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Short communication

Performance of a two-phase closed thermosyphon solar collector with a shell and tube heat exchanger

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Abstract

In the present study, a two-phase closed thermosyphon flat-plate solar collector with a shell and tube heat exchanger was investigated experimentally under the field conditions of Cairo, Egypt. The collector was designed, constructed, and tested at transient conditions to study its performance for different cooling water mass flow rates at different inlet cooling water temperatures. Also the effect of the number of the thermosyphon tubes on the performance of the collector was investigated. Under different climate conditions, the experimental results showed that the optimal mass flow rate is very close to the ASHRAE standard mass flow rate for testing conventional flat-plate solar collectors. Also, the experimental results indicated that the number of the thermosyphon tubes has a significant effect on the collector efficiency. The performance of the present collector with optimum number of thermosyphon tubes was compared with the performance of two-phase closed thermosyphon flat-plate solar collectors with tube in tube heat exchangers of previous investigators and a better performance for the present collector was obtained at high inlet water temperature.

Keywords: Two-phase closed thermosyphon; Flat-plate solar collector; Shell and tube heat exchanger

1. Introduction

A two-phase closed thermosyphon tube is a highly efficient device for heat transfer. It consists of an evacuated closed tube filled with a suitable amount of a working fluid. Heat is transferred by the processes of evaporation and condensation of the working fluid at the lower section of the

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Nomenclature

A	collector effective area, m^2
C_w	specific heat of water, J/kg K
F_R	collector heat removal factor, dimensionless factor
I	global solar radiation intensity on tilted surface of the collector, W/m^2
L_e	effective length of the collector, m
$\dot{m}_w$	cooling water mass flow rate, kg/s
n	number of pipes of the collector, dimensionless
P	pitch of the collector pipes, m
q_u	useful output energy, W
T_a	ambient temperature, °C
T_{wi}	inlet cooling water temperature, °C
T_{wo}	outlet cooling water temperature, °C
U_L	overall heat transfer coefficient, W/m^2 K
α_p	absorptivity of the absorber plate, dimensionless
η	collector efficiency, dimensionless
τ_g	transmittance of the glass cover, dimensionless

tube (evaporator section) and the upper section of the tube (condenser section), respectively [1]. Therefore, heat is transferred in a latent form (high heat rates) over considerable distances and extremely small temperature drop between the evaporator section (heated region) and the condenser section (cooled region) of the thermosyphon tube with a small degradation of energy. This make the thermosyphon tubes more recommended to be used in solar system, mainly in solar collectors [2,3]. Moreover, using the two-phase closed thermosyphon tube in solar collectors does not need moving parts or external pumping power, freezing of the working fluid inside the thermosyphon tube is not destructive, and the unit acts as a thermal diode preventing the reverse circulation problem in conventional solar collectors [4,5].

The previous work on two-phase closed thermosyphon flat-plate solar collectors was directed towards studying their performance theoretically and experimentally or comparing them with conventional solar collectors [4–10]. Most of these studies did not take in account the way by which heat is transferred from the thermosyphon tube to the cooling water in the heat exchanger. The common way used by most of the previous studies [3,6,10,11] was circulating the water from the heater's tank to a header through which the water was distributed to flow through a separate annulus around the condenser section of each thermosyphon tube of the collector. Then the water outlet from each annulus was collected in an outlet header to flow again to the tank. Terpstra and Van Veen [2] described a two-phase closed thermosyphon tube. At the collector outlet, vapours are brought together with liquid and they are condensed in an auxiliary tube in tube heat exchanger in the upper part of the system. Radhwan et al. [12] presented an experimental investigation for two R-11 charged integrated solar water heaters for forced and natural circulation water flows. The results showed that the inclination of the condenser integrated with the collector frame had a remarkable effect for natural circulation water flow, while it had no significant effect for forced circulation flow. Mathioulakis and Belessiotis [13] conducted an experimental and theoretical investigation to study the performance of a new ethanol thermosyphon solar water heater. In this system the condenser section of the thermosyphon tube was placed inside the tank of the water heater. Foster et al. [6] showed theoretically and experimentally that the major resistance to heat flow from the collector to the water to be heated was in the convective film of the water in the annulus of the condenser section of the thermosyphon tube. They recommended that a small shell and tubes heat exchanger might be considered.

