipes for the different configurations was measured and recorded using a data logger. Maximum condenser temperatures of 77°C and 72.29°C are obtained for heat pipes with wick for 1 m and 1.3 m respectively. For wickless heat pipes, maximum condenser temperatures of 72.37°C and 71 °C are obtained for 1 m and 1.3m respectively. These results are for the set up when there is no heat load at the condenser section. But when there is heat load at the condenser section, maximum condenser temperatures of 47.26°C and 45.62°C for heat pipes with wick are obtained for 1 m and 1.3 m respectively. For wickless heat pipes, maximum condenser temperatures of 39.55°C, 36.5 °C and 29.83°C are obtained for 1 m, 1.3 m and 1.6 m respectively. As a result of the experimental investigation, copper heat pipes can be employed to achieve low cost and compact heat exchangers for the application of solar thermal systems.

Keywords: heat pipe; experimental investigation; transient; thermal performance; wick.

1. Introduction

Several researchers have carried out a number of researches on various means of efficient energy transportation devices. The researchers develop from a simple to highly sophisticated systems to ensure the effective transfer of energy obtained from different sources. One of the researches that came out with fruitful output was the development of heat pipes as a means of heat transportation device. Over the past 30 years (Zou and Faghi, 1998), extensive studies have been conducted in order to provide a thorough understanding of the heat pipe operation and appropriate design schemes for practical applications.

As a highly-effective heat transfer element, heat pipes have gradually recognized, and are playing a more and more important role in almost all industrial fields. A heat pipe is an evaporation-condensation device for transferring heat in which the latent heat of vaporization is exploited to transport heat over long distances with a corresponding small temperature difference. The heat transport is realized by means of evaporating a liquid in the heat inlet region (called the evaporator) and subsequently condensing the vapor in a heat rejection region (called the condenser). A heat pipe is basically a sealed cylinder tube containing a wick structure lined on the inner surface and a small amount of fluid such as water at the saturated state. Closed circulation of the working fluid is maintained by capillary action and /or bulk forces (Yunus, 2002). In this study two types of heat pipes have been used, wickless and heat pipe with wick. In the case of thermosyphon the condensate is returned to the evaporator by gravitational force where as in the basic heat pipes a wick, constructed from a few layers of fine gauze, is fixed to the inside surface and capillary forces return the condensate to the evaporator(Dunn P. and Reay, 1994).

According to (Feng Y. et.al., 2002) compared with other convectional heat exchangers, heat pipe heat exchanger has many advantages, i.e. large quantities of heat transported through a small cross-sectional area with no additional power input to the system; less pressure drop of fluid; high reliability; simpler structure and smaller volume.

2. Methodology and Experimental Set up

2.1. Design and construction of heat pipes

The first task to produce a heat pipe was to design and construct a system which could easily be able to produce a heat pipe. A geometrically suitable device was designed and constructed successfully. Copper pipes were welded together to form the system shown in Figure (1) which was found to be appropriate in producing a heat pipe. Then vacuum valves were used to connect this arrangement with a pressure gauge, vacuum pump, pipe line that connect one end of the copper pipe, pipe line that connects to the working fluid.

