Middle-East Journal of Scientific Research 21 (12): 2359-2362, 2014
ISSN 1990-9233

DOI: 10.5829/idosi.mejsr.2014.21.12.1075

Heat Removal Factor of an Unglazed Photovoltaic Thermal Collector with a Serpentine Tube

[1,2]*M.A.M. Rosli*, [1]*K. Sopian*, [1]*S. Mat*, [1]*M. Yusof Sulaiman and* [1]*E. Salleh*

[1]Solar Energy Research Institute, Universiti Kebangsaan Malaysia, 43600 Bangi, Selangor, Malaysia
[2]Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

Abstract: Heat removal factor (F_R) is a vital parameter in determining the thermal efficiency of a photovoltaic thermal (PVT) system. As a main factor of thermal performance, F_R represents the ratio of the actual heat transfer to the maximum yield of heat transfer. In this study, $F_R$of an unglazed PVT with serpentine tube collector was determined, with focus on the flat plate thickness of the flat plate serpentine tubes. Thermal modeling was used to estimate the overall heat losses of the unglazed PVT. The highest F_R value of 0.88 was obtained in the 0.015 m-thick flat plate, followed by 0.84 in the 0.010 m-thick flat plate. The difference in F_R between the two designs was only 4.54%, which can be considered within the acceptable range for the flat plate thickness. This consideration was made based onan economical point of view and the handling issues of PVT systems.

Key words: Serpentine · Photovoltaic thermal (PVT) · Flat plate collector · Unglazed

INTRODUCTION

The thermal performance of a photovoltaic thermal (PVT) collector can be determined using three important parameters, namely,the heat removal factor (F_R), the overall losses (U_L) and the effective transmittance absorptance $(\tau\alpha)_{eff}$ that cover the PVT system, as follows [1]:

$$\eta_{th} = F_R\,(\tau\alpha)_{eff} - F_R\,U_L\left(\frac{T_i - T_a}{G}\right) \quad (1)$$

F_r is one of the crucial parametersin determining thermal efficiency [2]. F_R is defined as the ratio of useful energy to the maximum heat transfer. $F_R$has to be carefully determined because it is a gain factor for both $\tau\alpha$ and U_L. Based on Equation (1), $\tau\alpha$ should be maximum while U_L should be minimum for a PVT collector system to achieve high thermal efficiency.

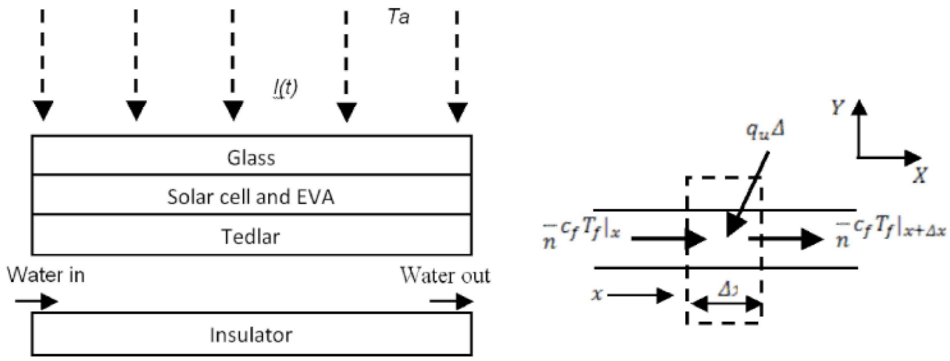
F_R is affected by several factors such as material selection, dimensions, geometry and operational variables. These parameters must be appropriately determined for the engineer to design a highly efficient and inexpensive collector [3, 4].

F_r in various collector geometries has been investigated extensively [5]. One collector geometry type is the serpentine arrangement, the attributes of which are different from that of the conventional tube collector. Consisting of one source of water that flows from the inlet to the outlet, the serpentine arrangement is considered as a complex geometry that needs to be expressed in simple equations. It is typically made of metal materials, such as copper, stainless steel and aluminum.

The serpentine tube collector has been investigated since the late 1970s [3,6,7]. The analytical study agrees well with single-bend cases and a difference of only 5% was found for cases with any number of bends [3]. They concluded that the analytical model is applicable to large numbers of tubes and the results are acceptable with a small error.

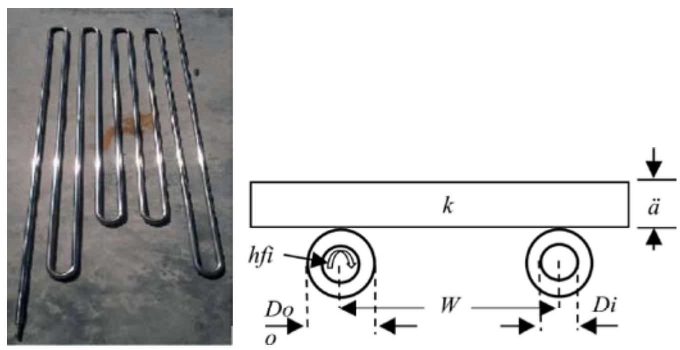
The current study aims to analyze the thermal performance of a sheet-and-tube unglazed PVT collector with serpentine tube configuration. The key objective is to determine F_R considering the variation in flat plate thicknesses of the plate absorber. Moreover, the optimum thickness for flat plates with the given operating parameters is proposed in this study.

