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EXPERIMENTAL STUDY OF PERFORMANCE OF SOLAR WATER HEATER BY USING HEAT PIPE

Vipul U Panara [1]***, Susheela V. Kumaril*** [2]***, Rushi B. Rawal*** [3]
[1]Student of M.E. Thermal Engineering at MIT, Piludara, *panara13vipul20@gmail.com*
[2]Asst. Professor in Mechanical Engineering Department at MIT, Piludara, *svkumari.me@spitcp.ac.in*
[3]Asst. Professor in Mechanical Engineering Department at GEC, Bhuj, *rushibmr1978@gmail.com*

Abstract

Today world is facing challenge of energy crisis. There are many ways of energy wastage in the form of heat in heat transfer applications. It is require recovering these waste heats by some means. The heat pipe is one of the remarkable achievements of thermal physics and heat transfer engineering because of its unique ability to transfer heat over large distances without considerable losses. The present study includes experimental investigation of performance of heat pipe with different working fluids. From that result selection of working fluid for solar water heating application has been done. After that, experimental set-up of heat pipe solar water heater is made. From experiments on this set-up, performance of heat pipe solar water heater has been investigated for whole day. From experimental data, analysis and comparison of result with domestic water heater has done. Experimental result shows that heat transfer rate from collector to water tank is large in case of heat pipe collector compared to conventional solar water heater. Thus the efficiency of solar water heater with heat pipe is about 7% more than the conventional water heater.

Keyword*- Heat pipe, solar collector, efficiency, heat transfer enhancement, working fluids*

I. INTRODUCTION

Heat pipes have been considered as promising means for effective heat transfer in energy transport and storage systems in medium-high temperature range. Heat pipes are two-phase flow heat transfer devices where processes of liquid to vapor and vice versa circulate between evaporator to condenser with high effective thermal conductivity. As its operation involves phase changes (i.e., evaporation and condensation) large amounts of heat can be transferred. Due to the high heat transport capacity, heat exchangers with heat pipes have become much smaller than traditional heat exchangers in handling high heat fluxes [1]. The heat pipe is a self-contained structure that achieves very high thermal energy conductance by means of two-phase fluid flow with capillary circulation. Heat added to the evaporator is transferred to the working fluid by conduction and causes vaporization of the working fluid at the surface of the capillary structure. Vaporization causes the local vapor pressure in the evaporator to increase and vapor to flow toward the condenser thereby transporting the latent heat of vaporization [2].

Heat pipe technology has found increasing applications in enhancing the thermal performance of heat exchangers in microelectronics; energy savings in heating, ventilating, and air conditioning (HVAC) systems for operating rooms, surgery centres, hotels, clean rooms etc.; temperature regulation systems for the human body; and other industrial sectors including spacecraft and various types of nuclear reactor technologies as a fully inherent cooling apparatus [3].

Several studies on heat pipe solar collectors are reported in the literature due to their lower effective cost. Hussein et al. [4] investigated the different design parameters of the natural circulation for two phase closed thermosyphon flat plate solar water heaters using the verified expanded model. Abreu et al. [5] carried out an experimental analysis of the thermal behavior of the two-phase closed thermosyphon for compact solar domestic hot water systems with unusual geometries for the retention of liquid characterized by a semicircular condenser and a straight evaporator.

Riffat et al. [6] constructed a thin membrane flat plate heat pipe solar collector and developed an analytical model that was used to simulate heat transfer processes occurring in the collector and calculate its efficiency.

Koffi et al. [7] studied batch type two-phase thermosyphon solar water heater where the vapor produced in the thermosyphon system passes through coil heat exchanger coil in the tanker. The system has been validated both theoretically and experimentally and a mean collector efficiency of 58% was obtained.

Taherian et al. [8] conducted both numerically and experimentally a two phase thermosyphon solar collector with distilled water working fluid. They found to have small temperature variation on absorber, cover and heat pipe during the test hours.