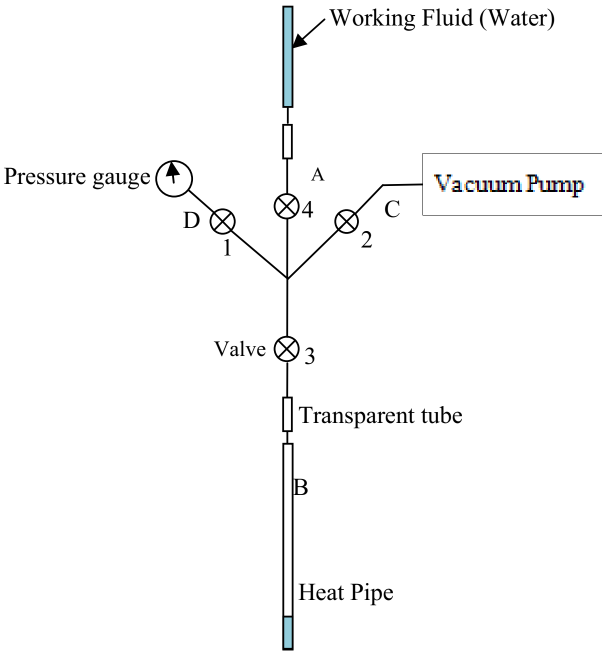

Figure 1: Schematic of the filling assembly

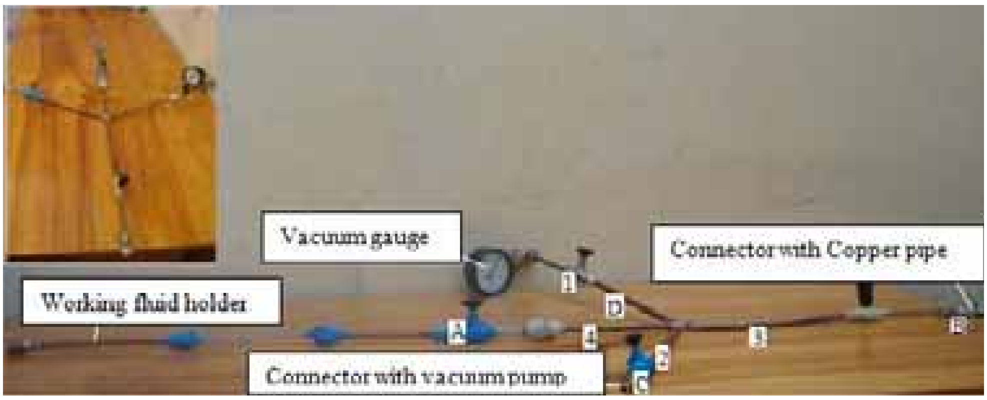

Figure 2: constructed device

With the help of the device, heat pipes with the following specifications were prepared for the experiment.

Table 1: Parameters of heat pipes (Heat pipe experimental configurations)

Material specification	Wickless heat pipe	Heat pipe with wick
Effective length(m)	1,1.30 & 1.60	1 and 1.30
Outside diameter(mm)	12.7	12.7
Wall thickness(mm)	1.00	1.00
Wall material	Copper	Copper
Wick structure		Mesh
Wick material		Copper
Working fluid	Distilled Water	Distilled Water
Liquid filled volume (%)	25	25
Evaporator section length(mm)	30	30
Condensation section length (mm)	25	25
Heat Pipe Inclinations	7^0,8^0,9^0 & 10^0	7^0,8^0,9^0 & 10^0

2.2. Data collection

Once heat pipes with different parameters were produced, experimental set up was prepared. An experiment was conducted for the various configuration options (length, wick and inclination angle); the collected data were used to observe the performance characteristics of the heat pipes.

2.2.1. Effect of varying length

Maintaining the other parameters constant the variation in length (1.00, 1.30 and 1.60 m) of heat pipes were considered during the experiment. These variations are for both the wickless and with wick heat pipes. The effect of this variation is observed on the heat transfer capability of the heat pipe.

2.2.2. Effect of using wick

Wickless heat pipes (thermosyphon) and heat pipes with wick were produced and experimental set up was prepared. Then experimental investigation was conducted to observe the variation in the thermal behavior of the two types of heat pipes.

2.2.3. Effect of inclination angle

An adjustable structure was made to observe the effect of inclination of the heat pipes' thermal performance. In all cases the angular variations considered were 7^0, 8^0, 9^0, 10^0.

