Temperature distribution and efficiency assessment of different PVT water collector designs

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Abstract. The aim of this work was to study different hybrid photovoltaic-thermal (PVT) collectors. Numerical simulation of the collectors was carried out using a 3-dimensional computational fluid dynamic (CFD) model. A set of experiments was conducted using a box-type PVT collector to validate the obtained computed data. Simulation results for water outlet temperature, especially at the lower flow rates, were closer to the measured data compared with average module surface temperature. Based on the simulation results, the spiral flow designs of heat exchanger provided the most uniform temperature distribution for module surface and the highest temperature gradient was observed with the direct flow designs. A hot zone was generated on the module surface of the direct flow box-type collector and the results indicated that the hot zone was considerably declined using cross-fins inside the box-type heat exchanger. The highest heat transfer to the ambient occurred with the spiral flow design and the box type heat exchanger presented a relatively low heat losses. The cross-fined box type and the spiral flow designs of the PVT collector had the highest electrical efficiencies at the different test conditions.

Keywords. Numerical computation; experimental set-up; photovoltaic-thermal collector.

1. Introduction

In the last decade, energy consumption has dramatically increased and in spite of the environmental problems associated with the fossil fuels, the demands for coal, crude-oil and natural gas are expected to grow at least until 2040. According to the International Energy Agency's reports, global energy consumption will rise by 37% by 2040 [1]. Solar energy is one of the best alternatives to the fossil fuel resources because of its clean, renewable and free nature. Photovoltaic (PV) modules generate electricity directly from the sun light [2]. However, they turn only a small part of the solar irradiance into electricity, while a greater part is transformed into heat and increases the cell temperature which leads to a decrease in the electrical performance [3]. The hybrid Photovoltaic Thermal (PVT) systems are new solar devices, which can simultaneously generate both electricity and heat from the incident solar radiation [4, 5]. PVT systems are made up of PV modules coupled with heat extraction devices in them a coolant fluid like air or water is warmed and at the same time the PV module temperature declines [6]. In this manner, a considerable part of the available solar energy can be converted into heat and consumed for low-temperature heating

applications like water and building heating systems [7–11] and solar dryers [12, 13].

A survey on the literature shows that the efficiency of PVT systems is affected by their configuration design and the location they are used [14–16]. It is understandable from the previous works that well-designed hybrid PVT systems can exhibit better performance than the two separate solar heating and PV systems [17]. A number of studies have been attempted to model and compare different designs of PVT air and water collectors [18, 19]. In a research, the suitable water flow rate in different designs of tube type PVT collector was determined using finite element method (FEM). The researchers reported that the PVT collector with the fewer strings in horizontal direction presented the best performance [20]. Zhou *et al* (2015) have applied FEM to investigate the effect of some environmental and structural parameters on temperature distribution of a PV module. It was shown that higher wind speed and lower ambient temperature are helpful for decreasing working temperature and enhancing electrical efficiency of the PV module [21]. In another research, it was tried to develop a transient model for combined PV module and different phase changed materials (PCM), and the researchers used finite difference technique for discretization of the energy balance equations. The results deduced a maximum thermal efficiency improvement of 5% using hybrid PV-PCM [22].

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Figure 1. Cross section of: (a) round tube, (b) square tube and (c) box-type PVT water collectors. 1: insulator, 2: glass, 3: silicon cell, 4: tedlar, 5: absorber sheet, 6: round tube, 7: square tube, 8: box channel.

Recently, in Iran, the government policy in subsidy reform and increase in fossil fuels price led to an increasing interest in use of renewable energies for residual and industrial applications. In spite of this, the PV power generator investment has not been economically feasible, so far [23]. Hybrid PVT devices are well-known as an alternative method to the conventional PV systems which can improve their energy performance and therefore, economic feasibility [24–27]. The present study is aimed at numerical simulation of three different types of water PVT collectors (including round-tube and absorber sheet, square-tube and absorber sheet and box type) using a computational fluid dynamic (CFD) model. Though a number of researchers have tried to simulate the PVT collectors, the literature review indicated a lack of reports about the comparison of temperature distribution and thermal performance of these types of PVT water collectors.

