International Journal of Engineering & Technology, 7 (4.36) (2018) 389-393

International Journal of Engineering & Technology

Website: www.sciencepubco.com/index.php/IJET

Research paper

Thermal Energy Recovery from a Grid Connected Photovoltaic-Thermal (PVT) System Using Water as Working Fluid

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Abstract

A Photovoltaic Thermal collector (PVT) is a combination of Photovoltaic (PV) and Thermal (T) collector. Many studies have tried to improve the electrical efficiency and thermal efficiency of this PVT system. The efficiency is influenced by many system design parameters and operating conditions such as the absorber temperature, velocity and pressure distributions. In this study, two new design concepts of absorber configuration of thermal collector have been investigated. This study also provides an important opportunity to advance the understanding of the effect of different geometrical configuration on the performance of the absorber. Simulations were performed using ANSYS FLUENT 16.0 for both absorbers to determine the best absorber design that gives the highest thermal efficiency. Based on the simulations performed, perpendicular serpentine absorber proved to be the best design with the higher thermal efficiency of 56.45%.

Keywords: *Photovoltaic (PVT); Thermal Efficiency; Absorber Collector; Pressure Drop*

1. Introduction

The demand for energy has been a crucial concern for the modern era because of rapid Technological advancement around the world. In turn, the world must depend on unsustainable ways (Coal, Natural Gas, Nuclear) of producing energy to quench the global thirst for high energy. These unsustainable ways are cheaper and quicker to produce but the nature must pay a huge cost in terms of climate change and global warming. Hence, there is a dire need to find a clean and sustainable energy source [1]. Although nature has provided us with abundant sources of clean energy (Solar, Wind, Tidal etc.), we are yet to find a feasible method to harness them. Nevertheless, Solar energy has shown a great promise as an alternative clean energy source. It is known as the most abundant, inexhaustible and cleanest energy resources among all other available energy resources [2].

Photovoltaic (PV) cells are the only device that are commercially available for conversion of solar radiation to electricity and it has many advantages over other renewable energy sources [3]. Nevertheless, the efficiency of a PV system is affected the rising temperature from the solar radiation. In fact, in some weather conditions the temperature of PV cell can reach up to 70°C [4]. Only about 20% of the solar radiation which falls onto a PV cell is being converted to electricity, the rest is either reflected back to the atmosphere or generate heat in the system [5]. Although, there are many research is being conducted to overcome this problem, the promising solution using a solar thermal collector beneath the PV cell [6–9]. This is system is generally known as the Photovoltaic Thermal (PVT) system.

Wolf [10] brought the idea of using a working fluid in PVT thermal collector. He investigated the performance of a PVT system using a fluid through the thermal collector. The major advantage of this system is that it would could cool down the PV cell and capture thermal energy which can be later used for domestic heating purposes. He also concluded that PVT system is the most economically feasible system to enhance the efficiency of PV system. In a PVT system, the thermal collector is a tube which is imbedded onto the PV cell to extract the heat by the cooling fluid [11].

Ji et al. [12] designed a PVT system with natural water circulation and he obtained an overall, thermal and electrical efficiency of 10.15%, 45% and 52% respectively. Kiran and Deviga [13] also studied the effects of using a solar thermal collector on a PV system. They study was concluded with an efficiency of 8.16% and 50.80% for electrical and thermal respectively. Further, [14] investigated the PVT system in the weather conditions of Mumbai, India and the study yielded and thermal and electrical efficiency of 68.2% and 12.9% respectively. Fudholi et al. [15] experimented on various designs of thermal collector to enhance the thermal and electrical efficiency of the PVT system. Fig. **1** shows the different thermal collector designs of the proposed system. It was concluded that the spiral flow configuration had the highest efficiency of 13.8%, 54.6% and 68.4% for electrical, thermal and overall under solar radiation of $800 W/m^2$ and with a mass flow rate of $0.041 kg/s$.

Similarly, Rosli et al. [16] investigated the performance of a serpentine tube thermal collector in an unglazed PV system. [17] Performed a simulation study with several different configurations of thermal collector and concluded that spiral tube design yielded the highest thermal efficiency of 50.12% and electrical efficiency of 11.98%.