Joudi and Witwit [9] improved on the performance of a thermosyphon by adding a separator in the adiabatic section and adding on a three-layered wick in the evaporator section of three lengths of 20 mm copper pipe filled with water. They noted that the presence of an adiabatic separator caused a remarkable improvement in all heat pipes tested for all lengths and inclination angles and the addition of screen wick at the evaporator can raise the heat pipe evaporator and adiabatic temperature.

Chun et al. [10] presented an experimental setup testing different heat pipes with and without wick structure; different working fluids (water, methanol, acetone and ethanol) and different absorber surface treatment were compared. Brahim et al. [11] conducted a parametric study testing three working fluids on wicked assisted heat pipe solar collector. Results show that acetone heat pipe solar collector outperformed methanol and ethanol heat pipe solar collector. Optimal design parameters are also founded. Ismail et al. [12] used a methanol wicked heat pipes only in the evaporator region. They note that the condenser having a slope greater than the evaporator can improve the collector performance compared to a conventional solar collector.

II. PRINCIPLE OF HEAT PIPE

Heat pipes are two-phase flow heat transfer devices where processes of liquid to vapor and vice versa circulate between evaporator to condenser with high effective thermal conductivity. The three basic components of a heat pipe are

1. The container
2. The working fluid
3. The wick or capillary structure

A heat pipe operates within a two-phase flow regime as 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. Heat added to the evaporator is transferred to the working fluid by conduction and causes vaporization of the working fluid at the surface of the capillary structure. Vaporization causes the local vapor pressure in the evaporator to increase and vapor to flow toward the condenser thereby transporting the latent heat of vaporization. Figure 1 shows the operation of heat pipe.[1]

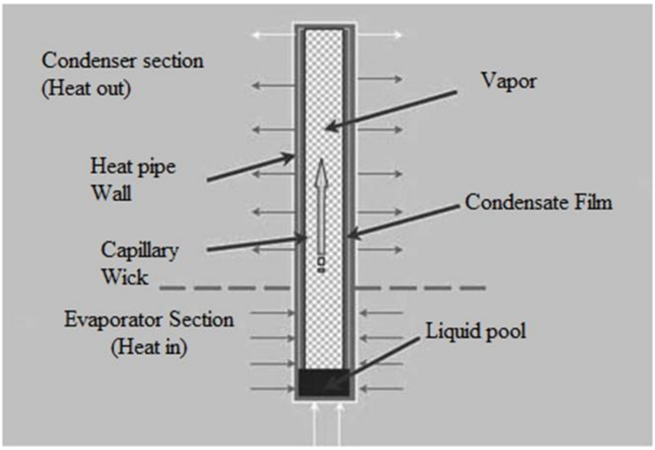

Figure 1 Heat pipe concept[1]

Since energy is extracted at the condenser, the vapor transported through the vapor space is condensed at the surface of the capillary structure, releasing the latent heat. Closed circulation of the working fluid is maintained by capillary action and/or bulk forces.

The function of the container is to isolate the working fluid from the outside environment. It has to therefore be leak proof, maintain the pressure differential across its walls, and enable transfer of heat to take place from and into the working fluid.

A high latent heat of vaporization of working fluids is desirable in order to transfer large amounts of heat with minimum fluid flow, and hence to maintain low pressure drops within the heat pipe. The thermal conductivity of the working fluid should preferably be high in order to minimize the radial temperature gradient and to reduce the possibility of nucleate boiling at the wick or wall surface.

The prime purpose of the wick is to generate capillary pressure to transport the working fluid from the condenser to the evaporator. It must also be able to distribute the liquid around the evaporator section to any area where heat is likely to be received by the heat pipe. Often these two functions require wicks of different forms. The selection of the wick for a heat pipe depends on many factors, several of which are closely linked to the properties of the working fluid.