2. Material and methods

2.1 Description of different hybrid PVT collector designs

One of the most conventional PVT water collectors that is comprised of an absorber sheet and tube type heat exchanger is known as sheet and tube water collector. In the present

study, thermal performance of various round and square-tube types of sheet and tube PVT collectors was investigated. The round tube and sheet PVT collector included a photovoltaic module, a round tube heat exchanger and a thermally insulated body. The heat exchanger used in this type of collector was comprised of a copper tube (with different flow designs) connected to a copper absorber sheet which was located underneath the PV module to extract its accumulated heat by means of thermal conduction. Figure 1 shows the cross-section of the different PVT collector designs. As it can be seen from the figure, the PV module is a conventional glass to tedlar type, which consists of glass, silicon cell and tedlar. The insulating body of the collector was assumed to be a glass-wool sheet. Design parameters used for different components of the PVT collectors were listed in table 1.

One of the problems of the round tube collectors is the weak connection between tubes and absorber sheet in the manufacturing process. Tubes with the square section can somewhat solve this problem. According to figure 1b, square tube has a larger contact surface with the absorber sheet compared with the round tube-type. The box or channel-type heat exchangers provide a fully connection between PV module and coolant water (figure 1c). The box-type heat exchangers have a relatively simple structure. However, edge sealing is the major manufacturing problem of this type of collectors.

Table 1. Design parameters of the PVT collectors.

Component	Parameter	Detail
PV module	Thickness of glass (m)	0.003
	Thickness of silicon cell (m)	0.00025
	Type of silicon cell	Monocrystalline
Round-tube heat exchanger	Thickness of tedlar (m)	0.0005
	Tube diameter (m)	0.015
	Thickness of tube (m)	0.001
	Length of tube (m)	15.6
	Material of tube	Copper
	Absorber sheet dimensions (m)	1.8×0.8×0.001
	Material of absorber sheet	Copper
	Flow configuration	Oscillating
		Spiral
		Direct
Square-tube heat exchanger	Section dimension (m)	0.015
	Thickness of tube (m)	0.001
	Length of tube (m)	15.6
	Material of tube	Copper
	Absorber sheet dimensions (m)	1.8×0.8×0.001
	Material of absorber sheet	Copper
	Flow configuration	Oscillating
		Spiral
		Direct
Box-type heat exchanger	Box dimensions (m)	1.5×0.75×0.15
	Thickness of box (m)	0.001
	Material of box	Copper
	Flow configuration	Oscillating
		Direct
		Glass wool
	Insulator thickness (m)	0.02

To investigate the effect of flow configuration on the temperature distribution and thermal performance of the PVT collectors, different conventional flow designs were considered. The flow configurations used for the sheet and tube collectors were oscillating, direct and spiral flow while those were direct (without fins) and oscillating (with cross fins) flow designs for the box-type collector (figure 2).

2.2 Governing equations and method of computation

Water flow inside the collectors is governed by the usual continuity, momentum and energy equations. These equations for incompressible fluid flow can be respectively expressed as follow:

$$\nabla \cdot \vec{V} = 0 \quad (1)$$

$$\frac{\partial \vec{V}}{\partial t} + (\vec{V} \cdot \nabla) \vec{V} = -\frac{1}{\rho} \nabla p + \nu \nabla^2 \vec{V} + \rho \vec{f} \quad (2)$$

$$\rho C_p \left[\frac{\partial T}{\partial t} + (\vec{V} \cdot \nabla) T \right] = k \nabla^2 T + \varphi \quad (3)$$

where $\vec{V}$ is the velocity vector (m/s), $\vec{f}$ is the total body force per unit mass (N/kg), ρ is the specific mass (kg/m³) and ν stands for kinematic viscosity of the fluid (m²/s), T and k respectively stand for temperature (K) and thermal conductivity (W/mK), and φ , the viscous dissipation, is expressed as:

$$\varphi = \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + 2 \left[\left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial y} \right)^2 \right] \quad (4)$$

where u and v are respectively the x- and y- components of velocity (m/s).

Numerical solution of the continuity, momentum and energy equations was carried out using a 3-dimensional computational fluid dynamic (CFD) model to predict the temperature distribution of the water flow and the PV panel. For this purpose, ANSYS CFX software (version 16) was employed. To build a suitable mesh network for the models, ANSYS meshing technology was employed. Using this technology, it was possible to modify and refine the mesh around the critical areas. Figure 3 shows a view of mesh applied to the domain in the software environment. To ensure the independency of obtained results from grid, the

Figure 2. Different flow designs: (a) oscillating flow, (b) direct flow, (c) spiral flow, (d) cross-fined box.

