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Solar Panel Cooling and Water Heating with an Economical Model Using Thermosyphon

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Abstract

In the present work, experimental and theoretical study has been carried out to investigate the effect of using heat thermosyphon on the performance of cooling photovoltaic thermal solar panel. Three test rigs are constructed. The first system (module I) constructed from photovoltaic panel with 0.07 mm thickness cooper plate base, four thermosyphon heat pipes and water box heat exchanger with a capacity of 16.2 litter. The second system (module II), which was made for a cheaper economic model than module I, comprises similar photovoltaic panel with 0.07 mm thickness aluminum plate base, six copper heat pipes with the same dimensions for (module I) and water cylindrical heat exchanger with a capacity of 9.537 litter. The novel panels compared with the traditional panel. The experiments are carried out in July 2017, Baghdad. A MATLAB program is used to compute the models and establishing characteristic curves. The experimental thermal results proved that the novel methods are successful in cooling the solar panel, the module I is colder than the module II and the two modules are cooler than the traditional panel in a rate of (15-35) % for module I and (10-14) % for module II. The experimental electrical results showed that the efficiency of module I is improved by (11-14) % and module II improved by (4-8) % compared with traditional one. The comparison between the experimental and theoretical results revealed a good agreement with a small deviation of about (3-6) %.

Keywords: cooling photovoltaic panel, water heating, thermosyphon;

Nomenclature

A	Module area	θ	angle, degree
c	specific heat capacity, J/kg K	ρ	density, kg/m^3, reflectance
D	diameter, m	σ	Stefan Boltzman constant, W/m^2 K^4
E	output electricity, W/m^2	τ	Transmittance
G	solar radiation intensity, W/m^2	$(\tau\alpha)$	Transmittance absorptance product, -
h	heat transfer coefficient, W/m^2 K		**Subscripts**
k	thermal conductivity, W/m K	a	air, ambient
L	length, m	b	base panel
M	mass, kg	i	inner, differential node “i”
R	thermal resistance, K/W	j	differential node “j”
T	temperature, ^{o}C	l	Liquid
t	time, s	o	Outer
u	flow velocity, m/s	p	heat pipe
Ex	exergy, W/m^2	s	thermal insulating material
Pr	Prandtl number	v	Vapor
Nu	Nusselt number	w	water, wall of the heat pipe
Ra	Raleigh number	con	condenser section of heat pipe
Re	Reynolds number	eva	evaporator section of heat pipe
	Greek letters	pv	PV cell
α	Absorptivity	sky	Sky
γ	PV cell covering factor	TPT	black tedlar-polyester-tellar
δ	thickness, m		
ε	emissivity, second-low efficiency,-		
η	Efficiency		

1. Introduction

Photovoltaic is the most useful way of employing solar energy by directly converting it into electricity. Energy conversion devices, which are used to convert sunlight to electricity using the photoelectric effect are called solar cells [1]. There are two distinguished types of energy that can be produced: electrical energy and thermal energy, it leads to increase the photovoltaic temperature. The overall efficiency of photovoltaic cells drops radically with an increase in temperature and the rate of decrease ranges from 0.25% to 0.5% per degree Celsius, depending on the cell material used [2]. Akbarzadeh and Wadowski [3] introduced a passive method based on thermosiphons

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which can effectively cool the Photovoltaic cells under concentrated light. Incorporating a thermosiphon cooling system for the photovoltaic cells has been manufactured and successfully tested. Tonui et al. [4] studied the photovoltaic/thermal (PV/T) panel with heat extraction by forced or natural air circulation, prepare a non-expensive and simple method of photovoltaic panel cooling and the solar preheated air could be used in manufactured, industrial and agricultural section. The paper presents the use of a suspended thin flat metallic sheet at the middle or fins at the back wall of an air duct as heat transfer augmentations in an air-cooled photovoltaic/thermal (PV/T) solar collector to improve its overall performance. William et al. [5] wrote a research about used heat pipes to cool the concentrating photovoltaic systems, this work demonstrated the feasibility of a heat pipe cooling solution for concentrating photovoltaic cells. Heat pipes can be used to passively remove the heat, accepting a high heat flux at the concentrating photovoltaic cell, and rejecting the heat to fins by natural convection. Tang et al. [6] introduced a new method by using the micro heat pipe arrangement to cooling photovoltaic panel. The experimentally implemented study used air or water to cool the solar panel, the solar panel temperature can decrease and increase the photoelectric conversion efficiency. The temperature decreases by 4.7 ^{o}C and output power increases by 8.4%, for air-cooling compared with ordinary solar panel, and the temperature decreases by 8 ^{o}C and output power increases by 13.9 % for water-cooling. Mutombo [7] presented study about the behavior of thermosyphon hybrid photovoltaic thermal panel when exposed to differences of environmental parameters and to prove the advantage of cooling photovoltaic modules using a rectangular channel shape with water. The simulation results showed that the overall efficiency of the PV/T module was 38.7% against 14.6% for a standard PV module while the water temperature in the storage tank reached 37.1 ^{o}C. During summer in South Africa, this is a great reassurance to the marketing of the hybrid photovoltaic thermal technology.