III. SELECTION OF HEAT PIPE

A design of gravity assisted wicked heat pipe for solar application is presented in this work. Container used for heat pipe is copper because of its high thermal conductivity and copper wire mesh wick is used. Selection of working fluid is based on basic experiments with different working fluids. Figure shows the heat pipe which is used in this work.

Figure 2 Heat pipe used

Thermal resistance with respect to input heat is shown in Table 1 for water, Methanol and Acetone.

Table 1. Thermal resistance according to heat input

		Thermal Resistance (^{0}C/W)		
		Water	Methanol	Acetone
Input Power(W)	10(W)	1	0.5	0.4
	20(W)	0.9	0.7	0.6
	50(W)	0.6	0.5	0.5

In solar application, the efficiency of heat transport capacity is also important selection criteria as for the working fluid. Hence, the use of working fluid with higher latent heat is beneficial. From Above result it is shown that thermal resistance of water, methanol and acetone are approximately same for high input power. But Water has higher heat transport capacity than methanol and acetone and water is compatible with copper which is used as a container. So water is selected for Heat pipe as working Fluid in this work.

IV. MODELING OF COLLECTOR AND HEAT PIPE

First of all mathematical modelling of flat plate collector and heat transfer limits of heat pipe are calculated.

4.1 Modelling of Liquid Heating Flat Plate Solar Collector

Modelling of Liquid Heating Flat Plate Solar Collector it is most common method to estimate the absorbed solar radiation and the thermal behaviour of conventional flat plate collector by a single-node. Single-node modelling is representing all type of heat loss by a single overall heat transfer coefficient. In this thermal analysis of flat plate collector, substantial assumption has been considered to simplify the thermal and optical behaviour of the collector. Though transient and unpredicted nature of solar energy in a given period, steady state heat transfer condition is the initial concept used in the modelling of conventional flat plate collector. To get the mathematical model which can describe the temperature behaviour of the collector, the following assumptions simplifies the complexity with minimum error (Duffie and Bekeman, 1991).

- Temperature gradient through the cover is negligible
- There is one-dimensional heat flow through the back and side insulation and through the cover system.
- There is no absorption of solar energy by cover
- There is negligible temperature drop through the cover
- The sky can be considered as a blackbody for long-wavelength radiations at an equivalent sky temperature
- Temperature gradient around the tubes, through the fluid is negligible
- Loss through front and back are to the same temperature
- Dust and dirt on the collector are negligible
- Shading of the collector absorber plate is negligible

Figure 3 shows thermal performance of typical flat plate solar collector
Where:
- Radiation exchange between ground and cover, 1
- Radiation exchange between sky and collector cover, 2
- Ambient wind convection heat loss, 3
-Convection heat exchange between absorber plate and cover by the hot air in the gap, 4
- Radiation exchange between cover and absorber plate, 5
- Absorbed solar radiation by the absorber plate, 6

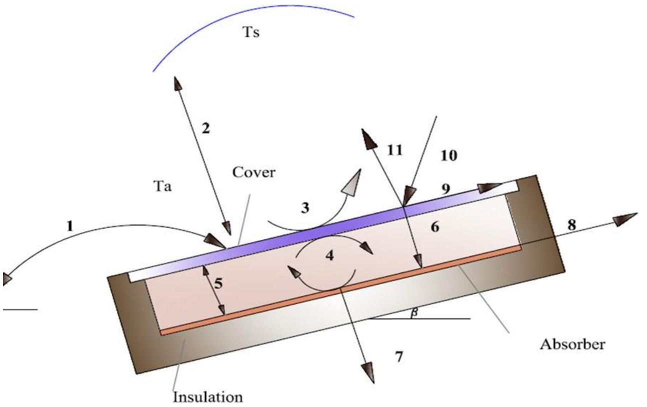

Figure 3 Thermal performance of typical flat plate solar collector

Figure 4.6 shows the thermal resistance network to calculate the total heat transfer coefficient, Uo for a single cover flat plate solar collector. From the thermal network shown in Figure 7 the top heat loss can be written,[13]

$$q_{loss} = Uo\ Ap\ (Tpm - Ta) \quad (1)$$

Where, Uo = Overall lose Coefficient
Ap = Area of absorber plate
Tpm = Absorber plate temperature
Ta = Temperature of the surrounding air