Figure 3. A view of mesh applied to one of the solar collectors in the software environment.

number of fine meshes was at least 1.5 times of coarse meshes in each dimension [28] and the grid independency test was done. The solutions converged when the values of residuals in the computational domain fell below 1×10^{-6} and an acceptable error of 0.6% was achieved [29]. Finally, the total number of elements for the different collector designs was between 1320000 and 3650000.

Thermal and physical properties of the different components, including: glass, silicon cell, tedlar, coolant water, copper tubes and sheet and glass wool were taken from [18, 21, 30, 31] (table 2). Modeling was conducted at the solar radiation intensities of 600, 800 and 1000 W/m², ambient temperature of 25°C, water inlet temperature of 25°C, wind speed of 0.5 m/s and three levels of water flow rate (including 30, 90 and 180 L/h). Table 3 illustrates the boundary conditions used in the simulation process. The convective heat transfer coefficient, in table 1, was given by the following expression [30].

$$h_o = 2.8 + 3V_w \quad (5)$$

where h_o is the convective heat transfer coefficient around the outer surfaces of the collector (W/m²K) and V_w shows the wind speed (m/s).

In order to, determine the average temperature of the module surface based on the graphical results of temperature contours, 20 points with the same distances were considered on the surface. The electrical efficiency (ζ_{el}) of PV module, as a function of cell temperature, is obtained by the following equation [32].

$$\zeta_{el} = \zeta_r (1 - 0.0045(T_c - 25)) \quad (6)$$

where ζ_r is the efficiency of PV module at the reference temperature (25°C) and T_c presents the average cell temperature (°C) which was assumed to be equal to the average module surface temperature.

2.3 Set-up description and experimental validation

To verify the simulation results, a direct flow box-type PVT water collector was developed in this research. Heat exchanger was constructed from galvanized steel with thickness of 0.4 mm. The walls and outside surface of the collector were insulated by 20 mm thick glass-wool. Diameter of the inlet and the outlet pipes of the heat exchanger was 15 mm, and length, width and height of the box were 150, 75 and 1.5 cm, respectively. A photograph of the designed PVT collector is shown in figure 4a.

The evaluation tests were conducted in the city of Kerman, Iran during September 27 to October 12, 2015. Each experiment was started at 8 am and continued until 3 pm. The tests were carried out at three different water flow rates of 30, 60 and 90 L/h.

Module surface temperature at 3 points, ambient air, inlet and outlet water temperatures were measured using LM 35

Table 2. Thermal and physical properties of the material.

Material	Density (kg/m ³)	Specific heat (J/kgK)	Thermal conductivity (W/mK)
silicon	2330	760	0.1
Tedlar	1200	1250	0.033
glass	2500	750	0.8
Glass wool	50	670	0.004
Water	997	4181.7	0.6069
Copper	8933	385	401

Table 3. Boundary conditions applied to the models.

Boundary condition	Description
Fluid flow rate	30, 90 and 180L/h
Inlet temperature	298K
Heat flux	600, 800 and 100 W/m ² , solar irradiance on top surface
Convection to ambient	Convection coefficient=7.6 W/m ² K, ambient temperature= 298.3K, heat transfer from the outer surfaces of the collector

temperature sensors. A solar power meter (TES 1333) was utilized to measure solar radiation intensity on the collector surface. Performance parameters of the PV module (including shortcut current, maximum current and voltage, open circuit voltage and maximum power) were determined using a PV solar analyzer (PROVA 200). Thermal and electrical efficiencies of the hybrid PVT collector were obtained by equations (7) and (8), respectively [33]. A schematic of the instrumentation used for the experiments is shown in figure 4b.

$$\zeta_{th} = \frac{\dot{m}_w C_w (T_o - T_i)}{I_t A_c} \times 100 \quad (7)$$

$$\zeta_{el} = \frac{P_m}{I_t A_c} \times 100 \quad (8)$$

where ζ_{th} represent thermal efficiency (%), $\dot{m}_w$ is the mass flow rate (kg/s), C_w is the specific heat of water (J/(kg K)), T_i and T_o are respectively the inlet and outlet temperatures (°C), I_t represents the solar radiation intensity (W/m²), P_m is the maximum power of PV and A_c shows the collector surface area (m²).

A set of CFD simulation process was separately conducted based on the test conditions and the parameters of the fabricated PVT collector, and the obtained results were compared with those of the experiments.