This research introduces a novel technique to increase the conduction heat transfer by using a copper and aluminum plates to increase the thermal conduction surface area from the panel to the heat pipe. Thus, the present work is concerned with carrying out experimental study and mathematical verification to study the performance of photovoltaic panel by using heat pipe as a new technique to increase the conduction heat transfer by using a copper plate for module I and aluminum plate for module II to increase the thermal conduction surface area from the panel to the heat pipe. Also, the study is coupled with electrical and thermal model for calculating various parameters related to the performance of photovoltaic cooling by heat pipe system and through solving equations of the problem numerically for all parts and determining PV model parameters.

2. Heat Pipe Photovoltaic Modules HP-PV/T

There are three photovoltaic panels used in the experimental work (two modified solar module panels and a traditional panel to compare with), there were made of monocrystalline solar module 80(72) M1240×541. The specifications of photovoltaic panels are; peak power (Pmax) which is 80 (Watt), voltage at maximum power (Vmp) of 33.3 (V), current at maximum power (Imp) 2.4 (A), open circuit voltage (Voc) of 41.5 (V) and short circuit current (Isc) of 2.6 (A), these are provided by the manufacturer for the reference conditions of 1000 W/m^2 of irradiance level, 25 ^{o}C of cell temperature, total number of cells are 68. Figure (1) showed the experimental setup for modules (I and II), which has been designed and manufactured in this work.

Figure 1. Experimental setup system.

2.1. Module I

The test system consists of four copper thermosyphon heat pipe, 1200 mm evaporator length with filing ratio 55% distil water as a working fluid, 14 mm inner diameter and 16 mm outer diameter, fixed on the back surface of the PV module. A copper plate of 0.07 mm thickness covered heat pipes and panel from the back, the new technique was done by envelope around the heat pipe to increase the contact surface, Fig. (2). 50 mm glass wool insulated the system from the back of the panel. 150 mm condenser length with 28 mm inner diameter and 30 mm outer diameter, immersed at (540×150×300) mm^3 water box. The space between the two adjacent heat pipes were measured to be approximately 140 mm. Schematic diagram of the experimental rig with thermocouples location is shown in Fig. (3).

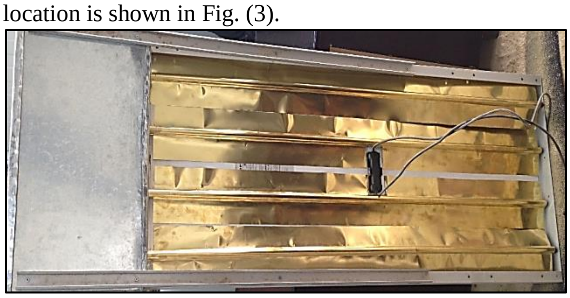

Figure 2. copper plate with heat pipe in the back of panel.

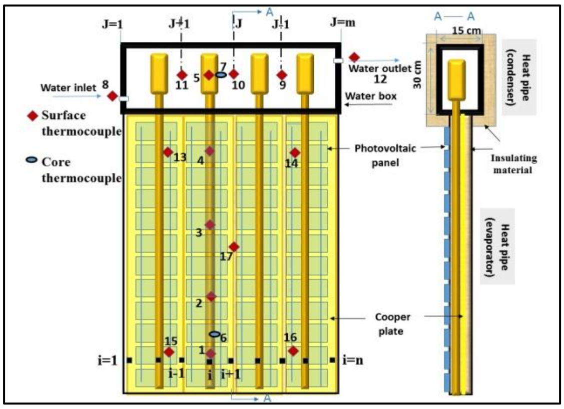

Figure 3. Schematic diagram of module (II) and thermocouples location.

2.2. Module II

The test system (I) is expensive (172 $) because of the use of expensive parts that made from pure copper 90-95%. Therefore, a second low cost and economic (39 $) test system (II) was used and placed next to system (I) to study the performance of systems and compare between them. This cheap and economic Photovoltaic module II was used for the first time which consists of six thermosyphon heat pipes with the same dimension in module I, aluminum plate instead of copper plate, heat exchanger is made of plastic cylindrical pipe of 3 mm thickness with 146 mm inner diameter and 540 mm length, water tank, storage tank and stand. Schematic diagram of the experimental rig with thermocouples location is shown in Fig. (4).