The heat loss from the collector is the sum of the heat loss from the top, bottom and side. Thus, Overall heat transfer coefficient,

$$U_o = U_t + U_b + U_s \quad (2)$$

For Top loss coefficient, the heat transferred by convection and radiation between (i) the absorber plate and the cover, (ii) the cover and surrounding must be calculated by following quation,

$$\frac{qt}{Ap} = h_{p\text{-}c}\,(T_p\text{-}T_c) + \frac{\sigma\,(Tp4 - Tc4)}{\left(\left(\frac{1}{\varepsilon p}\right)+\left(\frac{1}{\varepsilon c}\right)-1\right)} \quad \ldots (15)$$

$$\frac{qt}{Ap} = h_w\,(T_c\text{-}T_a) + \sigma\,\varepsilon_c\,(T_c^4\text{-}T_{sky}^4) \quad (16)$$

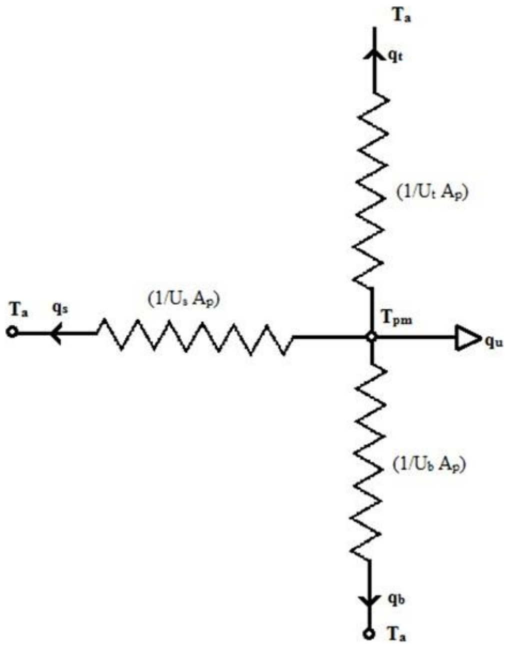

Figure 4 Thermal network for one cover flat plate collector [13]

Where σ is the Stefan-Boltzman constant and is equal to 5.6697 × 10-8 W/m2 k4, ε is the emissivity and Ts is the sky temperature. The sky temperature can be calculated using the correlation

$$Tsky = 0.0552\ (Tair)^{1.5} \quad (17)$$

The convective heat transfer coefficient (hp-c) between the absorber and the cover can be calculated using the concept of natural convection between flat parallel plates. The free convective heat transfer is related to two or three dimensionless parameters: Nusselt number Nu; the Rayleigh number Ra; and the Prandtle number Pr.

$NuL = 1$; $RaL \cos\beta < 1708$
$NuL = 1 + 1.446\,(1 - 1708/ RaL \cos\beta)$; $1708 < RaL \cos\beta < 5900$
$NuL = 0.229\,(RaL \cos\beta)^{0.252}$; $5900 < RaL \cos\beta < 9.23 \times 104$
$NuL = 0.157\,(RaL \cos\beta)^{0.285}$; $9.23 \times 10^4 < RaL \cos\beta < 10^6$ …(18)

Where,

$RaL \cos\beta = 9.81 \times 1/Tpc \times (Tp\text{-}Tc) \times Lc3 \times Pr/\nu \times \cos\beta$

So, $h_{p-c} = (Nu_L \times k) / L_c$ (3)

Where,

L_c = Cover-Plate spacing

The convective heat transfer coefficient (hw) at the top cover has generally define by j-factor and Reynolds number,

$L^* = (4\,A_c)/C_c$ (4)