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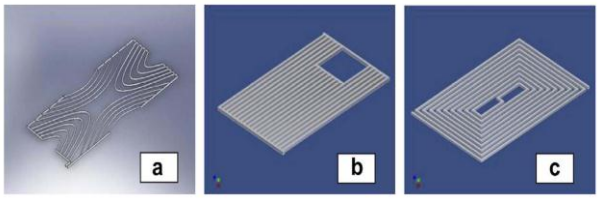


Fig. 1: (a) Web Flow (b) Direct Flow (c) Spiral Flow [18]

This study is focused on PVT water-based system where new thermal absorber configuration are designed and their thermal efficiency are investigated with various parameters such as solar radiation and flow rate condition through series of simulations. Numerical Simulations were performed through Microsoft Excel equation solver. Moreover, analysis using ANSYS Software 16.0 was also conducted to show the temperature distribution through the absorber.

2. Method

Fig. **2** shows the complete measuring setup for the PVT system. In an experimental laboratory, the PVT collector will be exposing to various level of solar radiation. The solar radiation falling on the PV induces movement of charges which eventually produces electrical power. The excess solar radiation is absorbed by the solar thermal collector in a control manner depending on the mass flow rate of the working fluid from storage tank. All experimental data can be collected through a data-acquisition system provided.

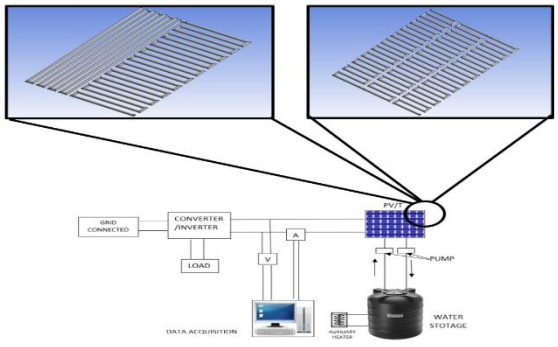


Fig. 2: Schematic diagram of PVT water collector

Table 1 shows the parameters of the absorber collectors designed. The absorbers are in the shape of round hollow tubes attached closely underneath the PV module to ensure a zero gap or no gap available between the tubes and the PV module where heat could be transferred. Rockendorf and later Chow have discovered that the fin efficiency and bonding quality between the collector and the sheet under the PV module is a crucial factor which often bring limitations to the overall efficiency achievable [19]. **Table 2** shows the system parameters for inputs to the simulation model.

Table 1: Parameters of the solar thermal collector

Absorber Type	Details
Perpendicular Serpentine	Absorber material: round hollow tube of copper Absorber collector module: 1 channel each of size 12.7mm diameter, 1mm(thick), 1000mm(length) and 640mm(width) Inlet/outlet no: 2
Oscillatory	Absorber material: round hollow tube of copper Absorber collector module: 1 channel each of size 12.7mm diameter, 1mm(thick), 1000mm(length) and 640mm(width) Inlet/outlet no: 2

Table 2: System parameters for inputs to the simulation model

Description	Symbol	Value	Unit
Ambient temperature	T_a	20	℃
Collector area	A_c	0.65	m^2
Number of glass cover	N	1	-
Emittance of glass cover	ε_g	0.88	-
Emittance of plate	ε_p	0.95	-
Collector tilt	θ	14	°
Fluid thermal conductivity	k_f	0.613	W/m ℃
Specific heat of working fluid	C_p	4180	J/kg ℃
Back insulation conductivity	k_b	0.045	W/m℃
Back insulation thickness	l_b	0.05	m
Insulation conductivity	k_e	0.045	W/m ℃
Edge insulation thickness	l_e	0.025	M
Absorber conductivity	k_{abs}	51	W/m ℃
Absorber thickness	l_{abs}	0.002	m
Fin conductivity	k_f	84	W/m ℃
Fin thickness	δ	0.0005	m
Heat transfer coefficient from cell to absorber	h_{ca}	45	W/m ℃
Heat transfer inside tube	h_{fi}	333	W/m ℃
Transmittance	τ	0.88	-
Absorptance	α	0.95	-
Absorptance		0.95	-