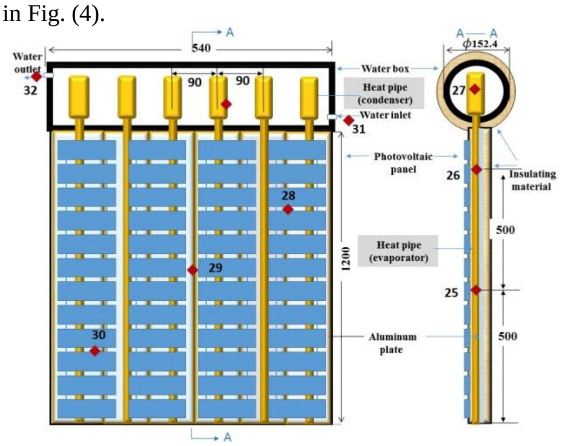

Figure 4. Schematic diagram of module (II) and thermocouples location.

The most previous studies tested the performance of the photovoltaic without a load, wherefore in this work, the performance of the photovoltaic with load is studied and, it showed the load effects on the photovoltaic behavior. Because PV panel produces electricity and warm water at the same time, the load will be a water electrical heater (DC 40 Watt) that receives the power from the PV panel. So, it can produce hot water instead of warm water to use it in the houses and industrial applications.

3. Thermal analyses

In the present study, a passive technique model was developed for an HP-PV/T system. The mathematical model consists of five main equation sets as follows [8]:

1. Heat-balance equation of the PV module.
2. Uni-dimensional heat conduction of the base panel (aluminum plate).
3. Heat-balance equation of the heat pipe.
4. Heat-balance equation for water in the heat exchanger.
5. Heat-balance equation for water in the storage tank.

The following assumptions were made in the model to simplify the calculation:

1. Heat conduction in the longitudinal direction of the aluminum plate was neglected.
2. The temperatures of the adhesive layer (EVA and TPT) and PV cells in the same direction were considered equal.
3. The heat capacity of the adhesive layer (EVA and TPT) was neglected.
4. Heat loss from the heat pipe condenser to the ambient was neglected.

Figure (7) depicts the section of the HP-PV/T solar collector.

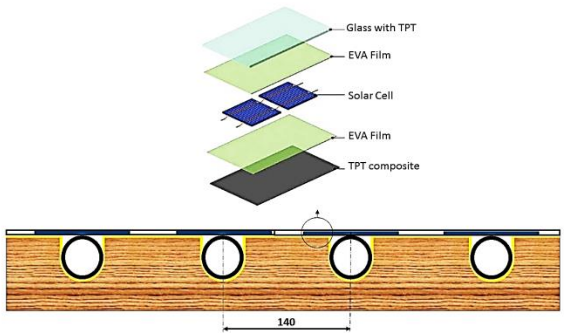

Figure 5. Section of the HP-PV/T solar collector.

Based on assumptions described above, for the photovoltaic layer, which includes the PV cells, EVA and TPT, the heat-balance equation is given by [8]:

$$\gamma\delta_{pv}\rho_{pv}C_{pv}\frac{\partial T_{pv}}{\partial t} = h_a(T_a - T_{pv}) + h_{sky,pv}(T_{sky} - T_{pv}) + {(T_b - T_{pv})}\big/{R_{b,pv}} + G(\tau\alpha)_{pv} - E_{pv} \quad (1)$$

Where, h_a and $h_{sky,pv}$ are convective and radiant heat transfer coefficients, respectively, between the PV and surroundings.

T_{sky} is the sky temperature with:

$$T_{sky} \equiv T_a \quad (2)$$

$$h_a = 2.8 + 3.0u_a \quad (3)$$

$$h_{sky.pv} = \varepsilon_{pv}\sigma\left(T_{sky}^2 + T_{pv}^2\right)\left(T_{sky} + T_{pv}\right) \quad (4)$$

γ is PV cell coverage ratio [9,10], and $\gamma = {A_{pv}}\big/{A_c}$ (5)

Where, σ is the Stefan-Boltzmann constant (5.6697x10^{-8} W/m^2 K^4) and $R_{b.pv}$ is the thermal resistance between the PV layer and base panel (copper plate) expressed as [9, 11]:

$$R_{b.pv} = {\delta_{ad}}\big/{K_{ad}} \quad (6)$$

E_{pv} is given by the instantaneous PV efficiency (η_{pv}) expressed as:

$$E_{pv} = G(\tau\alpha)_{pv}\tau_{ad}\eta_{ref}\left(1 - B_{ref}\left(T_{pv} - T_{ref}\right)\right) \quad (7)$$

Where, η_{ref} is the reference cell efficiency at the reference operating temperature, $T_{ref} = 25$ °C.

B_{ref} is the temperature coefficient, $B_{ref} = 0.0045$ °C^{-1}.