$Re_{L^*} = (L^* \times V_\infty) / \nu$ (5)

$j = 0.86 \times (Re_{L^*})^{-1/2}$ (6)

Where,

A_c = Area of top cover

C_c = circumference of the top cover

So, $h_w = j \times \rho \times C_p \times V_\infty \times (Pr)^{-2/3}$ (7)

$U_t = (q_t/A_p) / (T_p - T_a)$ (8)

Assuming the temperature difference between the bottom surface temperature and ambient is zero, bottom heat transfer coefficient, Ub is

$U_b = k_i/\delta$ (9)

Where, δ is the insulation thickness at the bottom and k is the conductivity of the insulation.

For Side loss coefficient,

$U_s = (L+W)\,L_2\,k_i / (LW\,\delta_s)$ (10)

From above all heat loss, the useful heat is given by,

$q_u = A_p\,S - q_{loss}$ (11)

To simplify the problem collector efficiency factor F' is introduced which includes bond and fluid resistances. The heat gain per unit length of the tube is now defined in terms of the fluid temperature:

$Q_u = WF' [S - U_L (T_f - T_a)]$ (12)

Where the collector efficiency factor F' is

$$F' = \frac{1/U_L}{W\left[\frac{1}{U_L\left[D+(W-D)F\right]}+\frac{1}{C_b}+\frac{1}{\pi D_i h_f}\right]} \quad (13)$$

The collector efficiency factor is a constant for any collector design. However, U_l, C_p, h_f, and F are variable with temperature.

Collector heat removal factor F_R is also the most common method to define the actual energy gained in the collector with the heat gained as if the absorber temperature is equal to the inlet fluid temperature.

$$F_R = \frac{\dot{m}C_p}{A_c U_L}\left[1 - e^{\left(-A_c U_L F' / \dot{m}C_p\right)}\right] \quad (14)$$

The heat removal factor times the maximum possible useful energy gained gives the actual possible useful energy gained.

$Q_u = A_c F_R [S - U_L (T_i - T_a)]$ (15)

This equation applies for liquid heating flat plate collectors. The efficiency of a collector can be then defined:

$$\eta = \frac{Q_u}{I A_c} \quad (16)$$

Where, I is the solar radiation in W/m^2.

4.2 Heat Transfer Limits of Heat Pipe

Based on the design dimensions of collectors and heat pipe, the critical design criteria is considered to check the limits of the heat pipe heat transport. The collector system is supposed to work in all conditions throughout the year and the maximum radiation level is considered as a critical value. A value of 1000 W/m^2 solar radiation is used for the design. With this radiation level, the maximum heat that the heat pipe transfers is 0.195 kW. Table 2 shows a summary of the heat transfer limits for the heat pipe. The values obtained are well above the critical design value and the heat pipe is safe for operation. Capillary limit can be ignored for this specific due to the gravity assistance for the return of the condensate. However, the presence of wick capillarity in the heat pipe can enhance the uniform distribution of condensate in the evaporator to avoid the drawback of thermosyphon in solar collectors.

Table 2 Heat Pipe Heat Transfer Limits [2]

Type	Formula	Value
Sonic Limit	$Q_{max} = A_v \rho_v H \left[\frac{\gamma_v R_v T_v}{2(\gamma_v + 1)}\right]^{0.5}$	≈3.39 kW
Entrainment Limit	$Q_{max} = A_v H \left[\frac{\sigma_l \rho_v}{2 r_b}\right]^{0.5}$	≈0.195 kW
Boiling Limit	$Q_{max} = \frac{2\pi l_t k_e T_v}{H \rho_v \ln\left(\frac{r_i}{r_v}\right)}\left[\frac{2\sigma_1}{r_o} - P_o\right]$	≈1.60 kW

V. EXPERIMNTAL SETUP AND EXPERIMENTS

5.1 Collector Systems

Two set of collectors have been fabricated. One with natural convection flat plate solar collector and second is heat pipe solar collector. Following Figure 5 Shows the Solar water heater with 40 litter water tank capacity. The dimensions of both collectors are based on 6 hours required for heating water up to 55^0C. Table 3 shows material used for manufacturing of solar water heater and Table 4 shows the dimension of both solar water heater.