3. Energy Analysis of PV/T system

The performance of a PV/T system is determined by various Operating conditions and design parameters. In this proposed study, the simulation is analysed by varying the level of solar radiation, ambient temperature, and the flow rate. Further, the thermal collector of the system is a flat-plate collector with a single glazing sheet. Hence, the Hottel-Whillier equations were used to evaluate the performance of the PV/T system [20]. The thermal efficiency of a flat-plate solar collector can be expressed as the following:

$$n_{th} = \frac{Qu}{s} \quad (1)$$

Where, it is the ratio of the useful thermal energy (Q_u) to the overall incident of the solar radiation (S). The useful heat gained by the soar thermal collector could be expressed as the following, where, mass flow rate (ṁ), heat capacity of the cooling fluid (C_p), inlet (T_i) and the outlet (T_o) temperature of the cooling fluid:

$$Qu = \dot{m}C_p(T_o - T_i) \quad (2)$$

Further, the thermal efficiency can be calculated from the following Hottel–Whillier equations. Where, A_c is the area of the thermal collector, F_r is the Heat removal factor, G_T is the amount of solar radiation at NOCT (It is assumed the radiation level of 800 W/m^2 with a wind velocity of 1 m/s and the ambient temperature at 26 ºC), T_i is the inlet and T_a is the ambient temperature [21]:

$$Q_u = A_c F_r[G_T(\alpha\tau)_{PV} - U_L(T_i - T_a)] \quad (3)$$

Heat removal efficiency factor can be calculate using the following expression [22] :

$$F_R = \frac{\dot{m}C_p}{A_c U_L}\left[1 - \exp\left(-\frac{F'U_L A_c}{\dot{m}C_p}\right)\right] \quad (4)$$

Where F′ is the thermal collector efficiency and can be determines using the following expression. Where, a and b is the width and the height of the duct, C_b is the conductance between the fin and the tube of the collector, h_{fi} is the heat transfer coefficient of the working fluid, D_h is the hydraulic diameter of the tube and the F is the fin efficiency factor:

$$F' = \left[\frac{\frac{1}{U_L}}{U_L(D_h + (W - D_h)F}\right] + \frac{1}{C_b} + \frac{1}{2(a+b)h_{fi}} \quad (5)$$

The fin efficiency factor can be expressed by:

$$F = \left[\frac{\tanh\left(M\frac{W-D_h}{2}\right)}{\sqrt{M\frac{W-D_h}{2}}} \right] \quad (6)$$

Where:

$$D_h = \frac{2ab}{(a+b)} \quad (7)$$

The coefficient of the M from the Eq (6) is analysed using the thermal conductivity of the absorber and PV cell [23]. Where, U_L is the overall heat loss coefficient, k_{abs} is the absorber thermal conductivity, l_{abs} is the thickness of the absorber, k_{pv} is the conductivity of the PV cell and the l_{pv} is the thickness of the PV cell :

$$M = \sqrt{\left(\frac{U_L}{k_{abs}l_{abs}k_{pv}l_{pv}}\right)} \quad (8)$$

The overall loss coefficient can be determined by the following expression where, U_e and U_t is the sum of edge and the top loss coefficient:

$$U_L = U_e + U_t \quad (9)$$

$$U_e = \frac{k_{el}pl}{L_eA_c} \quad (10)$$

$$U_t = \left\{ \frac{N}{\frac{C}{T_{pm}}\left[\frac{T_{pm}-T_a}{(N+f)}\right]} \right\}^{-1} + \frac{\sigma(T_{pm}+T_a)(T_{pm}+T_a)}{(\varepsilon_p+0.00591Nh_W)^{-1}+\frac{2N+f+1+0.133\varepsilon_p}{\varepsilon_g}-N} \quad (11)$$

