INTERCONNECTED HEAT PIPE SOLAR COLLECTOR

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Abstract This paper deals with the performance of a heat pipe solar collector. The solar collector consisted of an interconnected heat pipe so as to reduce the production cost by using an interconnected heat pipe because all the heat pipes can be evacuated, sealed and tested at once. Performance of a prototype of the heat pipe solar collector was experimentally examined, and the results were compared with those obtained through theoretical analysis. The results shown in this paper seem feasible.

Keywords Heat Pipe, Heat Exchanger, Solar Collector, Efficiency

چکیده این مقاله به‌کارآیی کلکتور خورشیدی با لوله‌های گرمایی پرداخته است. این لوله‌های گرمایی با یک لوله‌به یکدیگر متصل شده‌اند، که در این حالت سیستم به صورت یکپارچه خلأ و تست می‌شود. این عمل باعث کاهش هزینهٔ تولید نسبت‌به سایر کلکتورهای خورشیدی با لوله های گرمایی است. عملکرد کلکتور، آزمایش‌شده و نتایج به‌دست‌آمده باآنالیز تئوری مقایسه شده است؛ به‌صورتی که کاملاً" رضایت‌بخش است.

1. INTRODUCTION

The heat pipe is a device of very high thermal conductance; that is, it will transport thermal energy without an appreciable drop in temperature [1,2]. The heat pipe is suitable for a wide range of applications including solar collector. In a heat pipe, the process is evaporation-Condensation-convection. Thus, solar collectors with heat pipes have a lower thermal mass, resulting in a reduction of start-up time [3].

Thermal diode is important in designing of solar collectors, where heat is transferred only from the evaporator to the condenser, but never in the reverse direction. This feature can cut off the heat loss when the absorber temperature is lower than that of the liquid in the heat exchanger [4-7].

Several studies on heat pipe solar collectors are reported in the literature. Riffat, et al [3] studied developing a theoretical model to investigate thermal performance of a thin membrane heat-pipe solar collector. In their work thin membrane heat-pipe solar collector was designed and constructed to allow heat from solar radiation to be collected at a relatively high efficiency while keeping the capital cost low. Azad [8] investigated the heat pipe solar collector theoretically and experimentally, and the optimum ratio of heated length-cooled length of the heat pipe. Hull [9] investigated heat transfer factors and thermal efficiency for heat pipe absorber array connected to a common manifold and predicted that array with less than ten heat pipes have significantly less efficiency than a conventional flow-through collector. Hussein [10] investigated the different design parameters of the natural circulation two phase closed thermosyphon flat plate solar water heaters using the verified expanded model. Chun, et al [11] presented an experimental setup, where five individual modules of only one heat pipe coupled with a thermal reservoir were tested. Different working fluids (water, methanol, acetone and ethanol), heat pipes with or without wick, different thermal reservoir volumes and different absorber surface treatment were compared. In the present work the results were obtained with a compact solar heating system that is being designed to provide solar water heating for low income families. Radhwan, et al [12] investigated experimentally the thermal performances of two R11 charged integrated solar water heaters using forced and natural circulation water flows. The results showed that the inclination of the condenser integrated within the collector frame had remarkable effect on the natural circulation of the water flow system, while it had no significant effect on the forced circulation flow system. Soin, et al [13,14] investigated the thermal performance of a thermosyphon collector containing boiling acetone and petroleum ether, and presented the effect of insolation and the liquid level on the collector performance. Fanney, et al [15] conducted experiments to determine the effect of irradiance level on the thermal performance of the refrigerant charged domestic solar hot water system. The experiments concluded that the irradiance level was negligible for the levels considered. Akyurt [16] designed and manufactured numerous heat pipes. Each heat pipe was incorporated into a prototype solar water heater. An extensive testing program lasting for more than a year revealed that the heat pipe performs satisfactorily as heat transfer elements in solar water heaters. Rittidech, et al [17]. The interconnected heat pipe solar collector described in this study relies on the natural forces of gravity and capillary action and does not require an external power source.

From the current literature, it is noticed that all of the previous studies of wickless heat pipe flat plate solar collectors were limited to solar collector with independent heat pipes.

Therefore in the present study interconnected heat pipe solar collector was designed, constructed and tested under real conditions with the system installed outdoors and compared to theoretical results. The main aim of this investigation was to reduce the production cost by using an interconnected heat pipe solar collector because all of the pipes can be evacuated, sealed and tested at once.