Table 3 Material used for manufacturing

Material Used	Function
Glass Wool	Insulation
Copper	Heat pipe, Liquid pipe
Float Glass	Cover
GI	Collector Box
GI	Water Tank
Copper wire	Wick structure
Aluminium Sheet	Absorber Plate
Copper tube	Water circulation

Table 4 Dimensions of solar water heater

Types of collector	Absorber Dimensions m x m	Storage capacity ltr	Heat pipe Cond. length m	Heat pipe Evap. length m
Heat pipe Collector	1.35 x 0.6	40	0.25	1.35
Convent-ional Collector	1.35 x 0.6	40	-	-

Figure 5 solar water heater with heat pipe

5.2 Measurements

The performances of both types of solar water heater have been tested on 17th April 2015. The solar collector unit of both systems was sloped at 30^0 toward the south. K-type thermocouples were used to measure the temperature of cover, plate and water storage tank. The ambient temperature has been measured using mercury thermometer. A digital display data logger was used for reading the temperature scale.

Solar radiation was measured using Pyranometer. The experiments was carried outs from 8 am morning to 5 pm of evening with time interval of one hour.

VI. RESULTS AND DISCUSSION

Figure 6 show the hourly variation of the ambient and the water inlet temperature for solar water heater with heat pipe (HPSWH) and conventional solar water heater (CSWH). The ambient and water inlet temperatures continued to change throughout the day and they depend on environment data (solar radiation, wind speed, sky temperature etc.). The measurements were done for clear days, hence, a variation of the ambient temperature can was observed.

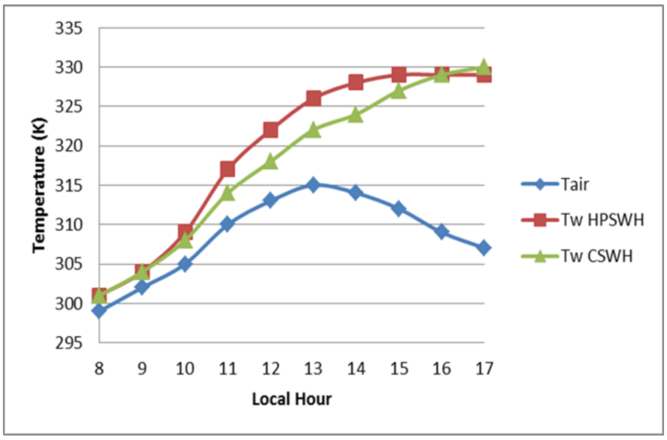

Figure 6 Variation of temperature with time

Fig. 7 shows the solar radiation intensity as function of time. In the model only the mean experimental inlet water temperature and the mean experimental mass flow rate are introduced for the calculation. Whereas the other parameters are those of the meteorological data from this location like the global and diffuse radiation, the hourly ambient temperature and the hourly wind velocity.

Results show that the maximum water temperature obtained is a function of solar insulation and the ambient air temperature. Therefore, the maximum water temperature occurred after the peak solar insulation. During the test, a maximum water temperature of 56^0C and 57^0C was obtained for heat pipe solar water heater and conventional solar water heater respectively. Its maximum ambient temperature was 42^0C and the solar radiation is maximal reach 992 W/m^2.

Figure 8 and 9 gives the instantaneous efficiency curve for heat pipe solar water heater and conventional solar water heater respectively.