Corresponding Author: M.A.M. Rosli, Solar Energy Research Institute, Universiti Kebangsaan Malaysia, 43600 Bangi, Selangor, Malaysia.



(a) (b)

Fig. 1: (a): Cross-section of the unglazed PVT collector, (b) heat flow balance over an element ▲ xlength below the tedlar.



(a) (b)

Fig. 3: (a) Serpentine tube arrangement (without plate) and(b) schematic diagram of the serpentine tube collector.

Thermal Modeling: An analytical study was conductedto evaluate F_R of the PVT system. The energy balance of the system was used to estimate U_L of the unglazed PVT collector. Figures 1(a), 1b (b) and 1(c) show the cross-section of the unglazed PVT collector, the balance of heat flow over an element length Δxunderneath the tedlar andthe thermal resistance circuit of the PVT system, respectively. The following assumptions wereconsidered to simplify the analysis [8,9]:

- The PVT collector is in a quasi-steady state.
- The transmittance of ethylene vinyl acetate (EVA) is 100%.
- An average temperature is assumed through each layer.
- The flow of water between the tedlar and insulation material (force mode) is uniform.
- Heat conduction is one dimensional.

$U_L$can be expressed as follows [10]:

$$U_L = U_{tw} + U_b \tag{2}$$

Table 1 shows the PVT parameters used to estimate U_L based on the thermal energy balance in Equation [10].

F_R of the serpentine tube collector was analyzed after obtaining U_L for the PVT system. In this study, bond conductance (C_b) was neglected to simplify the analysis. Figures 2(a) and 2(b) and Table 2 respectively show the serpentine tube arrangement (without plate), the schematic diagram of the serpentine tube collector and the parameters used for the calculations.

The equations below were used for the analysis [3]. Figure 3 shows the solution of F_R in terms of three dimensionless parameters known as F_1, F_2 and the dimensionless capacitance rate.

The parameters F_1 and F_2 can be obtained as follows respectively [3]:

$$F_1 = \frac{K}{U_L W}\frac{KR(1+\gamma)^2-1-\gamma-KR}{[KR(1+\gamma)-1]^2-(KR)^2} \tag{3}$$

$$F_2 = \frac{1}{KR(1+\gamma)^2-1-\gamma-KR} \tag{4}$$

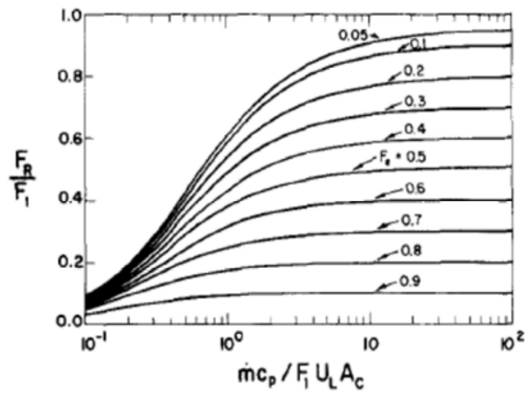


Fig. 4: Heat removal factor (F_R) for the serpentine collector plate [3].

Table 1: Parameters for the PVT collector.

Parameter	Value	Unit	Parameter	Value	Unit
hi	5.8	W/m^2.K	L_T	0.0005	m
ho	9.5	W/m^2.K	U_b	0.65	W/m^2.K
h_T	500	W/m^2.K	U_L	8.63	W/m^2.K
K_G	1	W/mK	U_T	66	W/m^2.K
Ki	0.035	W/mK	*Ut*	9.24	W/m^2.K
K_T	0.033	W/mK	U_{tT}	8.1028	W/m^2.K
L_G	0.003	m	*V*	1	m/s
Li	0.05	m			

Table 2: Parameters of the serpentine collector.

Parameter	Value	Unit	Parameter	Value	Unit
L	1.2	m	*k*	211	W/m. K
W	0.06	m	U_L	8.63	W/m^2. K
N	9		*m.*	0.01	kg/s
δ	0.005, 0.010, 0.015	m	*Cp*	4910	J/kg. K
Do	0.015	m	*hfi*	1000	W/m^2.K
Di	0.009	m			

where

$$k = \frac{(k\delta U_L)^{1/2}}{\sinh\left[(W-D)(U_L/k\delta)^{1/2}\right]} \tag{5}$$

$$\gamma = -2\cosh\left[(W-D)\left(\frac{U_L}{k\delta}\right)^{1/2}\right] - \frac{DU_L}{k} \tag{6}$$

$$R = \frac{1}{C_b} + \frac{1}{(\pi D_i h_{f,i})} \tag{7}$$

The dimensionless capacitance rate is given by

$$\frac{\dot{m}C_p}{F_1 U_L A_C} \tag{8}$$

RESULTS

The F_R results for the various plate thicknesses are presented in Table 3. The highest F_R of 0.88 was obtained using the 0.015 m-thick flat plate collector, followed by 0.84 and 0.72 for the 0.010 m and 0.005 m-thick flat plate collectors, respectively. The flat plates with the lowest and highest thickness had anapproximately 18% difference in F_R. However, F_R of the 0.010 m-thick flat plate collector and the highest F_R had a difference of only less than 5%.