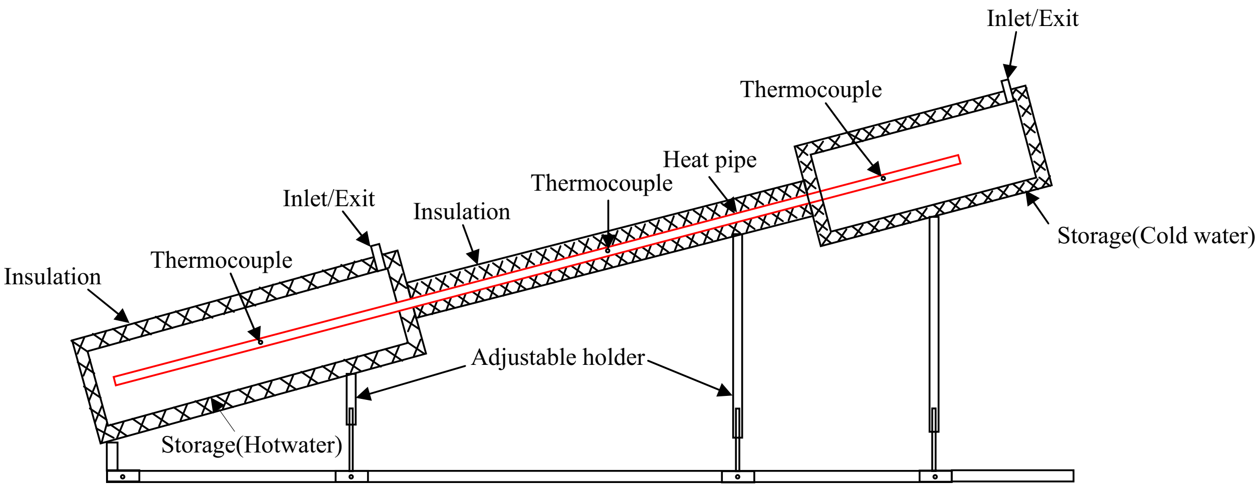

Figure 3: Schematic diagram of the overall arrangement of the system

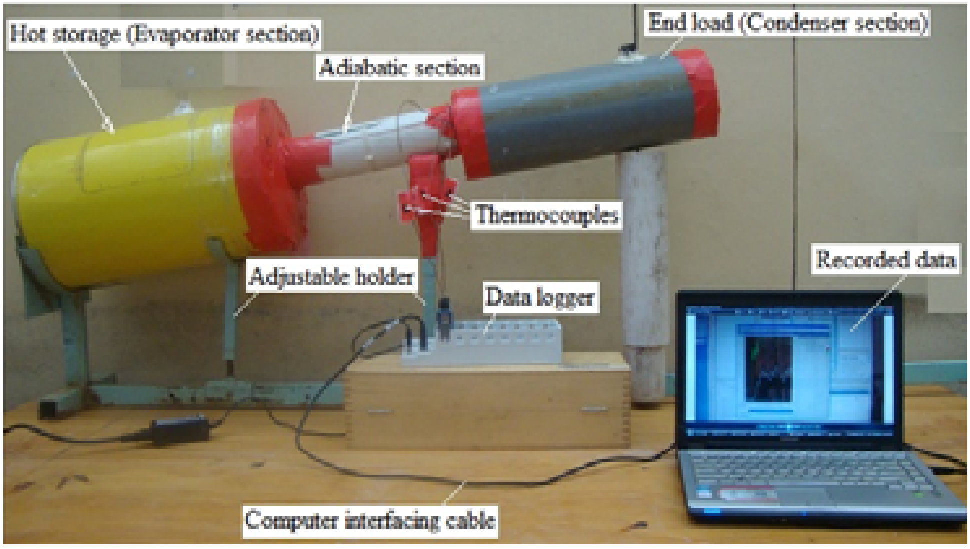

Figure 4: Experimental setup of the system

2.3. Data analysis

After the completion of the experimental process, data was collected and organized according to a certain set of category which was helpful to analyze the results. Then, the result was analyzed using different graphs for each category.

2.4. Instruments used

A number of materials have been used to carry out the study. A vacuum pump with a capacity of creating a vacuum pressure up to 200 mm-Hg ≈ 26.62 kPa has been used to evacuate the copper pipe. A hand press device and an oxyacetylene welding were used to seal heat pipe ends. A container made of stainless steel and insulated with a fiber glass was used as a heat storage material. A structure made of rectangular steel tube was built to hold the overall experimental materials. Thermocouples were attached at different points on the heat pipe and a data logger from the National Instrument by interfacing with a computer has been used to record the data in the experiment.

3. Thermal analysis

3.1. Working fluid compressibility

The compressibility of the working fluid in the heat pipe was checked using the following formulas (Chi, S.W., 1976).

$$M_v = \frac{Q_{max}}{A_v \rho_v h_{fg} \sqrt{\gamma_v R_v T_v}} \quad \text{(eq. 1)}$$

The calculated value is much smaller than 0.2, thus the fluid is incompressible and the vapor flow can be assumed as laminar flow.

3.2. Checking heat transfer limitations

Faghri (1995) stated that a heat pipe undergoes various heat transfer limitations depending on the working fluid, the wick structure, the dimensions of the heat pipe, and the heat pipe operational temperature. The type of the limitations restricting the heat transport capability of a heat pipe is determine by which limitation has the lowest value at the temperature considered. Heat pipe performance is affected by sonic, entrainment, capillary and boiling limitations. In this section, a theoretical calculation was made to check if these limitations are enough to cause a failure or an effect on the thermal performance of the heat pipe.

3.2.1. Capillary limitation

The maximum heat transfer rate due to the capillary limitation is expressed using the following formula (Chi, S.W., 1976).

$$Q_{c,max}=\left[\left[\left(\frac{\rho_l \sigma_l h_{fg}}{\mu_l}\right)\left(\frac{A_w K_l}{L_{eff}}\right)\left(\frac{2}{r_{c,e}}-\left(\frac{\rho_l}{\sigma_l}\right)gL_t \sin\psi\right)\right]\right] \qquad \text{(eq. 2)}$$

3.2.2. Sonic limitation

This equation is used to determine the maximum heat transfer rate due to the sonic limitation (Sami S. and Leblanc W., 1990).

$$Q_{s,\max} = A_v \rho_v h_{f,g}\left[\frac{\gamma_v R_v T_o}{2(\gamma_v+1)}\right]^{1/2} \qquad \text{(eq.3)}$$

3.2.3. Entrainment limitation

The following equation from (Sami S. and Leblanc W., 1990) is used to determine the conservative estimate of the maximum heat transfer rate due to entrainment of liquid droplet.

$$Q_{e,\max} = A_v h_{f,g}\left(\frac{\sigma\rho_v}{2r_{h,s}}\right)^{1/2} \qquad \text{(eq.4)}$$

3.2.4. Boiling limitation

This is used to determine the conservative estimate of the maximum heat transfer rate due to boiling limit. It is given by the following equation (Sami S. and Leblanc W., 1990).

$$Q_{b,\max} = \frac{2\pi L_{eff} K_e T_V}{h_{fg}\rho_V \ln\left(\frac{r_i}{r_v}\right)}\left[\frac{2\sigma}{r_n}-p_c\right] \qquad \text{(eq.5)}$$

From the above calculation results, all the limits are above the amount of the energy used in the evaporator (hot) section of the heat pipe. This means that the heat energy used at the evaporator section can be transferred without any failure of the limitations along the axis of the heat pipe.