As shown above most of the previous studies had a common feature that the condenser section of the thermosyphon tubes was a separate tube in tube heat exchangers where, the water enters and exits these heat exchangers through two common headers. Other previous studies showed that the major resistance to heat flow in the two-phase closed thermosyphon solar collector was in the condenser section of the collector and they recommended more studies of using different types of heat exchangers at the condenser section. Therefore in the present study a two-phase closed thermosyphon flat-plate solar collector with a shell and tube heat exchanger was designed, constructed and tested at transient conditions. The shell and tube heat exchanger was integrated with the collector frame at the top. The cooling water flows in the shell in cross flow around the condenser section of the thermosyphon tubes which were arranged in one row as shown in Figs. 1 and 2. The collector in this form is simplest in manufacturing and has less material cost. The transient performance of the present collector was investigated experimentally at different water flow rates. Also, the effect of the number of the thermosyphon tubes of the collector on the water temperature rise and the collector efficiency was investigated.

2. Experimental set-up and procedure

A schematic diagram of the test rig and cross-sectional views of the two-phase closed thermosyphon flat-plate collector under consideration are shown in Figs. 1 and 2, respectively. Heat

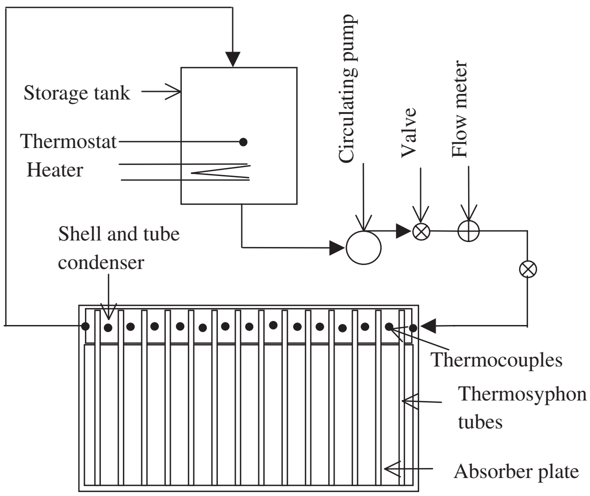

Fig. 1. Schematic diagram of the test rig components.

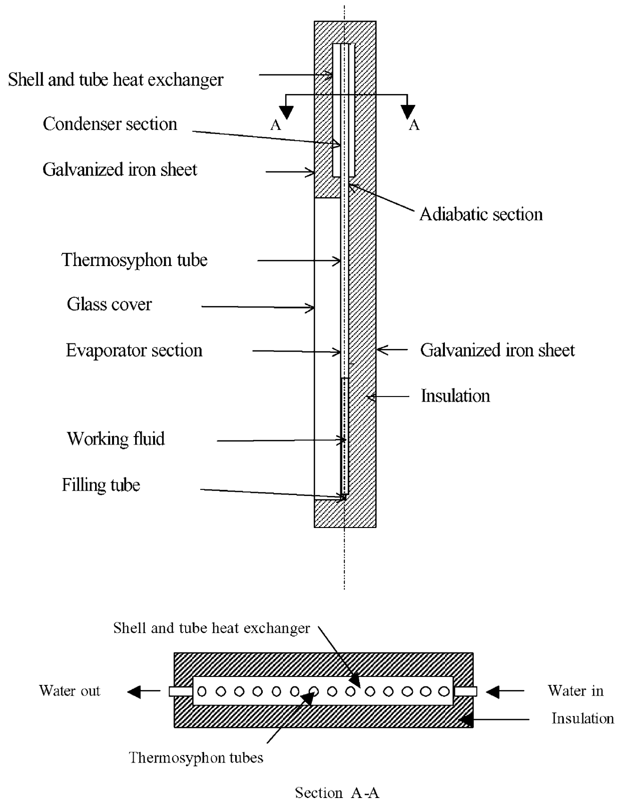

Fig. 2. Cross sectional views of the collector.

transfer from the collector to the cooling water is performed by the process of evaporation of the working fluid of the heat pipe, which was water, in the evaporator section of the thermosyphon tubes and condensation of the vapour by realizing its latent heat to the cooling water through the shell and tube cross flow heat exchanger. The evaporator of the thermosyphon tubes consists of 14 tubes in which absorber fins were welded together. The condenser sections of the thermosyphon tubes were placed inside a shell and tube cross flow heat exchanger where the cooling water flow in cross direction on the condenser sections of the thermosyphon tubes as shown in Fig. 1. The basic characteristics and specifications of the different components of the collector are summarized in Table 1. The two-phase closed thermosyphon flat-plate solar collector was mounted on a tiltable stand of a flat-plate solar collector closed loop test rig shown in Fig. 1.