2. SOLAR COLLECTOR CONSTRUCTION

A prototype interconnected heat-pipe solar collector which was designed and constructed to collect and distribute heat by means of vaporization and condensation of a heat transfer fluid is shown in Figure 1. It comprised mainly six copper heat pipes with outside diameter of 12.7 mm and an evaporator length of 1900 mm while the wick consisted of two layers of 100-mesh stainless steel screen fitted to the evaporator section. The evaporator section of the heat pipes are interconnected by a 12.7 mm copper tube connected at the lower end of the heat pipes so as to distribute the working fluid uniformly in the heat pipes that avoid dry-out problem that might be experienced in traditional heat pipe solar collectors. The upper ends of the heat pipes are brazed to the condenser to avoid any seal leakage. The condenser is a shell and tube heat exchanger, is built of four round tubes (12.7 mm. diameter) in a copper cylindrical shell (50 mm. diameter) with tube axis parallel to that of the shell. One fluid flows inside the tubes, the other flows across and along the tubes.

The vapor leaving the heat pipes enters the shell and condenses on the outside wall of the tubes. Condensation rate is higher in the entrance region of shell where the transfer fluid is colder. However, this rate decreases with the gradual increase of transfer fluid temperature in the tubes. The condensate runs back through the shell and returns to the heat pipes under the action of gravity. The flow of condensate in the first heat pipe is more than the other heat pipes. However, since the heat pipes are interconnected, finally the liquid level becomes the same in entire array of heat pipes. Both the shell and the heat pipes are sloped to aid the liquid return. When the striking solar radiation can no longer initiate, the evaporation of the working fluid, the heat pipes cool down rapidly and no heat can flow from condenser to evaporator. After the system had been vacuumed to 10^{-4} torr, achieved with a rotary vacuum pump and a diffusion pump, a specified amount of working fluid (200 ml. of ethanol) was charged into the system through the filling tube. The system was then prepared for crimping and final sealing. Figure 2 shows heat pipes array with the shell and tubes heat exchanger and Figure 3 shows the system filling rig. Ethanol filled heat pipes are more efficient and less susceptible to freezing. The heat pipes were mechanically bonded to the extruded aluminum absorber plates specially designed for this purpose at 160 mm pitch. The absorber plate is anodized matt black to enhance its ability to absorb heat. The absorber plate and heat exchanger were housed in an aluminum framework with a 0.5 mm thick aluminum sheet bottom. The panel rests on a backing insulation layer of 50 mm thick glass wool while the condenser section was insulated with Aeroflex sheet insulation. Ordinary glass window was chosen as the upper glazing for the collector.

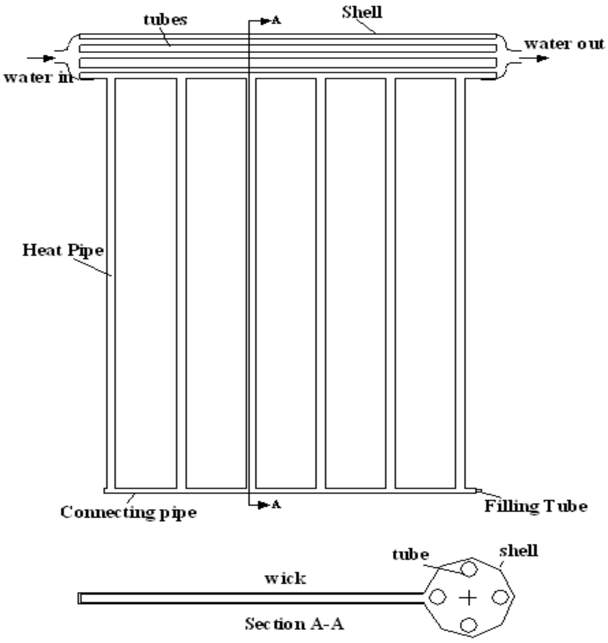

Figure 1. Interconnected heat pipes solar collector.

Figure 3. Heat pipe filing rig.