In order to validate the simulation results, two statistical criteria including coefficient of determination (R^2) and root mean square deviation (e) were calculated as follows [34, 35]:

Figure 4. (a) A photograph of the manufactured PVT collector, (b) schematic of the instrumentation: 1- temperature sensor, 2- temperature monitoring device, 3- solar power meter, 5- solar analyzer.

$$R^2 = \frac{\sum_{i=1}^n (X_i - Y_i)(\bar{X}_i - \bar{Y}_i)}{\left[\sum_{i=1}^n (X_i - Y_i)^2 \right] \left[\sum_{i=1}^n (\bar{X}_i - \bar{Y}_i)^2 \right]} \quad (9)$$

$$e = \frac{\sqrt{\sum (e_i^2)}}{n} \quad (10)$$

where

$$e_i = \left[\frac{X_i - Y_i}{X_i} \right]$$

where X_i and Y_i are respectively is the i^{th} computed and experimental temperature (including outlet and surface temperatures) and n represents the number of computed (or experimental) data.

Figure 5. Comparison of the experimental and theoretical data; (STT) theoretical surface temperature, (STE) experimental surface temperature, (OTT) theoretical outlet temperature, (OTE) experimental outlet temperature.

3. Results and discussion

3.1 Experimental data and model verification

Figure 5 compares the average module surface and water outlet temperatures measured during the experiments with

those computed by ANSYS CFX software. It is clear that the simulation results can be satisfactorily verified by the experimental data. The mean square deviation ranged from 5 to 12% while the observed correlation coefficient was between 0.89 and 0.98. Furthermore, compared with average surface temperature, simulation results for outlet temperature

Figure 6. Thermal efficiency of the designed PVT collector at the different flow rates.

Figure 7. Electrical efficiency of the designed PVT collector at the different flow rates.

especially at the lower flow rates were closer to the measured data. Figure 5 also indicated that working temperature of the PVT collector is lower at the higher water flow rates, owing to decrease in temperature of the moving fluid because of faster passing through the heat exchanger. Similar results were reported by previous researchers [36–38]. It can be said from the observations that increasing the flow rate from 30 to 90 L/h led to a reduction of 6 and 18% in surface and outlet temperatures, respectively.

Figure 6 illustrates thermal efficiency of the fabricated box-type PVT collector at the different water flow rates. Since convective heat transfer coefficient of water increases with its velocity, thermal efficiency was better at the higher flow rates. This finding is in accordance with the results of [3, 38]. A maximum average thermal efficiency value of

56% was experimentally achieved at flow rate of 90 L/h, while Zondag *et al* (2003) has reported a thermal efficiency of 52% for an uncovered sheet and tube PVT solar water collector at flow rate of 72 kg/h [19].

The effect of different flow rates on electrical efficiency of the PVT collector is shown in figure 7. Increasing the flow rate provides lower operating conditions for the PV cells and therefore, improves their electrical efficiency [39]. The results deduced that electrical efficiency increased by 0.45 % when the flow rate raised from 30 to 90 L/h and the highest electrical efficiency was found to be 13.6 %.

Since thermal and electrical efficiencies are products of outlet temperature, surface temperature, solar irradiance and maximum power of the PV collector which were

Table 4. Results of uncertainty analysis of the measured parameters.

Parameter	Unit	Average	Uncertainty
Surface temperature	°C	46.7	1.45
Outlet temperature	°C	35.8	1.12
Inlet temperature	°C	27.0	0.44
Solar irradiance	W/m ²	1078	32
Maximum power of PV	W	130.1	2.02
Thermal efficiency	-	47.48	1.43
Electrical efficiency	-	13.36	0.13

directly measured during the experiments, uncertainty analysis was performed are as follows [40]:

$$U_{\zeta_{el}} = \sqrt{\left(\frac{\partial \zeta_{el}}{\partial P_m}\right)^2 U_{P_m}^2 + \left(\left(\frac{\partial \zeta_{el}}{\partial I_t}\right)^2 U_{I_t}^2\right)} \quad (11)$$

$$U_{\zeta_{th}} = \sqrt{\left(\frac{\partial \zeta_{th}}{\partial T_o}\right)^2 U_{T_o}^2 + \left(\left(\frac{\partial \zeta_{th}}{\partial T_i}\right)^2 U_{T_i}^2\right) + \left(\left(\frac{\partial \zeta_{th}}{\partial I_t}\right)^2 U_{I_t}^2\right)} \quad (12)$$

Figure 8. Temperature profile of the module surface for: (OF-ST) oscillating flow square tube design, (OF-RT) oscillating flow round tube design, (DF-ST) direct flow square tube design, (DF-RT) direct flow round tube design, (SF-ST) spiral flow square tube design, (SF-RT) spiral flow round tube design, (CBF) cross finned box, (DF-BT) direct flow box-type design.