Physical quantities measured are: cooling water temperatures in several axial sections along the shell and tube heat exchanger, cooling water temperatures at the inlet and outlet of the collector, cooling water flow rate, incidence solar irradiance and ambient air temperature. For temperature measurements, 15 type K thermocouples are placed at several axial locations along the shell and tube heat exchanger as shown in Fig. 1. Another K-type thermocouple probe was used to measure the ambient air temperature. A rotameter was used to measure the cooling water flow rate and a weather station connected to a data logger was used for the measurements of the solar irradiance incident on the tilted surface of the collector.

Table 1
Specifications of the different components of the solar collector

Gross dimensions		
Length 1.0 m	Width 1.96 m	Depth 0.1 m
Transparent cover		
Material: white glass	Dimensions $0.76\times1.9\times0.004$ m Air gap 0.05 m	Collector effective area $(A) = 1.44$ m^2
Absorber plate		
Material: copper Coating: velvet black	Length (L_e) 0.75 m Thickness 0.001 m	Pitch distance 0.135 m
Thermosyphon tubes		
Material: copper Working fluid: water	Outer diameter 0.0127 m Inner diameter 0.0117 m	Length 0.92 m No. of pipes 14
Heat Exchanger		
Type: Shell and tube Material: galvanized iron	Cross section 0.1×0.03 m Length 1.9 m	Flow direction: cross
Insulation		
Material: glass wool	Thickness 0.05 m	Position: Back & sides
Casing material		
Frame: aluminum	Back cover: galvanized iron	

The two-phase closed thermosyphon tube solar water collector was installed and tested under the actual field conditions of Cairo, Egypt. The experiments on the collector have been conducted during May and June 2003. The experiments were carried out at different cooling water flow rates 0.0125, 0.0292, 0.0458 and 0.0625 kg/s. For each water flow rate, two sets of experiments were carried out: (1) experiments along the standard local time of the day from 8:00 a.m. to 6:00 p.m., and (2) experiments along several days during the quasi-steady state period around the solar noon. At each experimental run, the efficiency of the collector was calculated from

$$\eta = \frac{\dot{m}_w C_w (T_{wo} - T_{wi})}{IA} \tag{1}$$

The uncertainties in the various variables used in the determination of the collector efficiency are: 0.1 °C for any temperature measurements, 2% for the water flow rate, 2% for solar irradiance and 0.001 m for any distance measurements. Following the procedure of Holman and Gajda [14], the uncertainty of the collector efficiency was estimated to be within 5%.

3. Results and discussion

For an understanding of the performance evaluation of the collector, Fig. 3 gives the instantaneous variation of the ambient temperature, the inlet and outlet cooling water temperatures and the experimental measured global solar radiation intensity along the standard local time of the day for four different cooling water flow rates. As shown in the figure, as expected, for the same

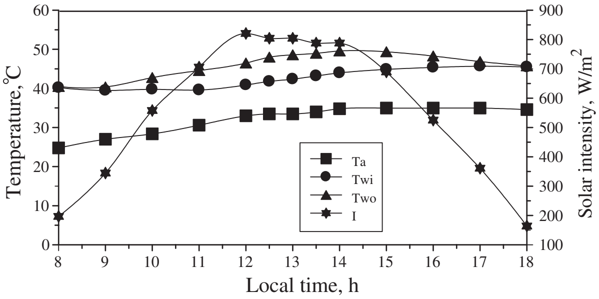

Fig. 3. Instantaneous variation of T_a, T_{wi}, T_{wo} and I with local time.

solar intensity the cooling water temperature rise increases with the decrease of the cooling water flow rate. Fig. 4 shows the comparison between the instantaneous collector efficiency for the four different cooling water flow rates and at the conditions shown in Fig. 3. For different cooling water flow rates, the instantaneous efficiency of the collector is a maximum at solar noon as shown in Fig. 4. Fig. 4 also shows that the instantaneous efficiency of the collector increases with increasing the flow rate from 0.0125 to 0.0292 kg/s, then the efficiency decreases with the increase of the flow rate from 0.0292 to 0.0625 kg/s.