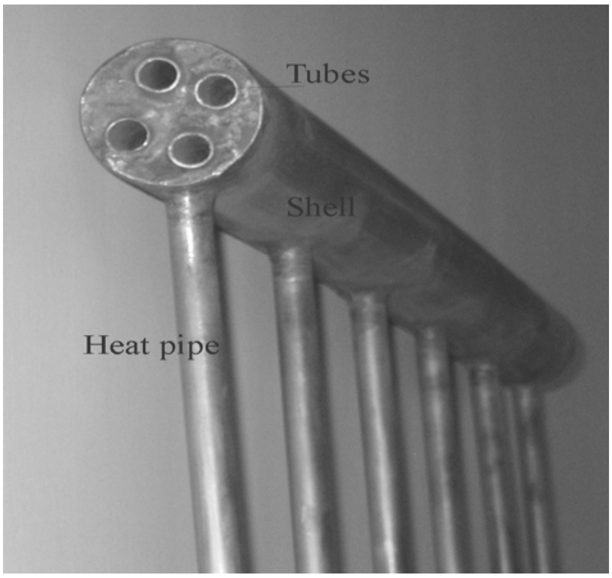

Figure 2. Interconnected heat pipes with heat exchanger.

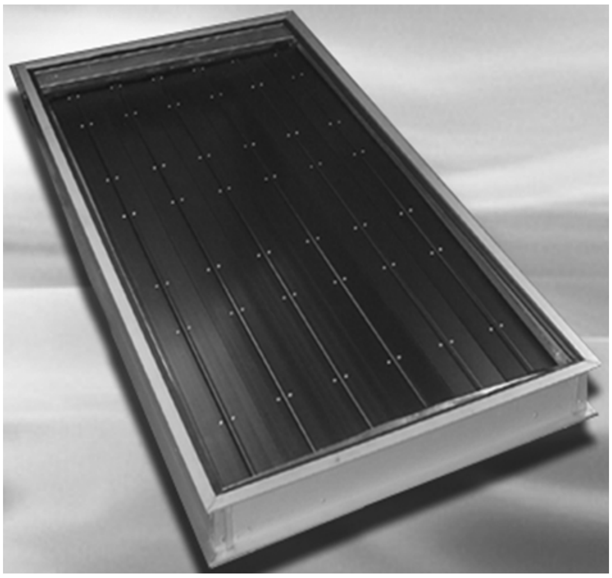

Figure 4. Solar collector.

The air gap between glass cover and the absorber plate was 40 mm. The glass was secured on the top of the frame by rubber gasket and aluminum angles, which permitted thermal expansion but prevented the entrance of dust and rain, Figure 4 shows the solar collector.

Because no evaporation or condensation above the phase-change temperature is possible, the heat pipe offers inherent protection from freezing and overheating. This self-limiting temperature control is a unique feature of the heat pipe solar collector.

Compared to conventional heat pipe flat plate solar collector, the present system has several advantages. These advantages may be outlined as follows:

- Ease of production: In conventional heat pipe solar collector each heat pipe must be vacuumed, filled with working fluid, and tested individually. However, for the present design all of the construction steps happen simultaneously.
- Lower production cost: Production cost would be reduced by using an interconnected heat-pipe, since all pipes can be vacuumed, filled, sealed, and tested at once.
- Uniform distribution of working fluid in heat pipes.
- Thermal flux transformer: thermal flux transformer, i.e. the ratio of the absorber area to the condensing in this design is more than the conventional heat pipe solar collector.

$$q_{hm} = I\eta \frac{A_a}{A_{hm}} \quad (1)$$

Where

$A_{hm} =$ Condenser area $= \pi d L_c n_c$

$A_a =$ Absorber area $= L_e \times W$

$q_{hm} =$ Condenser heat flux

$n_c =$ Number of condenser tubes = 4

$L_c, L_e =$ Condenser tube and evaporator length

In this design $L_e = (L_{collector} - D_{shell}) > (L_{conventional,heatpipe} - L_c)$ since $D_{shell} < L_c$ as a result the absorber area is larger and therefore it absorbs more thermal energy.

3. THERMAL ANALYSIS

The following assumptions have been made to facilitate the analysis. The assumptions limit the accuracy with which collector performance can be analytically predicted.

- For ease of analysis, the system is considered to be in steady state conditions
- The temperature gradient in the longitudinal direction of the collector can be neglected.
- The overall heat loss coefficient between the collector and the ambient is assumed to be constant.
- The heat loss from the heat exchanger and the ambient is negligible.