Figure 9. Average temperature of the module surface for: OF-ST) oscillating flow square tube design, OF-RT) oscillating flow round tube design, DF-ST) direct flow square tube design, DF-RT) direct flow round tube design, SF-ST) spiral flow square tube design, SF-RT) spiral flow round tube design, CBF) cross finned box, DF-BT) direct flow box-type design.

where the uncertainty of each parameter was calculated by the following expression.

$$U_F = \sqrt{\frac{\sum_{j=1}^m (F_j - \bar{F})^2}{m(m-1)}} \quad (13)$$

where $F = \{P_m, I_r, T_o, T_i\}$, $\bar{F}$ is the average value of the parameter and m is the number of measurements.

Total thermal and electrical efficiencies and results of the uncertainty analysis of the measured parameters are listed in table 4.

3.2 Comparison of different collector designs

Simulation results of temperature distribution on the module surface in the different heat exchangers at the flow rate

Figure 10. Average outlet Temperature of the module surface for: OF-ST) oscillating flow square tube design, OF-RT) oscillating flow round tube design, DF-ST) direct flow square tube design, DF-RT) direct flow round tube design, SF-ST) spiral flow square tube design, SF-RT) spiral flow round tube design, CBF) cross finned box, DF-BT) direct flow box-type design.

of 30 L/h and solar intensity of 1000 W/m² are shown in figure 8. As mentioned before (section 2), these results were obtained at the inlet and ambient temperatures of 25°C. Clearly, the most temperature gradient was observed with the direct flow heat exchangers, especially with those of tube-types. This comes from the fact that the water flow is not uniformly distributed between the parallel pipes in direct flow designs. Conversely, the spiral flow designs presented the least range of temperature variations. Meanwhile, square-tube heat exchangers provided slightly more

uniform temperature profile compared to the round-tube types. This was especially observable in the case of oscillating and direct flow designs. It is also clear from figure 8 that a hot zone surrounded by relatively cooler zones was generated on the module surface of the direct flow box-type heat exchanger and cross fins inside the box-type heat exchanger could dramatically decline the range of the surface temperature variations.

Figure 9 indicates average module surface temperatures for the eight collector designs at the different water flow

Table 5. Variations of thermal and electrical efficiencies in different collector designs: OF-ST) oscillating flow square tube design, OF-RT) oscillating flow round tube design, DF-ST) direct flow square tube design, DF-RT) direct flow round tube design, SF-ST) spiral flow square tube design, SF-RT) spiral flow round tube design, CBF) cross finned box, DF-BT) direct flow box-type design.

Collector design	Flow Rate (kg/h)	I = 600 W/m ²		I = 800 W/m ²		I = 1000 W/m ²	
		$\zeta_{el}(\%)$	$\zeta_{th}(\%)$	$\zeta_{el}(\%)$	$\zeta_{th}(\%)$	$\zeta_{el}(\%)$	$\zeta_{th}(\%)$
OF-ST	30	11.20	24.48	10.94	25.62	10.62	25.05
	90	11.55	30.75	11.37	29.46	11.20	29.28
	180	11.64	37.58	11.37	33.31	11.40	29.28
OF-RT	30	11.22	60.36	10.95	62.78	10.58	53.69
	90	11.54	58.00	11.35	61.50	11.21	64.42
	180	11.67	78.58	11.54	75.35	11.39	77.12
DF-ST	30	11.48	28.47	11.32	26.05	11.11	28.96
	90	11.61	37.58	11.48	34.59	11.30	34.16
	180	11.65	51.25	11.64	41.00	11.38	42.95
DF-RT	30	11.46	23.34	11.29	22.20	11.06	22.45
	90	11.58	30.75	11.46	30.75	11.29	30.26
	180	11.62	41.00	11.51	35.87	11.35	41.00
SF-ST	30	11.60	31.88	11.49	32.45	11.31	26.683
	90	11.79	44.41	11.74	42.28	11.66	36.119
	180	11.86	47.83	11.81	46.12	11.75	48.810
SF-RT	30	11.60	33.02	11.47	30.32	11.32	29.61
	90	11.80	39.29	11.75	32.031	11.64	40.02
	180	11.85	44.41	11.81	48.688	11.73	44.90
CBF	30	11.66	21.06	11.55	23.490	11.42	22.45
	90	11.79	35.87	11.71	34.594	11.61	40.02
	180	11.82	44.41	11.77	53.813	11.70	54.66
DF-BT	30	11.16	54.09	11.10	42.281	10.56	44.57
	90	11.33	63.20	11.14	66.875	10.76	65.13
	180	11.47	68.33	11.29	69.188	11.03	66.38