3.3. Heat energy transferred to load

The amount of heat energy gained at the condenser section of the heat pipe is determined using the following formula (Incropera et.al, 2007):

$$Q = m_{cw} c_p \left(T_{cwf} - T_{cwi}\right) \qquad \text{(eq.6)}$$

3.4. System efficiency

The system efficiency is determined by dividing the heat energy gained in the condenser section to the heat energy given from the evaporator section. The maximum heat transfer rate is attained when the outside evaporator temperature is equal to the outside condenser temperature.

$$\eta_{sys} = \frac{m_{cw} c_p \left(T_{cwf} - T_{cwi}\right)}{m_{bw} c_p \left(T_{bwi} - T_{bwf}\right)} \qquad \text{(eq.7)}$$

3.5. Storage and adiabatic section insulation thicknesses

Figures (5) and (6) show schematic diagrams of the radial heat transfer and thermal resistance from the hot storage and heat pipe adiabatic section to the exposed surface ambient temperature respectively. Some assumptions have been considered to determine the insulation thickness to minimize heat transfer loss from the hot storage and heat pipe adiabatic section.

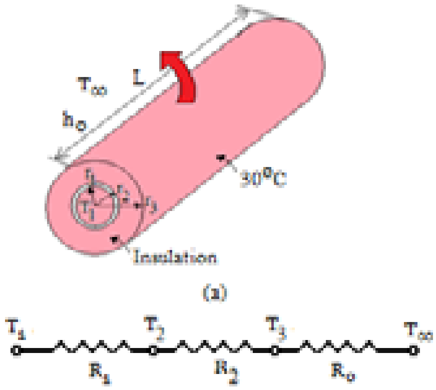

Figure 5 Radial heat transfers of the hot storage

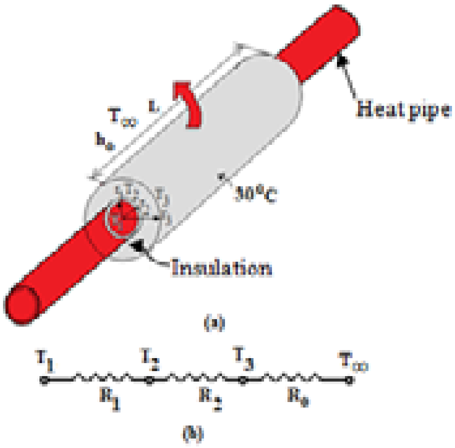

Figure 6: Radial heat transfers along the adiabatic section of the heat pipe

Calculations have been made based on the above thermal resistance network and the minimum thickness of insulation required were found to be 19 mm and 16 mm for the hot storage and heat pipe adiabatic section respectively.

4. Experimental procedure

Measurements of axial transient temperature gradients were carried out during the experiment in different configurations. The inclination angles that have been used for all tests were 7°, 8°, 9° and 10° from the horizontal, with the evaporator section located at the bottom and the condenser section at the top, Figures (3) and (4). Two type of heat pipes, wickless and with wick, have been used in lengths of 1.00, 1.30 and 1.60 m.

Once the experimental set up was prepared, thermocouples attached in the heat pipe were connected to the data logger. Temperature at three positions of the heat pipe (evaporator, adiabatic and condenser sections) were measured and recorded before the start of the experiment for some time. Then, boiling water was poured to the evaporator section while the condenser section was filled with cold water at ambient temperature. Two liters of water at ambient temperature was used as a load in the condenser section of the heat pipe. For heat pipe without end load the condenser section is exposed to the ambient temperature. This was done repeatedly for each configuration. Then, for each experiment, the temperature distribution along the heat pipe was measured and recorded. During each test run, the condenser, the adiabatic, the evaporator and the ambient temperatures were recorded every second.

In this study, boiling water with an initial temperature of 80°C was used as a heat source for each test. The benefit of using boiling water as a heat source instead of a convectional electric heater is that boiling water can easily supply a sudden heat load than a conventional electric heater (Wang 2009). Besides, the study deals with an investigation of heat pipes as a heat transport device for the application of solar thermal systems from a hot storage to a load at some distance. Hence, using boiling water as a heat source will be more close to the practical application instead of using conventional electric heater.

5. Results and Discussions

5.1. Effect of different lengths and angles of inclination on the axial transient temperature profile

5.1.1. Wickless heat pipes with no end load

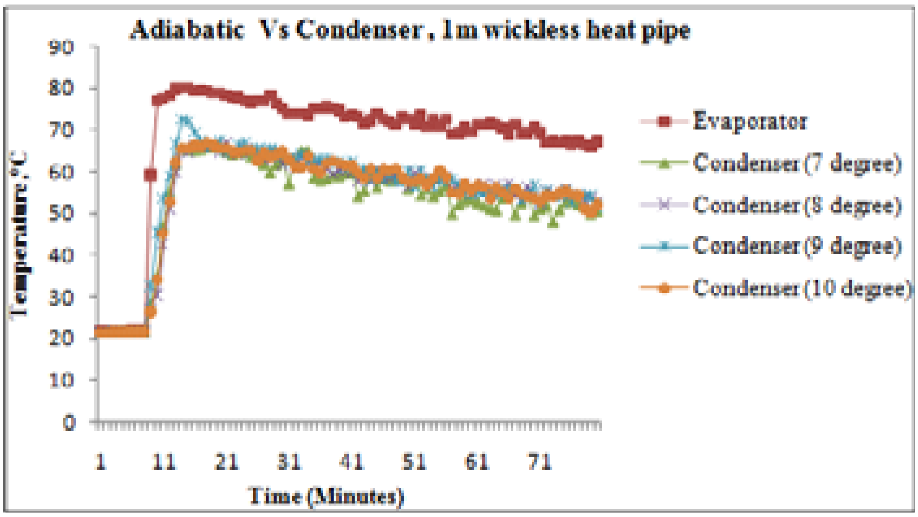

Figure 7: Comparison of condenser section temperatures for 1 m, wickless heat pipe