Where:

$$C = 520(1 - 0.000051\beta^2) \quad (12)$$

$$f = (1 + 0.089h_w - 0.1166h_w\varepsilon_p)(1 + 0.07866N) \quad (13)$$

$$e = 0.43\left(1 - \frac{100}{T_{pm}}\right) \quad (14)$$

$$T_{pm} = T_i + \frac{\frac{Q}{A_c}}{F_rU_L}(1 - F_r) \quad (15)$$

p - Perimeter of the thermal collector
σ - Stefan-Booltzmann constant
N - The number of glass cover
ε_p - Emittance of the plate
ε_g - Emittance of the glass cover
β - Collector tilt Angle
T_{pm} – Mean Plate temperature
h_w – Wind Transfer Coefficient

The forced convection (h_w) can be determined the following expression [24] :

$$h_w = 2.8 + 3.0v \quad (16)$$

And, the natural heat transfer (h_{nat}) can be calculated using:

$$h_{nat} = 1.78 + (T_{pm} - T_a) \quad (17)$$

Further, the overall convection heat transfer coefficient (h_c) and the overall top loss heat transfer coefficient for the collector can be determined by combining the natural and the forced convection (Eq (16) and (17)) [25]:

$$h_c = \sqrt{{h_w}^3 - {h_{nat}}^3} \quad (18)$$

Hence, the overall heat gained by the PV/T system can be calculated using the Eq (1)-(18). The thermal efficiency of the thermal collector can be expressed as [26]:

$$\eta_{th} = F_R(\alpha\tau)_{PV} - F_RU_L\frac{T_i-T_a}{S} \quad (1)$$

4. Results and Discussion

The numerical model has been validated by comparing the experimental results of thermal efficiency with those obtained numerically. Spiral design of absorber was selected for carrying out the experimental validation due to its highest thermal efficiency. As can be seen in Figure 3 and **Table 3**, the experimental values of thermal efficiencies are comparable with the corresponding numerical values over the entire range of mass flow rate for the spiral absorber design.

Table 3: Data for validation of numerical simulations with experimental data

Mass flow rate (kg/s)	Experimental Thermal Efficiency (%)	Numerical Thermal Efficiency (%)	Percentage Error (%)
0.011	46.43	42.35	8.79
0.013	48.53	46.35	4.48
0.016	50.01	49.45	1.11
0.020	50.98	52.61	3.20
0.024	52.47	55.83	6.40
0.027	52.67	56.89	8.02
0.029	52.84	57.86	9.50
0.032	52.97	58.49	10.41
0.035	53.08	59.08	11.30
0.038	53.19	59.76	12.36

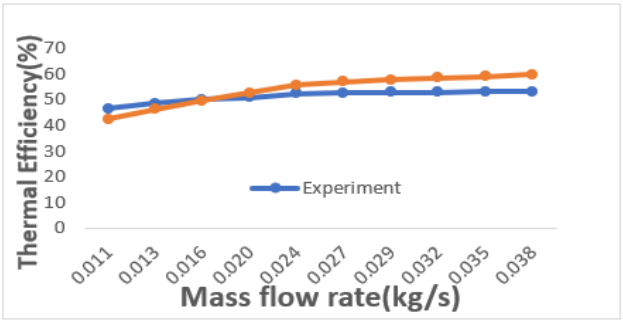


Fig. 3: Validation of the experimental results by thermal efficiency

This slight difference is due the uncontrollable outdoor meteorological conditions such as ambient temperature, water inlet temperatures and wind speed of the experimental site[20]. Hence, the thermal efficiency curves in Fig. **3** is found to be in agreement with each other qualitatively as well as quantitatively. Therefore, upon validation of the proposed mathematical model, it can now be used to generate numerical simulation results for performing the parametric study of the other absorber designs which is perpendicular serpentine flow design and oscillatory flow design.