Table 3: F_R Forvariousflat plate thicknesses, δ (m).

δ (m)	$\frac{\dot{m}C_p}{F_1 U_L A_C}$	F_2	F_R/F_1	F_R
0.005	0.500	0.944	0.041	0.72
0.010	0.409	0.954	0.039	0.84
0.015	0.378	0.957	0.038	0.88

CONCLUSIONS

F_R is a crucial factor in determining the thermal efficiency performance of a serpentine collector. The thickness of a flat plate serpentine collector linearly increases with F_R. Considering the economic aspect and handling issues, thickness values between 0.010 and 0.015 m area considered within the acceptable range, with only a 4.54% difference in F_R. A thicker flat plate indicates greater cost and weight, which, in turn, lead to handling issues. Thermal modeling and parametric study are important in determining an economically competitive PVT design.

ACKNOWLEDGMENT

The authors would like to thank the Solar Energy Research Institute of the Universiti Kebangsaan Malaysia and the Universiti Teknikal Malaysia Melaka for sponsoring this project.

Nomenclature:

Cp	=	Fluid specific heat
L_T	=	Thickness of tedlar
Di	=	Tube inside diameter
m.	=	Fluid mass flow rate
Do	=	Tube outside diameter
N	=	Number of segments
hfi	=	Fluid to tube heat transfer coefficient
U_b	=	Overall heat transfer coefficient from water to ambient

hi = Heat transfer coefficient insulator
U_L = Overall heat loss coefficient
ho = Heat transfer coefficient glass to ambient
U_T = Conductive heat transfer coefficient from solar cell to water through tedlar
h_T = Conductive heat transfer through the tedlar
Ut = Conductive heat transfer coefficient from solar cell to ambient through glass cover
k = Plate thermal conductivity
U_{tT} = An overall heat transfer heat transfer coefficient from glass to tedlar through solar cell
K_G = Conduction of glass
U_{tW} = An overall heat transfer heat transfer coefficient from glass to water through solar cell and tedlar
Ki = Conduction of insulator
V = Wind speed
K_T = Conduction of tedlar
W = Distance between tubes
L_G = Thickness of glass
L = Length of one serpentine segment

Greek Letter:

Li = Thickness of insulator
δ = Plate thickness

Appendix: The unknown parametersin Equation (2) are derived as follows:

$U_{tw} = \frac{h_T U_{tT}}{U_{tT} + h_T}$, $U_{tT} = \frac{U_T U_t}{U_T + U_t}$, $U_t = \left[\frac{L_G}{K_G} + \frac{1}{h_o}\right]^{-1}$,
$h_T = \left[\frac{L_T}{K_T}\right]^{-1}$, $U_b = \left[\frac{L_i}{K_i} + \frac{1}{h_o}\right]^{-1}$, $h_o = 5.7 + 3.8V$ and
$U_T = \frac{K_T}{L_T}$, $A_c = N \times W \times L$.

REFERENCES

1. Florschuetz, L.W., 1979. Extension of the Hottel-Whillier Model to the Analysis of Combined Photovoltaic/Thermal Flat Plate Collectors. Solar Energy, (22): 361-366.
2. Zondag, H., 2008. Flat-plate PV-Thermal collectors and systems: A review. Renewable and Sustainable Energy Reviews, 4(12): 891-959.
3. Abdel-Khalik, S.I., 1976. Heat Removal Factor for a Flat-Plate Solar Collector with Serpentine Tube. Solar Energy, (18): 59-64.
4. Rosli, K.S.M.A.M., S. Mat and H. Ruslan, 2013. Parametric Study on Water Based Photovoltaic Thermal Collector. In 7th International Conference on Renewable Energy Sources (RES '13), pp: 135-140.
5. William, J.A.D. and A. Beckman, 1980. Solar Engineering of Thermal Processes in Solar Engineering of Thermal Processes. New York: John Wiley & Sons, Inc.
6. He-Fei Zhang, Z.L., 1985. Thermal performance of a serpentine absorber plate. Solar Energy, 2(34): 175-177.
7. Akgiln, M.A., 1988. Heat Removal Factor for a Serpentine Absorber Plate. Solar Energy, 1(41): 109-111.
8. Dubey, S. and G. N. Tiwari, 2008. Thermal modeling of a combined system of photovoltaic thermal (PV/T) solar water heater. Solar Energy, 7(82): 602-612.
9. Dubey, S. and G.N. Tiwari, 2010. Fundamentals of Photovoltaic Modules and their Applications. RSC Publishing.
10. Tiwari, A. and M.S. Sodha, 2006. Performance evaluation of hybrid PV/thermal water/air heating system: A parametric study.Renewable Energy, 15(31): 2460-2474.