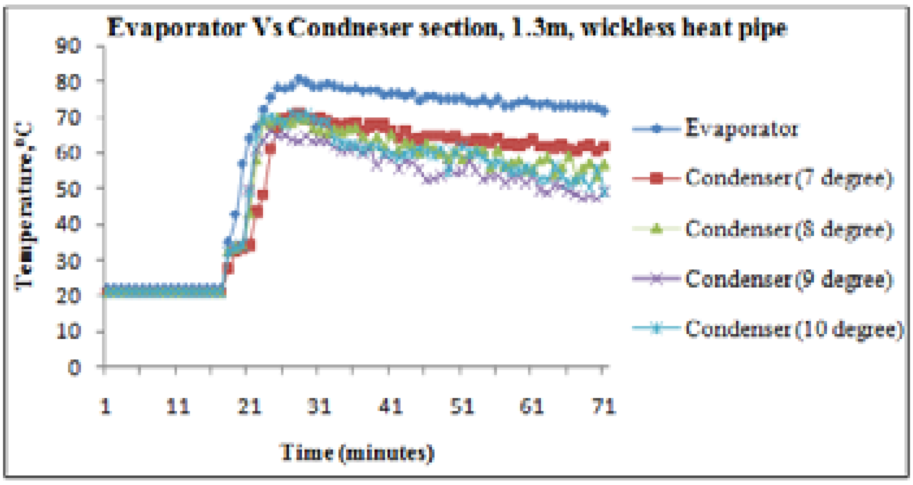

Figure 8: Comparison of condenser section temperatures for 1.3 m, wickless heat pipe

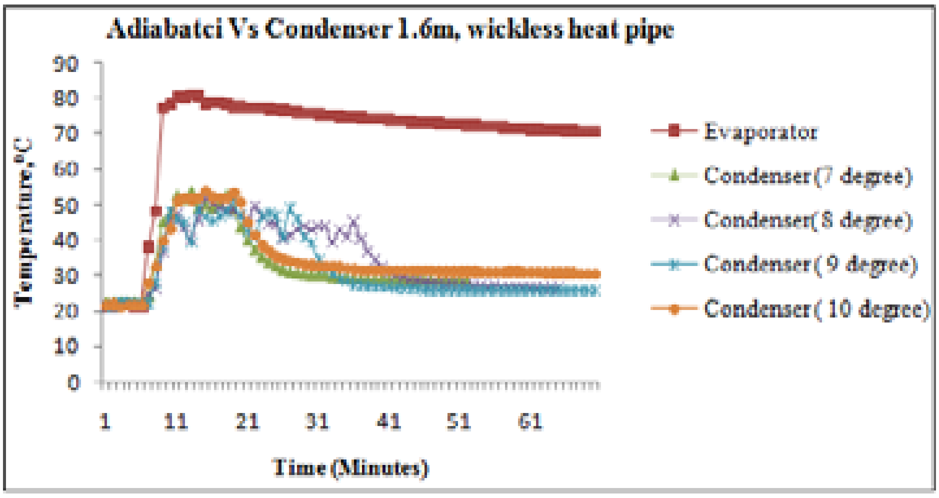

Figure 9: Comparison of condenser section temperatures for 1.6 m, wickless heat pipe

Figures 7-9 show the axial transient temperature gradient of wickless heat pipes at different inclination angles. In all cases, the temperature of the evaporator section has a sudden rise from the ambient temperature of 21°C to the maximum 80°C under a sudden heat load. Then, the temperature of the evaporator decreases gradually with time. The maximum temperature of the evaporator section on all configurations is the same as the boiling water poured into the hot storage. The temperature of the boiling water is controlled to maintain at the required temperature. The maximum temperatures of the condenser section achieved with such arrangement are 72.37°C, 71°C and 53.84°C for 1 m, 1.3 m and 1.6 m respectively. The temperature difference between the heat pipes condenser section for the different inclination angles is small. This implies that the thermal resistance of the heat pipes for each inclination angles is small and the heat pipes have nearly the same heat transport capacity for these inclination angles. From the results it can also be observed that the heat transport capacity of the heat pipes decreases as its length increases. This is mainly due to the increase in thermal resistance with increase in length which in turn reduces the heat transport capacity of the heat pipe.