τ_{ad} is the transmittance of the adhesive layer.

$(\tau\alpha)_{pv}$ is the effective absorptance and is given as:

$$(\tau\alpha)_{pv} = \frac{\tau\alpha}{1 - (1 - \alpha)\rho_d} \quad (8)$$

α is the effective absorptance of PV/T plate given as:

$$\alpha = \gamma\alpha_{pv} + (1 - \gamma)\alpha_{TPT} \quad (9)$$

ρ_d is the reflectance of inner cover for diffuse radiation and is given as [9, 11 and 12]:

$$\rho_d = 1 - \alpha_{ad} - \tau_{ad} \quad (10)$$

The base panel divided the differential grid, as shown in Fig. (6). The two types of grid are labeled, where one grid is connected to a heat pipe node and the other is not (middle node). The heat-conduction equations in these two types of grid are different and are given by Eqs. (7) and (8), respectively [9, 11 and 12].

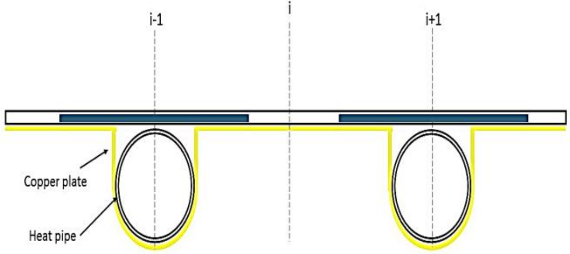

Figure 6. Differential grid partition of the base panel.

The expressed of heat-pipe node is:

$$\rho_b c_b \frac{\partial T_b}{\partial t} = k_b \frac{\partial^2 T_b}{\partial x^2} + \frac{1}{\delta_b}\left[(T_a - T_b)/_{R_{b.a}} + (T_{pv} - T_b)/_{R_{b.pv}} + (T_{p.eva} - T_b)/_{(R_{p.b})} \right] \quad (9)$$

$$\rho_b c_b \frac{\partial T_b}{\partial t} = k_b \frac{\partial^2 T_b}{\partial x^2} + \frac{1}{\delta_b}\left[(T_a - T_b)/_{R_{b.a}} + (T_{pv} - T_b)/_{R_{b.pv}} \right] \quad (10)$$

Were $R_{b.a}$ is the thermal resistance between the base panel and the ambient air, given by:

$$R_{b.a} = {}^{\delta_s}/_{k_s} + {}^{1}/_{h_a} \quad (11)$$

Where, δ_{pb} and A_{pb} are the thickness between the base panel and evaporator section of heat pipe and contact area, respectively.

For the heat pipe, the heat-balance equations were provided for the evaporator and condenser sections, respectively. Heat transfer from the evaporator section to the condenser section was calculated using total thermal resistance $R_{eva.con}$.

Having known that the pressure decrease that is caused by vapor flow along the axial length of the heat pipe is very small, the vapor space is assumed to operate at a constant saturation pressure. Therefore, the temperature gradient of the working fluid along the axial length of the heat pipe can be neglected.

The value for $R_{eva.con}$ can be derived based on the following parts:

- $R_{eva.p}$, thermal resistances across the thickness of the container wall and thickness.
- $R_{eva.wick}$, thermal resistance across the wick thickness.
- $R_{eva.i}$, thermal resistance that occurs at the vapour–liquid interfaces in the evaporator.
- $R_{con.i}$, thermal resistance associated with the condensing process.
- $R_{con.p}$, thermal resistance associated with the conduction process through the pipe wall.
- That's will be:

$$\sum R_{eva.con} = R_{eva.p} + R_{eva.wick} + R_{eva.i} + R_{con.i} + R_{con.p} \quad (12)$$

$$R_{eva.p} = \frac{\ln(d_o/d_i)}{2\pi k_p L_{eva}} \quad (13)$$

$$R_{eva.wick} = \frac{ln(d_{o.wick}/d_{i.wick})}{2\pi k_{wick} L_{eva}} \quad (14)$$

$$R_{eva.i} = \frac{2}{h_{eva.i}\pi d_i L_{eva}} \quad (15)$$

$$h_{eva.i} = \frac{k_l}{t_{wick}}$$

k_l and t_{wick}, the thermal conductivity of the fluid and the wick thickness, respectively.

$$R_{con.p} = \frac{\ln(d_o/d_i)}{2\pi k_p L_{con}} \quad (16)$$

$$R_{con.i} = \frac{1}{h_{con.i}\pi d_i L_{con}} \quad (17)$$

Where $h_{con.i}$ is the condensing film coefficient that may be obtained from the Nusselt analysis for film wise condensation as [7 and 8]:

$$h_{con.i} = 1.13\left[\frac{g\, sin\theta \cdot \rho_l(\rho_l - \rho_v)k_l^3 h_{fg}}{\mu_l \Delta T_{cr} L_{con}}\right]^{1/4} \quad (18)$$

Because of using the wickless heat pipe, the $R_{eva.wick}$ and $R_{eva.i}$ were neglected.