For a single glazed flat-plate collector the useful energy gained by the absorber, is the difference between solar energy absorbed and the heat loss to the ambient over the length of the absorber. Under steady-state conditions, the useful heat delivered by a solar collector is equal to the energy absorbed by the heat transfer fluid minus the direct or indirect heat losses from the surface to the surroundings. The useful energy collected from a collector may be modeled according to well known Hottel, et al [18] equation:

$$Q_u = A_a F'[I(\tau\alpha)_e - U_L(T_{hp} - T_a)] \quad (2)$$

The overall heat loss coefficient is calculated by:

$$U_L = U_t + U_b + U_e \quad (3)$$

Where

U_t, U_b and U_e represent top loss coefficient, back loss coefficient and edge loss coefficient, respectively. The top loss coefficient from absorber plate to ambient can be written as follows [19].

$$U_t = \left(\frac{1}{h_{p-c} + h_{r,p-c}} + \frac{1}{h_w + h_{r,c-a}} \right)^{-1} \quad (4)$$

The convective heat transfer coefficient between the absorber plate and the cover, h_{p-c}, may be evaluated from the following relation:

$$h_{p-c} = \frac{h.NU_{p-c}}{\delta_a} \quad (5)$$

Where, the free convective Nusselt number correlation is that for an enclosure with two parallel flat plates inclined at an angle ϕ, given by Hollands, et al [20].

$$Nu_{p-c} = 1 + 1.44\left[1 - \frac{1708}{Ra\cos\phi}\right] \times \left[1 - \frac{1708(\sin 1.8)^{1.6}}{Ra\cos\phi}\right] \times \left[\left(\frac{Ra\cos\phi}{5830}\right)^{1/3} - 1\right] \quad (6)$$

Where $Ra = g\beta\rho^2\delta_a{}^3\Delta TC_{pa}/\mu K$ and ΔT is the temperature difference between the absorber plate and glass cover and β is the coefficient of thermal expansion for air.

The radiation heat transfer coefficient between the absorber plate and the cover, $h_{r,p-c}$, is given by:

$$h_{r.p-c} = \sigma\left(T_{hp}{}^2 + T_c{}^2\right)\left(T_{hp} + T_c\right)/\left(1/\varepsilon_p + 1/\varepsilon_c - 1\right) \quad (7)$$

The radiation coefficient from cover to ambient is given by:

$$h_{r,c-a} = \sigma_c\left(T_c{}^2 + T_a^2\right)\left(T_c + T_a\right) \quad (8)$$

It is customary to calculate the external convective heat transfer coefficient h_w from an expression which linearly relates the coefficient to the wind speed [19].

$$h_w = 5.7 + 3.8v \quad (9)$$

for $0 < v < 10$ m/s

Rear heat loss coefficient may be evaluated from:

$$U_b = \left(\Sigma\frac{\delta_i}{k_i} + h_b^{-1}\right)^{-1} \quad (10)$$

Tabor [21] recommended that h_b = 12.5 to 25 Wm^2K^{-1}.

The edge heat loss coefficient is

$$U_e = \frac{A_e}{A}\left(\Sigma\frac{\delta_i}{k_i} + h_e^{-1}\right)^{-1} \quad (11)$$

Willier [22] recommends that $h_e = 0.5\ Wm^{-2}K^{-1}$.

The variation of fluid temperature in a heat exchanger with constant temperature gradient is given by:

$$\int_{\Delta T_i}^{\Delta T_o} \frac{d(\Delta T)}{\Delta T} = -\frac{A_{hm}}{\dot{m}C_p}\int_0^L U_{hm}dx \quad (12)$$

$$\frac{T_{hp} - T_{o,theory}}{T_{hp} - T_i} = \exp\left(-\frac{A_{hm}U_{hm}}{\dot{m}C_p}\right) \quad (13)$$

$$T_{o'theory} = T_i\exp\left(-ntu_{hm}\right) + T_{hp}[1 - \exp\left(-ntu_{hm}\right)] \quad (14)$$

Where

$$ntu_{hm} = \frac{\left(U_{hm}A_{hm}\right)}{\left(\dot{m}C_p\right)} \quad (15)$$

The heat transferred to the condenser is then

$$Q_{hm} = \dot{m}C_p\left(T_{hp} - T_i\right)\left[1 - \exp\left(-ntu_{hm}\right)\right] \quad (16)$$