From the validated module obtained, thermal efficiency of the two new design of absorber are simulated. Fig. **4** shows increase in thermal efficiency with mass flow rates at 800 W/m^2 solar radiation. The perpendicular serpentine absorber produced higher thermal efficiency, exhibited an increase in the thermal efficiency from 40.09% to 56.45% because of the increased mass flow rate from 0.11 kg/s to 0.038 kg/s. The oscillatory absorber exhibited an increased thermal efficiency from 35.65% to 50.00% when the mass flow rate increased from 0.011 kg/s to 0.038 kg/s. Solar radiation also affected the thermal efficiency of PVT collector. The effects of mass flow rates on the absorber collectors are shown in Fig. **5**. The mass flow rates used in this analysis (0.011-0.038 kg/s) were later applied under various solar radiation levels. The results show that thermal efficiency increased simultaneously with increased in mass flow

rates and solar radiations levels respectively. The Perpendicular serpentine absorber has been designed with smaller(tight) spacing between the tubes, while the Oscillatory absorber have slightly larger spacing gap between the tubes. It is believed that the closer the spacing gap covered by the entire PV module, the more heat can be absorbed[17]. This concurrently increase the efficiency of the thermal system.

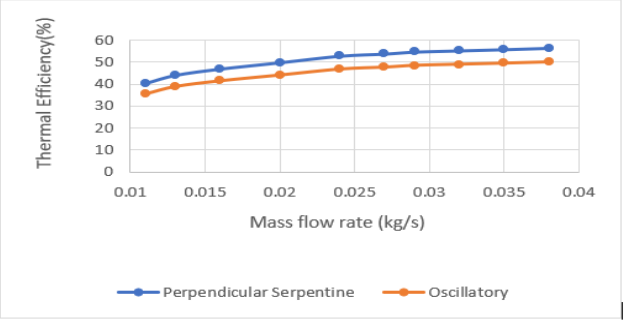


Fig. 4: Changes in thermal efficiency of 800 W/m^2 solar radiation with the mass flow rate for different flow absorbers

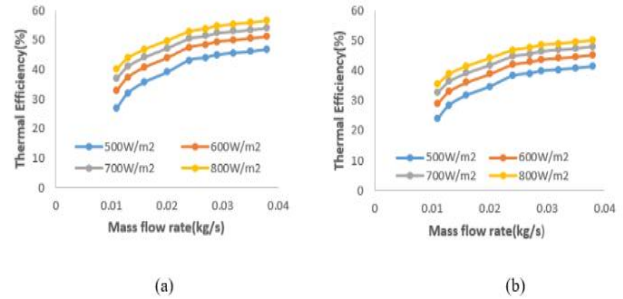


Fig. 5: Changes in thermal efficiency of (a) perpendicular serpentine (b) oscillatory absorber with the mass flow rate under different solar radiation levels.

4.1. Grid Independent Testing

Meshing for the absorber was carried out with three types of meshing which is coarse, medium and fine meshing. In order to develop both absorber models, free tetrahedral and free triangular mesh setting were used. It was observed that there was no considerable change in value of outlet temperature between medium and fine meshing. Hence, the model with number of elements 3,224,717 was considered for the simulations.
The effect of temperature, velocity and pressure distribution along the flow channels of absorber is illustrated as 3D simulated. This evaluation has been done at a constant inlet temperature at 27°C.

Table 4: Grid independent test with inlet velocity of 0.225 m/s

Meshing Type	Coarse	**Medium**	Fine
No. of Elements	2,716,891	**3,224,717**	3,779,409
T_{out} °C	303.29 K	**305.74 K**	306.41 K

4.2. Effect of Inlet Velocity on Temperature, Velocity and Pressure Distribution Throughout the Flow Channel of Both New Designed Absorber.

The inlet velocity is 0.225 m/s. In PV/T systems, the velocity of the heat transfer fluid is usually kept in the laminar flow to accumulate more heat from the PV module. Figure 6 shows the temperature distribution for both flow configuration, it can be observed that the exit temperature of the Perpendicular serpentine is about 308.2K and that of oscillatory is 307.7K, this shows that both configuration demonstrated almost equal level of thermal energy absorption.