5.1.2. Wickless heat pipes with end load

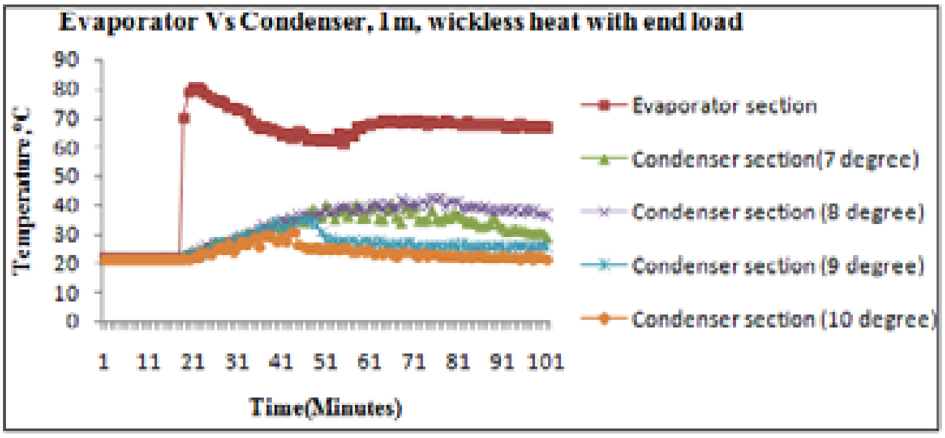

Figure 10: Comparison of condenser section temperatures for 1 m, wickless heat pipe with end load

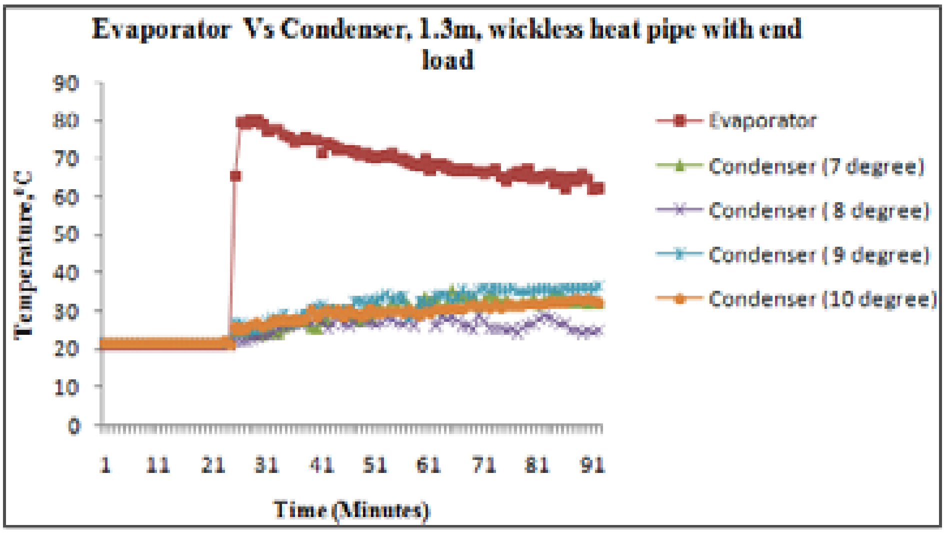

Figure 11: Comparison of condenser section temperatures for 1.3 m, wickless heat pipe with end load

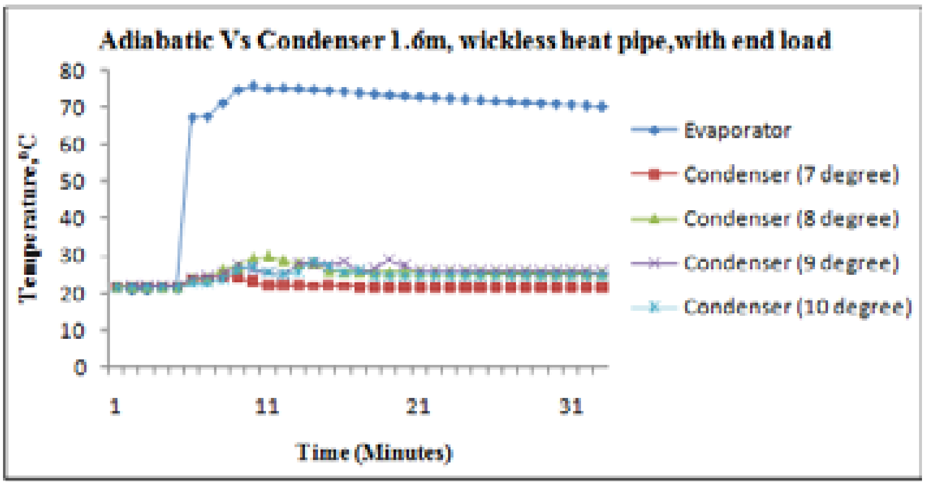

Figure 12: Comparison of condenser section temperatures for 1.6 m, wickless heat pipe with end load

Figures 10-12 show that with an initial evaporator temperature of 80^{o}C the maximum temperature of the condenser section achieved are 39.55^{o}C, 36.51^{o}C and 29.83^{o}C for 1 m, 1.3 m and 1.6 m respectively. The maximum efficiencies obtained using (eq.7) are 27%, 18.38% and 8.73% for 1 m, 1.3 m and 1.6 m respectively. The heat transfer capacity for the 1.6 m heat pipe is very low .i.e. this length is so long to transfer heat into the condenser section.