Therefore, $\Sigma R_{eva.con}$ becomes:

$$\Sigma R_{eva.con} = R_{eva.p} + R_{con.i} + R_{con.p} \quad (19)$$

The heat-balance equation for the evaporator section, *is expressed as:*

$$M_{p.eva} c_p \frac{\partial T_{p.eva}}{\partial t} = \left(T_{p.con} - T_{p.eva}\right)/R_{eva.con} + \left(T_b - T_{p.eva}\right)/R_{p.b} \quad (20)$$

and for the condenser section as:

$$M_{p.con} c_p \frac{\partial T_{p.con}}{\partial t} = \left(T_{p.eva} - T_{p.con}\right)/R_{eva.con} + A_w h_{w.con}\left(T_w - T_{p.con}\right) \quad (21)$$

$h_{w.con}$, is the convection heat transfer coefficient between the heat pipe condenser and water.

$$h_{w.con} = Nu\frac{k_w}{D_{con.o}} \quad (22)$$

$$Nu = CRe^m Pr^n \quad (23)$$

Values of , m and n depend on the Reynold's number [7 and 13].

In the water box, the differential grid partition for water is shown in Fig. (7). In addition, the upwind scheme is used in the water differential equation, and for grid (j), the equation can be expressed as:

$$m_w c_w \frac{\partial T_{wj}}{\partial t} + \dot{m}_w c_w (T_{w.j} - T_{w.j-1}) = {(T_a - T_{w.j})}/{R_{a.w}} + A_w h_{w.con}(T_{p.con} - T_{w.j}) \quad (24)$$

Where, m_w is the mass of the water in a single control volume.

$\dot{m}_w$ is the mass flow rate of the water, $\dot{m}_w = \rho_w u_w A$.

$R_{a.w}$ is the equivalent thermal resistance between water and ambient air.

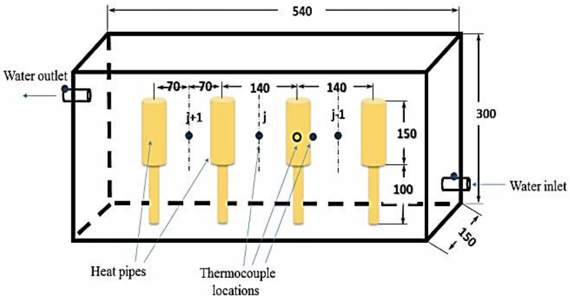

Figure 7. Differential grid partition of water in the water box.

The heat balance equation for the water in the storage tank is given by [8 and 11]:

$$M_{w.tank} c_w \frac{\partial T_{w.t}}{\partial t} = (T_a - T_{w.t})/R_{a.wt} + n \cdot \dot{m}_w c_w (T_{w.out} - T_{w.in}) \quad (25)$$

Where, $M_{w.tank}$ is the mass of the water in the storage tank.

$R_{a.wt}$ is the equivalent thermal resistance between the water and the ambient air.

$T_{w.in}$ and $T_{w.out}$ are the inlet and outlet water temperatures, respectively, of solar collector.

n is the number of solar panel.

The instantaneous useful heat gain of the system is given by:

$$Q_w = M_w c_w (T_{w,out} - T_{w,in}) \quad (26)$$

The instantaneous useful heat gain of the system is expressed as

$$Q_w = M_{w,tank} c_w (T^1_{w,t} - T^0_{w,t}) \quad (27)$$

The total efficiency of the HP-PV/T system can be described by an equation based on the first-law of thermodynamics (energy efficiency) and is introduced as follows [14]:

$$\eta_{pvt} = \frac{Total\ thermal\ energy + total\ electrical\ energy}{Total\ radiation\ over\ the\ PV/T}$$

$$\eta_{pvt} = \frac{\int_{t_1}^{t_2} (Q_w + A_c E_{pv}) dt}{\int_{t_1}^{t_2} A_c G dt} = \eta_w + \gamma \eta_{pv} \quad (27)$$

A MATLAB 2016 was used to solve thermal equations and establishing characteristic curves.

4. Experimental Results

The experimental results taken on 18 and 21 July-2017 were about the average temperature, temperature on the modules parts and the electrical characteristics which were obtained by the multi-channel thermometer, solar power meter and solar module analyzer device.

4.1. Average Temperature

Several factors influenced the solar panel efficiency. These factors are: direction and intensity of solar radiation, angle of inclination, in addition to the ambient temperature. Direction and angle are fixed to the south and at 45^o, therefor, the influencing factors will be the solar radiation and temperature on which the photovoltaic depends on the electricity generation.