By equating Equations 1 and 16, T_{hp} [6] can be calculated as follows:

$$T_{hp} = \frac{\left(I\alpha_o/U_L\right) + T_a + \left[1 - \exp\left(-ntu_{hm}\right)\right]T_i/ntu_{hp}}{1 + \left[1 - \exp\left(-ntu_{hm}\right)\right]/ntu_{hp}} \quad (17)$$

Where

$$ntu_{hp} = \frac{\left(F'A_cU_L\right)}{\left(\dot{m}C_p\right)} \quad (18)$$

The overall effectiveness in term of temperature is given by

$$E = \frac{T_o - T_i}{T_{hp} - T_i} \quad (19)$$

The overall heat transfer coefficient in manifold can be calculated as:

$$U_{hm} = \frac{h_{co} h_{ci}}{h_{co} + h_{ci}} \tag{20}$$

h_{ci}, the heat transfer coefficient for laminar flow in tube can be obtained by Sieder, et al [23] relation:

$$h_{ci} = \frac{k}{d} Nu_d \tag{21}$$

$$Nu_d = 1.86\left(Re_d\, Pr\right)^{1/3}\left(\frac{d}{L}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{22}$$

In this formula the average heat-transfer coefficient is based on arithmetic average of the inlet and outlet temperature differences.

A traditional expression for calculation of heat transfer in fully developed turbulent flow in smooth tubes is that recommended by Dittus and Boelter through reference [24]:

$$Nu_d = 0.023\, Re_d^{0.8}\, Pr^{0.4} \tag{23}$$

For laminar film condensation on horizontal tubes [24]:

$$\bar{h}_{co} = 0.728\left[1 + 0.2\frac{C_p\left(T_{sv} - T_w\right)}{h_{fg}}(N-1)\right]\left[\frac{\rho_f\left(\rho_f - \rho_v\right) g h'_{fg} k_f^3}{N \mu_f d\left(T_{sv} - T_w\right)}\right]^{1/4} \tag{24}$$

Where:

N = number of horizontal tubes

$h'_{fg} = h_{fg} + \frac{3}{8} C_p\left(T_{sv} - T_w\right)$

d = tube diameter

T_{sv} = temperature of saturated vapor

T_w = wall surface temperature

Theoretical efficiency is defined as:

$$\eta = \frac{Q_u}{I A_a} \tag{25}$$

$$Q_u = Q_{hm} = \dot{m} C_p\left(T_{hp} - T_i\right)\left[1 - \exp\left(-ntu_{hm}\right)\right] \tag{26}$$

$$\eta_{theory} = \frac{\dot{m} C_p\left(T_{hp} - T_i\right)\left[1 - \exp\left(-ntu_{hm}\right)\right]}{I A_a} \tag{27}$$

4. EXPERIMENT

The experiments were performed at the Solar Energy Lab. IROST, Tehran (latitude 35.7˚N; longitude 52.3˚E, and altitude 1190 m). The solar collector was installed and tested under outdoor field conditions. The collector was mounted on the stand, oriented N-S, and tilted 35.7˚N towards the south. A closed loop configuration was employed for testing the collector as shown in Figure 5.

Physical quantities measured are: cooling water temperatures at the inlet and outlet of the heat exchanger, ambient temperature, circulating water flow rate, and the incident solar irradiance on the plane of the solar collector.

The experiments are performed for global solar radiation between 120 and 1000 W/m^2 on a 35.7˚ tilted collector surface. This was measured with Kipp and Zonnen pyranometer CM21. The unit was mounted alongside the absorber panel and inclined at the same angle. The maximum instrument error was 2 per cent. The output from pyranometer was recorded by Kipp and Zonnen flatbed single channel model BD 111. Chromel-Alumel thermocouples (Type K) thermocouples were used to measure the inlet and outlet water temperature of the heat exchanger and ambient temperature, with an accuracy of ± 0.1˚C for temperature measurements. The thermocouple measuring outdoor temperature was placed in a shaded position immediately in the back of the collector. A 12-point Leeds and Northrop Speedomax W chart recorder with compensated reference junction prints out temperatures in degree C. The water flow rate was set by a control manual valve in the cold line to provide an almost constant flow rate of 1.8 lit/min.