rates and solar intensities. The results showed that there was a difference between the average working temperatures of direct flow and the other designs of the PVT collectors. Since water flows with a significant lower velocity through the direct flow heat exchanger, temperature accumulation in this type was dramatically higher than the others. Similar results were also reported [19]. This difference is more observable at the lower water flow rates and it seems that increasing the flow rate was more effective on direct flow designs compared with the other types. Figure 9 also illustrated that the lowest surface temperature of the tube-type collectors was achieved with the spiral flow designs. The fins inside the box-type heat exchanger, beside extending heat transfer surface, provides cross channels through which water flows at a relatively higher velocity than the flat box type. This, especially at the higher solar radiation intensities, led to a significant reduction in working temperature of the PV module.

Outlet temperature of the different collector configurations was illustrated in figure 10. Despite similar surface temperature of direct flow tube type collectors, outlet temperature was considerably higher with the square tube type. This means that heat dissipation from the module to ambient when using direct flow round tube PVT collector is higher than that with the square tube. There was no

significant difference observed between outlet temperatures of the square and the round tubes in the other flow configurations. Although spiral flow designs had lower surface temperatures, their outlet temperatures were not higher than the other tube type collectors. Therefore, it can be said that the highest heat transfer to the ambient was occurred with the spiral flow design. Similarly, it can be concluded that the box type heat exchanger because of its relatively high outlet and surface temperature, possesses low heat losses. Similar results were reported [19].

Thermal and electrical efficiencies of the different PVT collector designs at various solar radiation intensities and water flow rates were given in table 5. The cross-finned box type and the spiral flow designs of the PVT collector had the highest electrical efficiencies at different test conditions. While, the maximum values of thermal efficiency were obtained with the direct flow square tube and the box type heat exchangers. Increasing the air flow rate from 30 to 180 L/h led to increase of 0.35% and 18.75% in average electrical and thermal efficiency, respectively. This is mainly due to improvement of heat transfer coefficient of the water flow inside the heat exchanger with raising the water velocity which enhances the heat absorption rate from the PV module. Generally, since intensifying the solar radiation led to increase in the cell temperature, the results

showed that average electrical efficiency decreased by 0.31% when solar intensity increased from 600 to 1000 W/m².

4. Conclusions

Different PVT collector designs were simulated using computational fluid dynamic method. Module surface and water outlet temperature profiles were predicted at the solar radiation intensities of 600, 800 and 1000 W/m², ambient and water inlet temperatures of 25°C and flow rates of 30, 90 and 180 L/h. To validate the simulation results, a set of experiments were conducted using a box-type PVT collector at different water flow rates. It was revealed from the experimental observation that increasing the flow rate from 30 to 90 L/h led to a reduction of 6 and 18% in surface and outlet temperatures, respectively. In addition, a maximum average thermal efficiency value of 56% was experimentally achieved at flow rate of 90 L/h. The simulation results indicated that compared with average surface temperature, computational results for the outlet temperature, especially at the lower flow rates, were closer to the measured data. The most temperature gradient was observed with the direct flow heat exchangers and the spiral flow designs presented the least range of temperature variations. Investigation of the temperature profiles indicated that a hot zone surrounded by relatively cooler zones was generated on the module surface of the direct flow box-type heat exchanger. The highest average surface temperature was observed with the direct flow designs and it was found that increasing the water flow rate reduces the difference between surface temperatures of direct flow with the other designs of PVT collectors. The highest heat transfer to the ambient was occurred with the spiral flow design and the box type heat exchanger presented a relatively low heat losses. The cross-fined box type and the spiral flow designs of the PVT collector had the highest electrical efficiencies at the different test conditions. The maximum values of thermal efficiency were obtained with the direct flow square tube and the direct flow box type heat exchangers. Average electrical efficiency of the PVT collectors decreased by 0.31% when solar intensity increased from 600 to 1000 W/m².

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