For the quasi-steady state period around the solar noon, the experimental results of the second set of the experiments are represented graphically in Fig. 5 for the different cooling water flow rates. By using a curve fitting technique, it could be found that the instantaneous efficiency of the collector could be represented as a linear relation of the measuring parameter $[(T_{wi} - T_a)/I]$. The intercept of the efficiency line shown in Fig. 5 with the y-axis represent the product $F_R(\tau_g\alpha_p)_e$, while the slope of the efficiency line represents $F_R U_L$. So, the collector efficiency equation can be represented as follows:

$$\eta = F_R(\tau_g\alpha_p)_e - F_R U_L[(T_{wi} - T_a)/I] \tag{2}$$

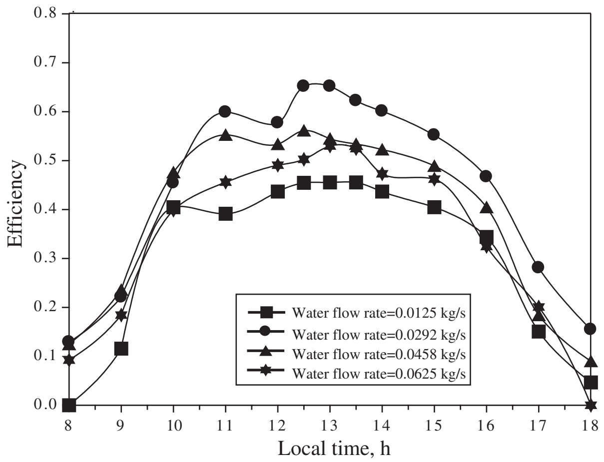

Fig. 4. Instantaneous variation of collector efficiency at different cooling water flow rates.

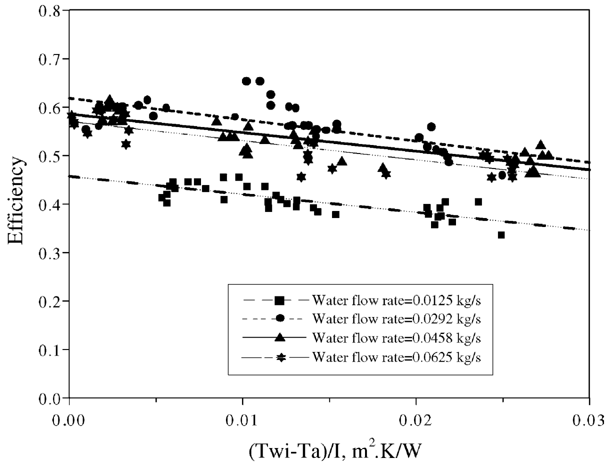

Fig. 5. Efficiency curve of the collector at different water flow rates.

The values of $F_R(\tau_g\alpha_p)_e$ and F_RU_L for the different cooling water flow rates were summarized in Table 2. As shown in Fig. 5 and Table 2 the collector efficiency is a maximum at a cooling water flow rate of 0.0292 kg/s. This cooling water flow rate is very close to the ASHRAE standard mass flow rate for testing flat-plate solar water collectors ($\dot{m}_w = 0.02A$, kg/s [15]).

To study the effect of the number of the thermosyphon tubes of the collector on the useful output and on the collector efficiency, these parameters were calculated as a function of the number of the thermosyphon tubes from:

$$q_u = \dot{m}_w C_w (T_{wn} - T_{wi}), \quad \eta = \frac{q_u}{I(L_e nP)} \tag{3}$$

where T_{wn} is the outlet cooling water temperature from the nth thermosyphon tube of the collector, and P is the pitch distance of the absorber plate. Fig. 6 shows the variation of the instantaneous useful output and the instantaneous collector efficiency for quasi-steady state period around the solar noon at the optimum mass flow rate ($\dot{m}_w = 0.0292$ kg/s) for different numbers of collector thermosyphon tubes. As shown in the figure, as the number of the thermosyphon tubes increases from 8 to 12 pipes the instantaneous useful output and the instantaneous collector efficiency increases. Increasing the number of the thermosyphon tubes over 12 decreases the instantaneous useful output and the instantaneous collector efficiency. This can be attributed to the increase of the cooling water temperature with increasing the number of tubes

Table 2
Values of $F_R(\tau_g\alpha_p)_e$ and F_RU_L for different liquid flow rates

$\dot{m}_w$ (kg/s)	$F_R(\tau_g\alpha_p)$	F_RU_L
0.0125	0.04574	3.7113
0.0292	0.6184	4.4204
0.0458	0.5859	3.8263
0.0625	0.5707	3.9666