On 18/7, the test was done on both models with constant water flow rate ($\dot{m}$=10 l/h), and the average temperature on the two modules panel at 12:00 is 68.68 ^{o}C for module I, 76.7 ^{o}C for module II and for the traditional panel is 85.36 ^{o}C with ambient temperatures is 48.7 ^{o}C. This indicates that the two modules operate at hot weather effectively and their average temperatures are less than that for the traditional panel in the rate of (35.7, 30.58) % for module I and (12.76, 12.3) % for module II, respectively, Fig. (8).

On 21/7, the flow rate is increased to 15 l/h, and the average temperature on the two modules panel at 12:00 is 64.06 ^{o}C for module I, 75.53 ^{o}C for module II and for the traditional panel is 86 ^{o}C in the rate of 34.2 % for module I and 13.86 % for module II with ambient temperatures 47.2 ^{o}C. The average temperature for the modules increases with the increase in the ambient temperature and radiation, Fig. (8). The increasing of the flow rate to 15 l/h did not have a significant effect, but the effect of ambient temperature is noticeable, indicating that the amount of heat withdrawn from the condenser needs a lower flow rate.

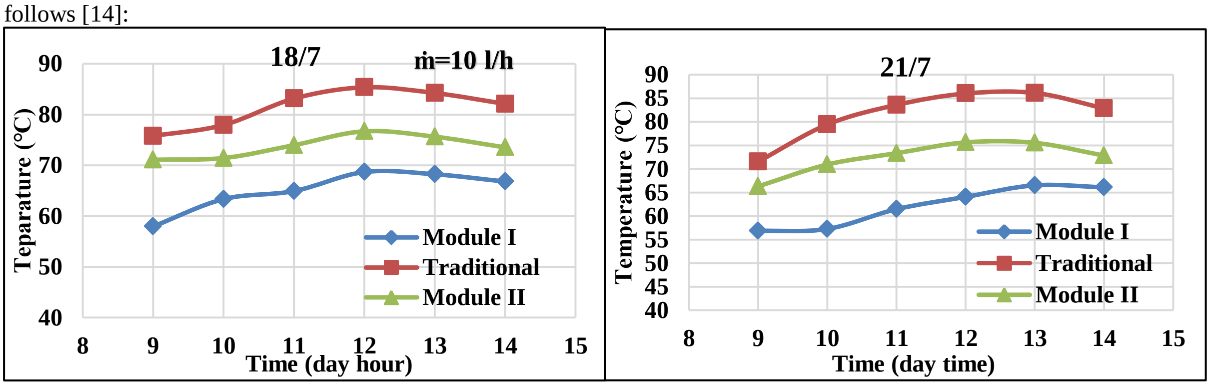

Figure 8. Average temperature.

4.2. Characteristics of Panel

The most important part in this study is the characteristics of the panel, which is the result of the study about the short and max current, the open and max voltage, the max power and the efficiency of the solar panel obtained by the solar module analyzer device. On 18/7, at 12:00 the open voltage is 33.84 V, short current is 2.1453 Ampere, max power is 50.5623 Watt, max voltage is 26.384 Volt, max current is 1.9164 Ampere and the efficiency is 16.8372 % for the module I. The open voltage is 33.493 V, short current is 2.148 Ampere, max power is 50.57426 Watt, max voltage is 26.96 Volt, max current is 1.8759 Ampere and the efficiency is 16.84 % for the module II, while for traditional panel: the open voltage is 32.366 Volt, short current is 2.0571 Ampere, max power is 45.4297 Watt, max voltage is 24.908 Volt, max current is 1.8239 Ampere and the efficiency is 15.128 %. with average temperature is 64.06 °C for module I, 76.7 °C for module II and 85.36 °C for traditional panel. The overall characteristics for module I and module II are close and better than that for the traditional panel with photovoltaic temperature decrease, as shown in 18/7 the 21/7 as well, Fig. (9).

5. Theoretical Results

To solve the four-parameter model, a MATLAB computer program is used to evaluate the characteristics of photovoltaic panel. The temperature of panels and solar radiation intensity are adopted from the experimental data. Figure (10) shows that the theoretical results are very closer between the modules like experimental results. Table (1) shows the difference between the experimental and theoretical photovoltaic characteristics results. Theoretical efficiency of module I and module II are less than experimental in a rate of 12.7% and 15.6% respectively for 18/7 and 4% and 10% respectively for 21/7. Solar panel is influenced by several external factors, such as dust, wind, humidity, and interior factors such as multicellularity which led to a difference between the experimental and theoretical results.