Water circulated from an insulated 200 liter tank to the inlet of the heat exchanger passing throughout the condenser tubes and exiting from the other end back to the tank. The tank was elevated to anticipate a constant head.

The experiment was carried out at a different water inlet heated to the desired temperature as it passed through an electric heater controlled by a variable-output AC voltage-transformer. The circulating fluid flow rate was regulated by means of a needle valve and was measured with a flow meter. The fluid was circulated by a centrifugal pump. The water passed through the circulating pump, the collector, and outer jacket of the storage tank in order to exchange the heat with the water inside the 200 liter tank, and back to the water heater. The water flow rate was almost constant flow rate of 1.8 lit/min.

During each experiment, the mass flow rate of cooling water was kept constant, and the solar radiation intensity (I), ambient air temperature (T_a), inlet cooling water temperature (T_i) and outlet cooling water temperature (T_o) of the collector were recorded.

The energy absorbed by solar collector was determined from the measurements by:

$$Q_u = \dot{V}\rho C_p\left(T_{o,exp.} - T_i\right) \tag{28}$$

The experimental efficiency of the collector is calculated as the product of the heat capacity rate per unit collector area and water temperature rise in the heat exchanger divided by the solar irradiance. The instantaneous efficiency is determined by:

$$\eta_{exp.} = \frac{Q_u}{I.A_a} \tag{29}$$

5. RESULTS AND DISCUSSION

To get results for different inlet temperatures an electric heater with variable power was used. Figure 5 represents the experimental facility for solar collector testing. The inlet and outlet water temperature was measured with calibrated type K thermocouples.

The experimental results are represented by Figure 6. The variations of solar radiation (I), the ambient temperature (T_a) and the water inlet (T_i) were measured experimentally. T_a, T_i, $T_{o,theory}$, T_{oexp}, and T_h are shown on the left scale of this figure, and solar insolation data, I, is shown on the right scale. The ambient and water inlet temperatures continued to change throughout the day. The water outlet temperature increases from early morning time until sometime in the afternoon (about 13:00 local time).

The decline of water outlet and heat pipe temperatures is the result of declining solar insolation. The measured and calculated water outlet temperatures are compared. It can be observed that there is a good agreement between the measured and the calculated temperatures. The comparison of the collector efficiency at steady-state conditions as obtained by outdoor measurements and by theoretical model is presented in Figure 7. The left scale represents the efficiency values and the right scale represents the solar insolation. The maximum difference between theoretical and experimental efficiencies occurs at 17:00 PM. As it could be observed from figure, at 17.00 PM, the calculated theoretical efficiency is 58.6%, while the experimental efficiency is 58.1% (difference of 0.5%).

6. CONCLUSIONS

Interconnected heat pipe solar collector was designed and constructed. This design would reduce the production cost of the heat pipe panel and provides a better working condition for the system to act as thermal flux transformer compared to conventional heat pipe solar collectors. An analytical method was developed to examine the heat transfer occurring in the collector. The model was used to determine the collector efficiency, heat pipe temperature, water outlet temperature, and useful heat absorbed by cooling water.

The validation of the model developed in this study has been confirmed by comparison of the results of the theoretical study with the results of the experimental study developed results in literature [8]. The comparison showed the results were in good agreement. To enhance the performance of collector, the following points should be considered:

- Increasing collector efficiency can be achieved by increasing the number of heat pipes in the collector with only minimal increase in cost.

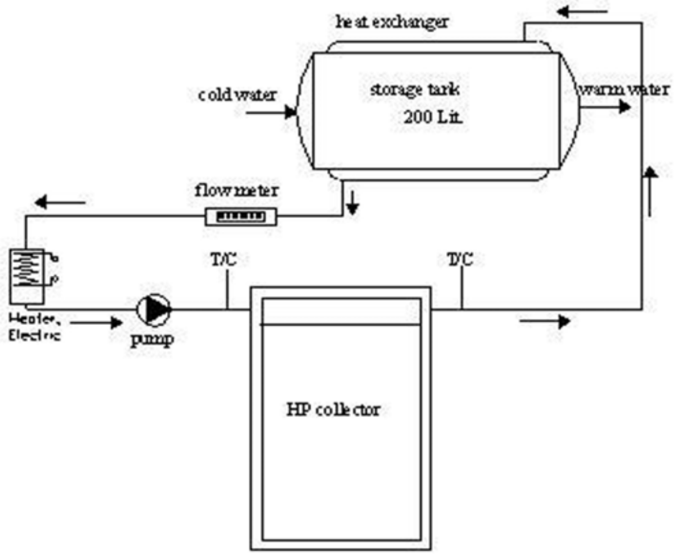

Figure 5. Solar collector test rig.