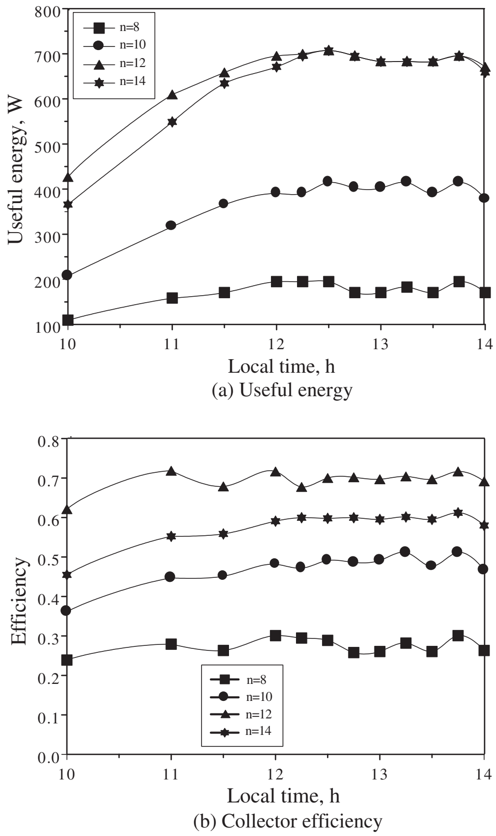

Fig. 6. Effect of the number of collector thermosyphon tubes on the efficiency.

over 12 which leads to a decrease in the temperature difference between the water and condenser section of the thermosyphon tube. Thus the rate of heat transfer from the condenser section to the cooling water decreases and this leads to a decrease of the condensation rate of the working fluid inside the thermosyphon tube. This makes the thermosyphon tube ineffective for transferring the heat from the absorber plate to the cooling water which leads to more heat losses and less collector efficiency.

Fig. 7 shows the comparison of the efficiency curve of the present design with Hussein et al. [3] two-phase closed thermosyphon tubes solar collector with tube in tube heat exchanger at the ASHRAE standard mass flow rate and under the same test rig of the present study. As shown in the figure, using the optimum number of the thermosyphon tubes, the efficiency curve of the present design has a lower slope than that of Hussein et al. [3]. Lower slope of the efficiency curve means smaller $F_R U_L$ of the collector and this means smaller heat losses from the collector. Also, Fig. 7 shows that the efficiency of the present design is higher than that of Hussein et al. [3] at high inlet water temperatures. This makes the present collector is preferred to be used in solar water heaters that work at high water temperatures.

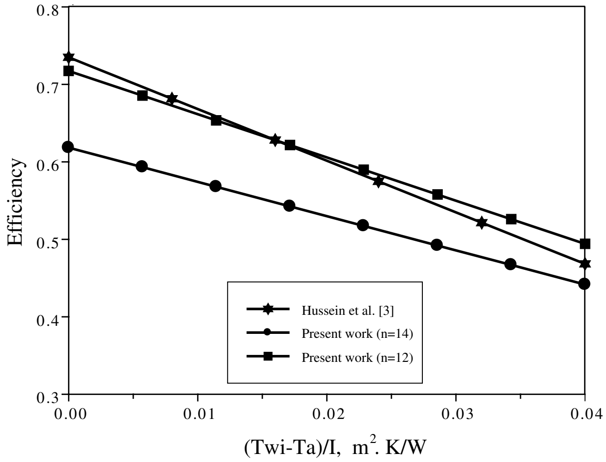

Fig. 7. Comparison between the efficiency curve of the present collector with the conventional one.

4. Conclusions

From the experimental investigation of the present two-phase closed thermosyphon flat-plate solar collector with shell and tube heat exchanger, it is concluded that:

1. The experimental results show an optimum cooling water mass flow rate at which the collector efficiency is a maximum. This optimum cooling water mass flow rate agree well with the ASHRAE standard mass flow rate for testing conventional flat-plate solar water collectors (i.e. $\dot{m}_w = 0.02$ A, kg/s).
2. The experimental results show that at the optimum cooling water mass flow rate there is an optimum number of thermosyphon tubes at which the collector efficiency is a maximum.
3. The present thermosyphon solar collector with shell and tube heat exchanger, with its simplicity in manufacturing and less material cost, gives efficiency curve close to but having better slope than that of the thermosyphon solar collector with tube in tube annular heat exchanger if optimum number of thermosyphon tubes was used.
4. Theoretical study and more experimental data are needed to optimize the collector parameter.

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