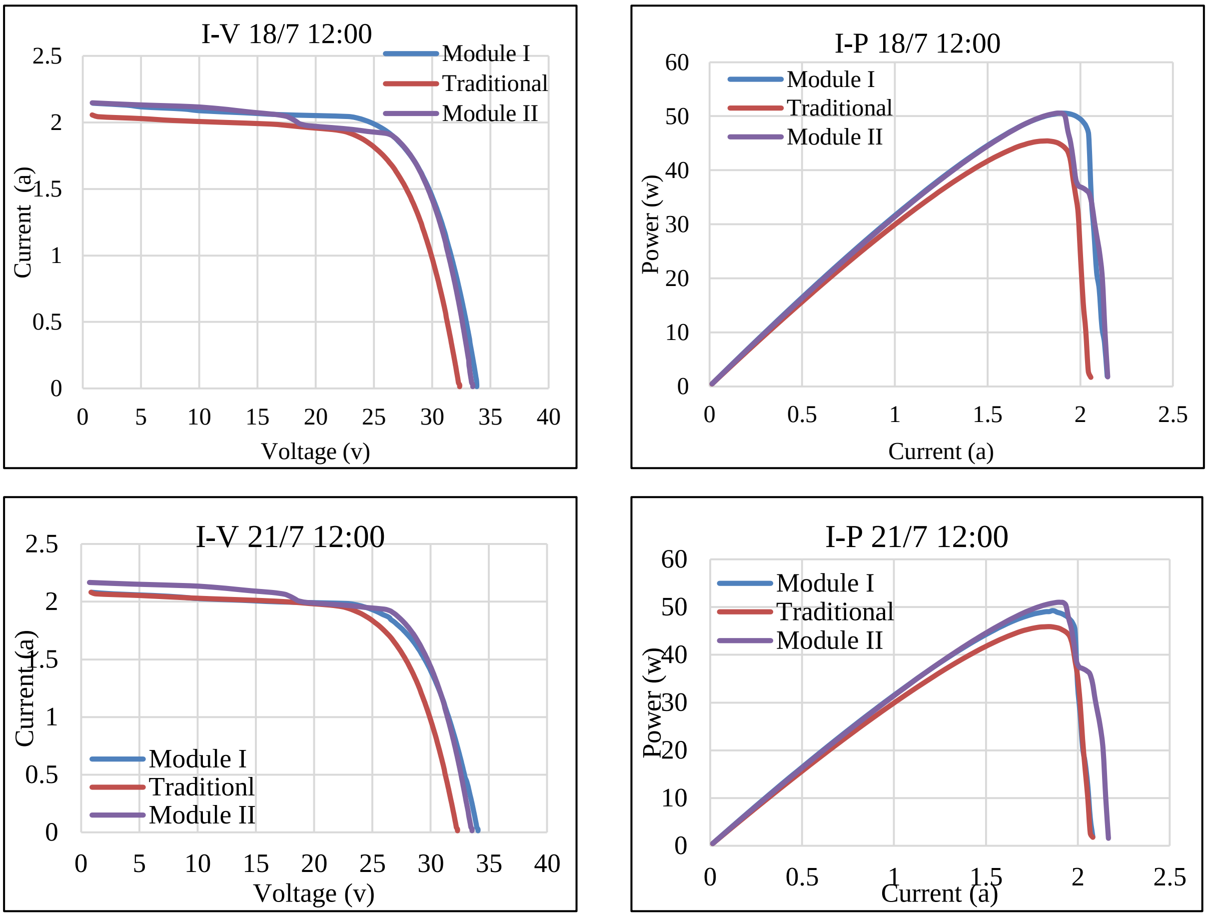

Figure 9. Photovoltaic characteristics.