5.1.3. Heat pipes with wick without end load

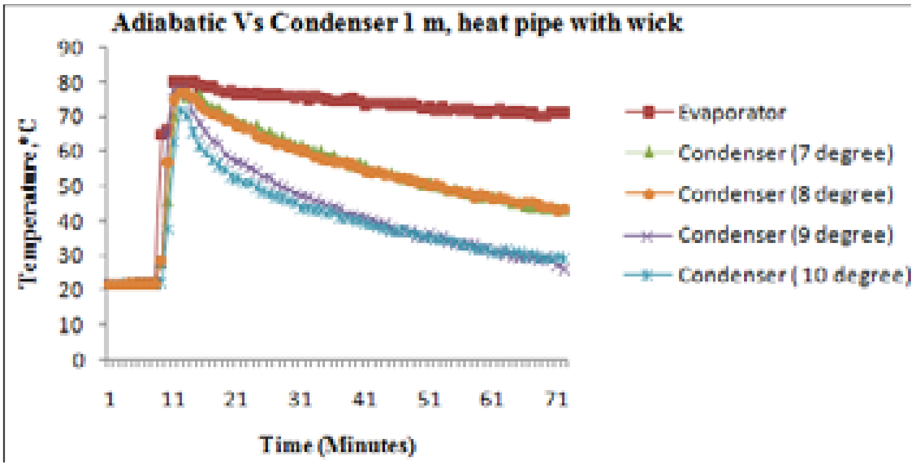

Figure 13: Comparison of condenser section temperatures for 1m, heat pipe with wick without end load

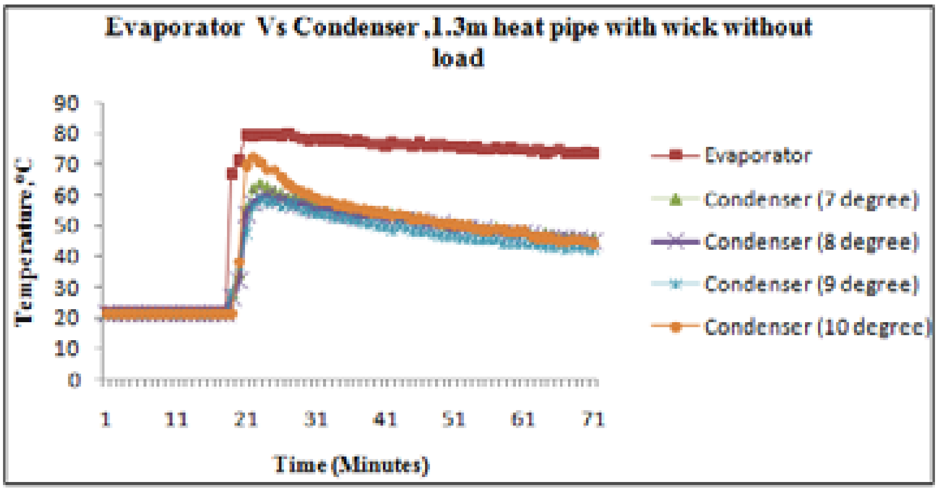

Figure 14: Comparison of condenser section temperatures for 1.3m, heat pipe with wick without end load

Figures 13 and 14 show a transient temperature gradient along the axis of 1 m and 1.3m heat pipes with wick. It can be noticed from the figures that the maximum condenser surface temperature is higher than that of the wickless heat pipe with the same lengths and inclinations angles. This implies that the heat pipe with wick has a lower thermal resistance and a higher heat transport capacity at the beginning of the experiment when the temperature of the evaporator is high. Afterwards when the evaporator temperature starts decreasing, the condenser temperature sharply drops to a lower temperature and the gap keeps on increasing with further decrease in temperature. This shows heat pipes with wick are very sensitive to temperature gradient compare to wickless heat pipes. This may happen due to the slow vapour and condensate flows with a decrease in temperature in addition to the obstruction of the wick which also increase as the temperature goes down. The maximum temperature of the condenser section achieved using these arrangements are 77°C, 72.29°C for 1 m and 1.3 m respectively. From Figure (13), it is observed that the heat transfer is decreasing with increasing in inclination angle. This is due to the negative effect of gravitational force to the flow of working fluid against gravity.