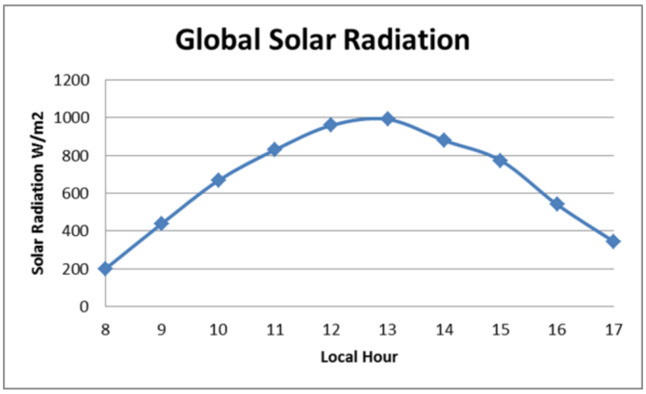

Figure 7 Variation of global solar radiation with time

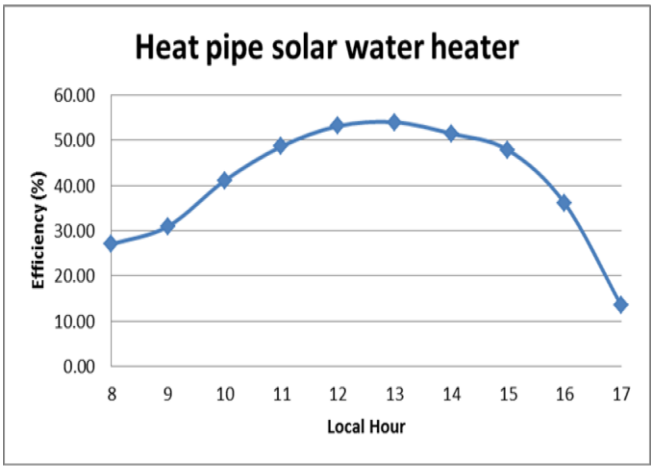

Figure 8 Efficiency curve for heat pipe solar water heater

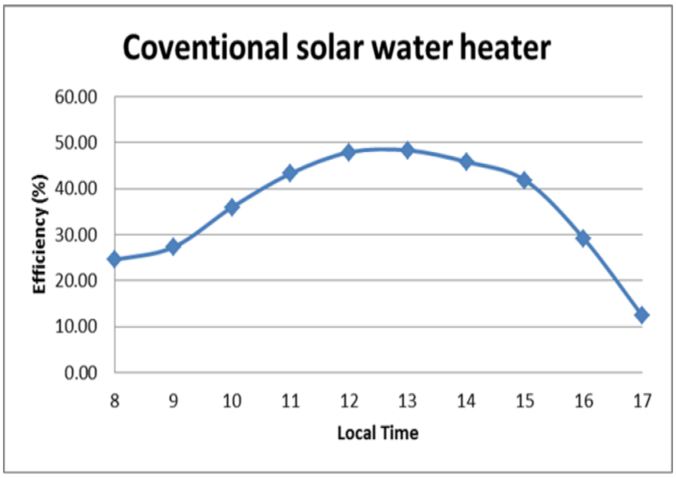

Figure 9 Efficiency curve for Conventional solar water heater

From Above figure 8 and 9 it is clear that system efficiency for solar water heater with heat pipe is nearly 6% higher than the efficiency of conventional solar water heater and maximum efficiency 54.06 % have been achieved. Here heat removal rate for heat pipe higher because maximum heat transfer capacity of heat pipe is 585 W and it is reason for higher efficiency.

VII. CONCLUSION

Compared with conventional flat plate solar water heating systems, the heat pipe flat plate solar collector has the advantage of operating as thermal diode (when the collector temperature is less than the storage water temperature, the heat energy will not be lost from storage tank); good resistance against corrosion; elimination of contamination of portable water, easy freeze protection; and no pumping or control system requirements (means that the system will not require an electrical connection). Heat removal rate of heat pipe is higher than heat removal rate of fluid flowing through pipe, so water heating rate is also high for heat pipe solar water heater. From this study it is concluded that use of heat pipe is advantageous for solar water heating application.

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