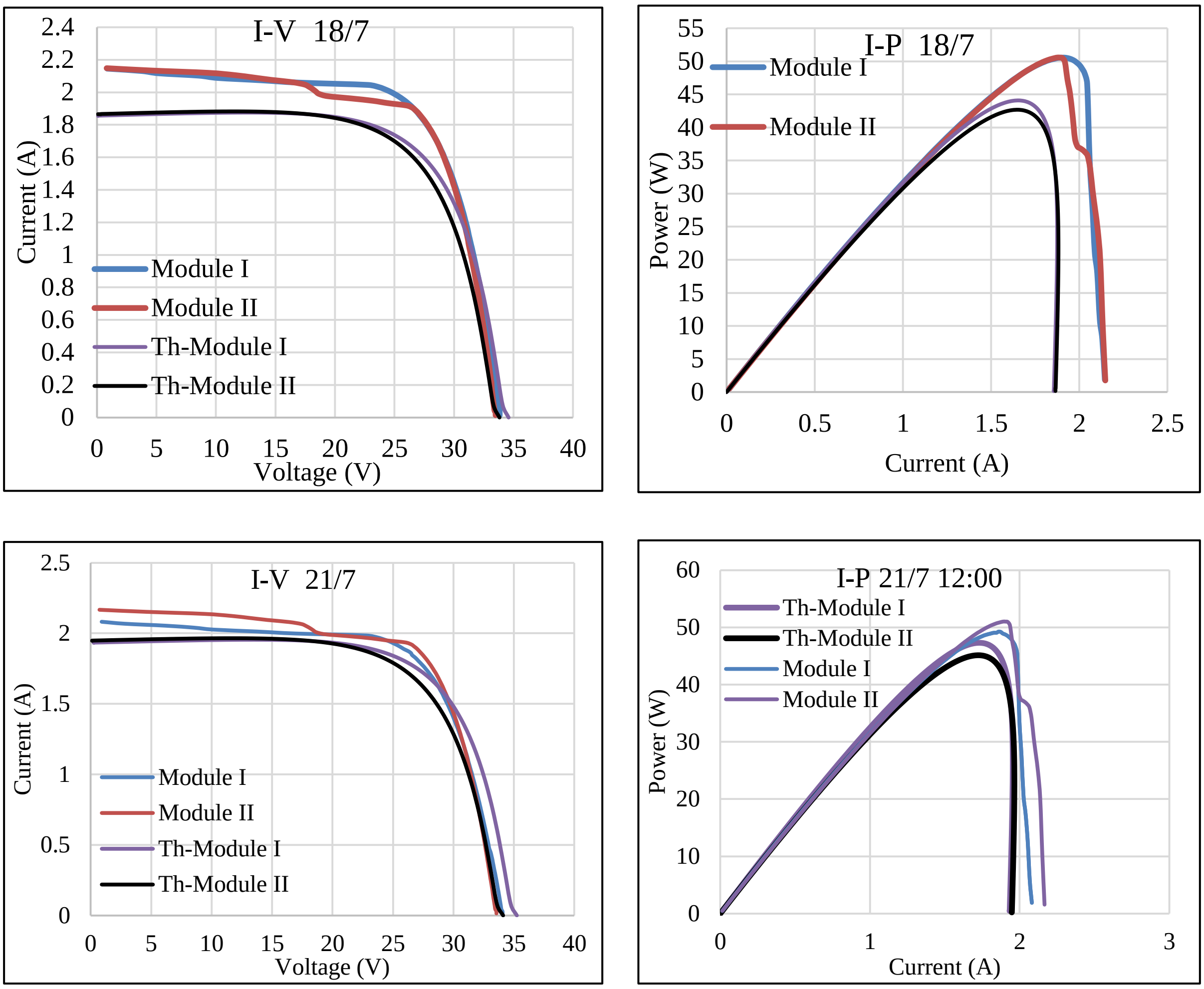

Figure 10. (I-V) and (I-P) for 18-21/7/2017 at 12:00 with radiation (724.7 W/m^2).

Table1. Experimental and theoretical characteristics of PV for 18, 21/7/2017 at 12:00.

Property 18/7/2017	Module I	Theoretical I	Module II	Theoretical II	Traditional
Temperature ℃	64.06	64.06	76.7	76.7	85.36
Vopen V	33.84	34.5867	33.493	33.82	32.366
Ishort A	2.1453	1.927	2.148	1.865	2.0571
Pmax W	50.5623	44.08986	50.57426	42.68206	45.4297
Vmaxp V	25.26.384	26.58674	26.96	25.82052	24.908
Imaxp A	1.9164	1.658341	1.8759	1.653.29	1.8239
EFF%	16.8372	14.685	16.84	14.212	15.128
Property 21/7/2017	Module I	Theoretical I	Module II	Theoretical II	Traditional
Temperature ℃	64.06	64.06	75.63	75.63	86
Vopen V	34.062	35.24228	33.553	34.10882	32.303
Ishort A	2.0818	1.932858	2.1667	1.947297	2.0818
Pmax W	49.2523	47.29924	51.0137	45.12669	45.9124
Vmaxp V	26.484	27.24228	26.96	26.10882	24.874
Imaxp A	1.8597	1.736243	1.8922	1.728408	1.8458
EFF%	16.4	15.75	16.983	15.3	15.294

6. Conclusions

The present research done to improve the performance of photovoltaic panel by cooling it by using thermosyphon heat pipe, the experiments carried out at different intervals proved the success of this method in reducing the temperature of the solar panel compared to the traditional panel, which improved the characteristics of the panel and the resulting in higher capacity and efficiency. Average temperature for module I is between 55-65 ^{o}C, module II 72-76 ^{o}C and for traditional panel is 70-more than 80 ^{o}C in July. Module I temperature is less than the module II and the two modules are less than the traditional panel in a rate of (15-30) % for module I and (10-14) % for module II.

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