5.1.4. Heat pipes with wick with end load

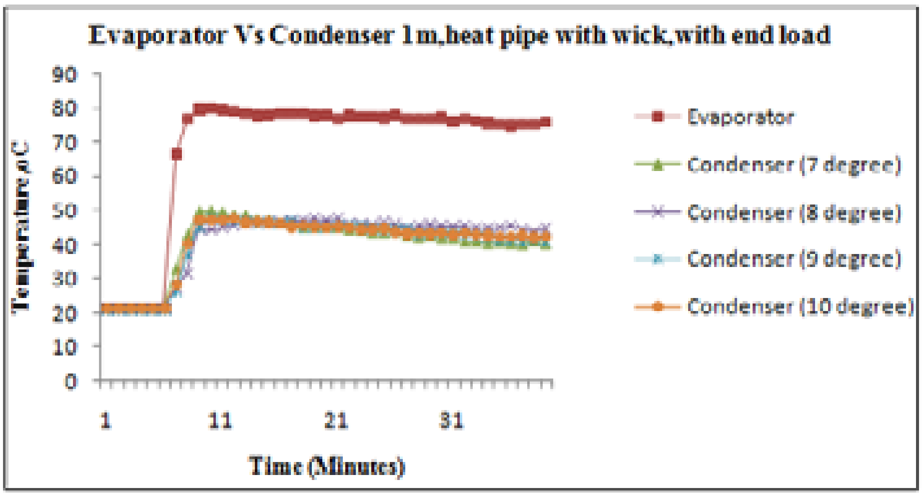

Figure 15: Comparison of condenser section temperatures for 1m, heat pipe with wick with end load

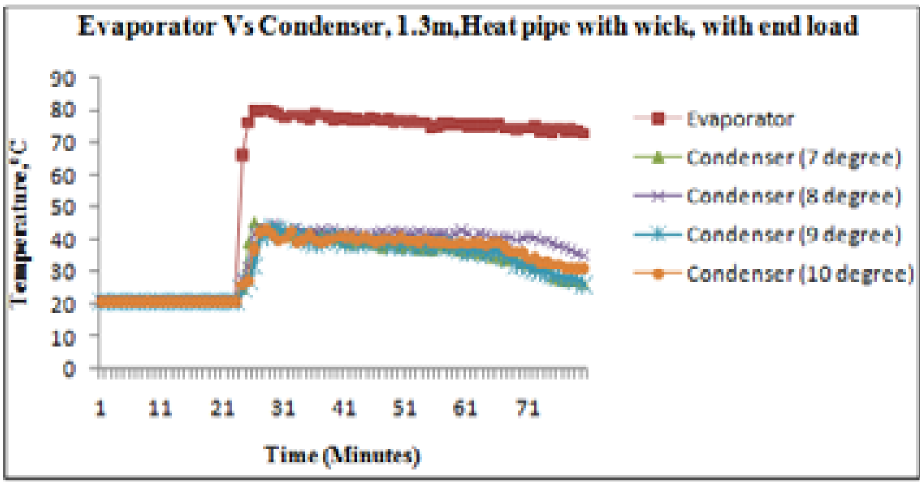

Figure 16: Comparison of condenser section temperatures for 1.3m, heat pipe with wick with end load

Figures 15 and 16 show the axial transient temperatures of 1 m and 1.3 m heat pipes. The maximum condenser temperatures obtained with such arrangement are 47.26 °C and 45.62°C for 1 m and 1.3 m respectively. The maximum efficiencies using (eq. 17) are 24.68 % and 20.95 %. By making comparisons between Figures 9-12 and Figures 13-16 it can be observed that unlike wickless heat pipes the variation in heat transfer capacity of heat pipes with wick are independent of inclination angles. In addition, heat pipes with wick have very good heat transfer capacity compare to wickless heat pipes.

6. Conclusion

The results of the experimentally measured values indicate that both types of heat pipes (Wickless and wick) can be used for the application of solar thermal systems. This is mainly from where the heat energy is stored to some distance with a small temperature difference as long as the temperature of the heat source does not drop very much below the operating temperature, 80°C. By making comparisons between Figures 9-12 and Figures 13-16 it is observed that unlike wickless heat pipes the variation in heat transfer capacity of heat pipes with wick are independent of inclination angles. In addition, heat pipes with wick have very good heat transfer capacity compare to wickless heat pipes. Thus, in the lengths considered heat pipes with wick are found to work better compared to wickless heat pipes.

Nomenclature and Symbols

$Q_{b, max}$	Boiling heat transport factor	k_l	Liquid thermal conductivity
$(Q)_{c,max}$	Capillary heat transport factor	μ_l	Liquid Viscosity
P_c	Capillary Pressure	m_{cw}	Mass of cold water
T_{cw}	Condenser cold water Temperature	T_0	Operating temperature
$r_{c,e}$	Effective capillary radius	r_1	Pipe inner radius
K_e	Effective conductivity	r_2	Pipe outer radius
L_{eff}	Effective length	R	Thermal resistance
$Q_{e, max}$	Entrainment heat transport factor	$Q_{s, max}$	Sonic heat transport factor
m_{bw}	Evaporator mass of boiling water	γ_v	Specific heat ratio
T_{bwf}	Final temperature of boiling water	C_p	Specific capacity of water
T_{cwf}	Final temperature of cold water	σ	Surface tension
g	Gravitational acceleration	η_{sys}	System efficiency
Q	Heat transfer rate	T_∞	Temperature of the environment
$r_{h,s}$	Hydraulic radius	L_t	Total pipe length
Ψ	Inclination angle	r_v	Vapor core radius
T_{bwi}	Intial temperature of boiling water	A_v	Vapor cross sectional area
T_{cwi}	Initial temperature of cold water	ρ_v	Vapor density
r_i	Internal radius of the pipe	R_v	Vapor gas constant
$h_{f,g}$	Latent heat of vaporization	T_v	Vapor temperature
ρ_l	Liquid density	A_w	Wick area

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