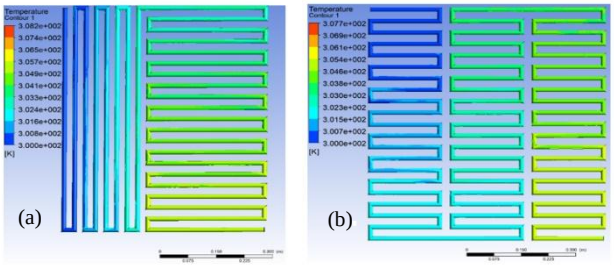


Fig. 6: Temperature distribution under Inlet Velocity = 0.225 m/s, (a) Perpendicular serpentine (b) oscillatory

Velocity distribution is illustrated in Figure 7 for both absorbers with different inlet velocity. It was observed that the velocity gradually increases along the way through the absorber. This is because of the changes in pressure along the tube where velocity is inversely proportional to pressure [25]. The bending part of tube shows that decreasing of velocity occur at each corner of bending of tubes. The velocity also decreases when the fluid near the wall of the tubes because of the effect of friction between fluid and wall while the velocity at the middle of the tube has higher velocity.

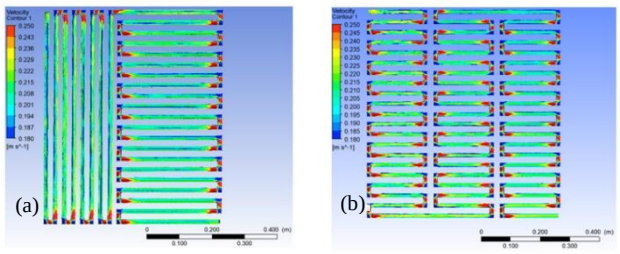


Fig. 7: Velocity distribution under Inlet Velocity = 0.225 m/s, (a) Perpendicular serpentine (b) oscillatory

The pressure distribution for both absorbers at inlet velocity 0.225 m/s is illustrated in Figure 8. At the inlet area of the absorber, the pressure is high, and the pressure gradually decrease along the tubes flow due to wall friction and local pressure drop. The minimum pressure is observed at the outlet of the tubes. It is observed that Perpendicular serpentine absorber has slower pressure drop along the tube flow because of its bending. On the other hand, the oscillatory absorber has higher pressure drop due to more turns in its tube flows. Therefore, this higher-pressure drop will lead to higher pumping power needed to maintain the flow inside the tube of oscillatory absorber. In addition, according to the result, it shows that increase in inlet velocity will increase the pressure.

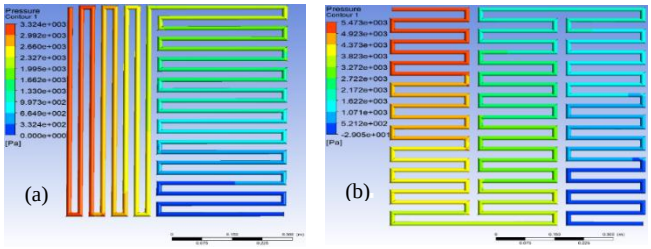


Fig. 8: Pressure distribution under Inlet Velocity = 0.225 m/s, (a) Perpendicular serpentine (b) oscillatory

5. Conclusion

A theoretical model was developed based on this present study. It was found that different configurations of absorber collector affected the thermal efficiency of thermal collector directly. The performances of the two new designs of absorber is determined. In comparing the two absorbers, the result indicates that at solar radiation level of 800 W/m2 and mass flow rate of *0.225 m/s* m/s, the perpendicular serpentine absorber produced thermal efficiency of 56.45%, which is higher than oscillatory absorber that produced

50.00% of thermal efficiency. It was found that the perpendicular serpentine absorber has lower pressure drop, thus, lower pumping power needed to operate the system using this absorber. This proved that perpendicular serpentine absorber is better than oscillatory absorber and should be considered at least to run an experiment with exact size of the absorber. For further study, it is advisable to perform simulation software at early stage of design to make sure correct justification of design configuration of any designed proposed. Next, the gap or spacing between the tubes of absorbers play an important function in the design configuration of absorber. The lower the gap enable to obtain higher thermal efficiency.

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