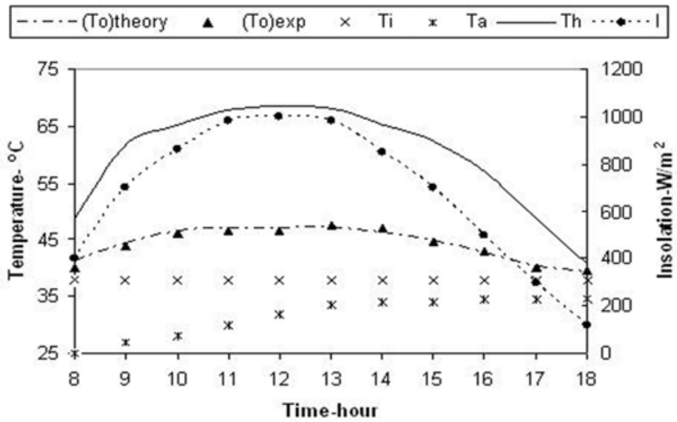

Figure 6. Variation of T_a, T_i, T_{hp}, $T_{o,exp.}$, $T_{o,theory}$ and insolation with time.

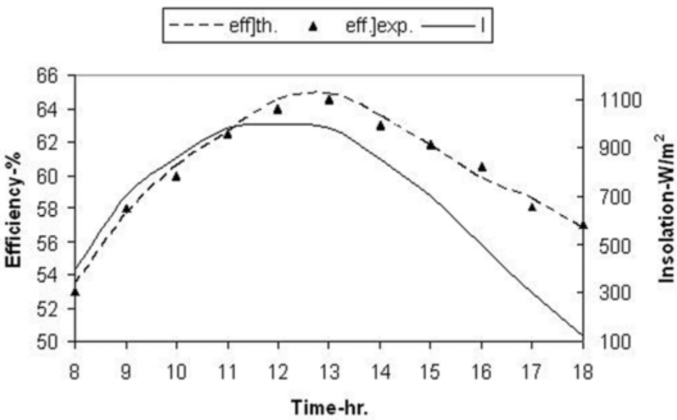

Figure 7. Comparison between experimental and theoretical efficiencies.

- Using a selective surface instead of anodized matt black paint or the use of alternative materials would improve collector performance.

7. NOMENCLATURE

A_a	Absorber area = $L_e \times W$, m^2
A_{hm}	Condenser area = $\pi d L_c n_c$, m^2
C_p	Heat capacity, J/kg/˚C
D	Diameter, m
F'	Collector efficiency factor
h_{p-c}	Convective heat transfer coefficient between the absorber plate and the cover, W/m^2˚C
$h_{r.p-c}$	The radiation coefficient from plate to cover, W/m^2˚C
h_w	Wind heat transfer coefficient, W/m^2˚C
I	Insolation, W/m^2
K	Thermal conductivity, W/m˚C
L_c, L_e	Condenser tube length and evaporator length, respectively, m
$\dot{m}$	Mass flow rate, kg/s
n_c	Number of condenser tubes = 4
NTU	Number of transfer unit = AU/C_{min}
Nu	Nusselt number = h.d/k
Q	Heat transfer rate, W
q_{hm}	Condenser heat flux, W/m^2
T	Temperature, ˚C
U_b	Back loss coefficient, W/m^2˚C
U_e	Edge loss coefficient, W/m^2˚C
U_L	The overall heat loss coefficient, W/m^2˚C
U_t	Top loss coefficient, W/m^2˚C
W	Width of absorber plate, m
$(\tau\alpha)$	Transmittance-absorptance product, dimensionless
h_{fg}	Latent heat, J/Kg

Subscripts

a	Ambient, Absorber
c	Condenser
e	Evaporator
hp	Heat Pipe
hm	Heat Pipe to Exchanger
i	Inlet, Inside
l	Liquid
o	Outlet
u	Useful

8. ACKNOWLEDGEMENT

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