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1 **Novel Solar PV/Thermal Collector Design for the Enhancement of Thermal**
2 **and Electrical Performances**

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35 **ABSTRACT**

36 The main objective of this study is to develop a novel photovoltaic thermal collector (PVT) to improve
37 the electrical and thermal efficiencies of the solar collector. The goal is to maximize the electrical power
38 and minimize the thermal losses of the solar panel. A novel photovoltaic thermal collector is designed and
39 tested. The new PVT collector includes: (1) An optical anti-reflective and low-emissivity coating to
40 reduce the radiation losses; (2) A thermal resistance to reduce the conduction losses between the
41 photovoltaic and absorber plate; and (3) A channel heat exchanger to decrease the thermal losses between
42 the solar cells and the cooling fluid. A transient two-dimension multi-physics model for the PVT sheet-
43 tube and the advanced PVT collector is developed. The state variable variations are predicted by the finite
44 volume method. A comparison between the two considered hybrid collectors in terms of thermal and
45 electrical efficiencies and temperature distribution is performed. Moreover, the impact of arrangement
46 (anti-reflective and low-emissivity coating, thermal resistance between the absorber plate and the cooling
47 fluid, enhanced exchange surface area between the flat plat exchanger and the cooling fluid) on the new
48 PVT collector is studied and analyzed. The simulation results showed clearly the advantages of using this
49 evolution of the PVT collector compared to the basic one. Indeed, this new PVT configuration represents
50 a series of improvements that lead to a lower PV module and higher fluid operating temperatures. Higher
51 electrical and thermal efficiencies for the proposed PVT (15.4%, 73%) are obtained compared to the basic
52 PVT collector (13.7%, 58%), respectively under no loss and standard test conditions.

53

54 **Keywords:** *Hybrid solar PVT, Back Cooling, Electrical Efficiency, Thermal Efficiency, Novel Collector,*
55 *Channel Heat Exchanger.*

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Nomenclature

A	Surface area (m ²)
C_p	Specific heat (J/Kg.K)
E	Electrical power output (W)
K	Thermal conductivity (W/m.K)
G	Solar irradiation (W/m ²)
h	Heat transfer coefficient (W/m.K)
Nu	Nusselt number
Ra	Rayleigh number
Pr	Prandtl number
Pe	Perimeter (m)
T	Time (s)
\dot{m}	Mass flow rate (Kg/s)

Greek

β	PV cell temperature coefficient (K ⁻¹)
ρ	Density (Kg/m ³)
δ	Thickness (m)
ε	Emissivity
η	Efficiency (%)
τ	Solar transmission

Subscripts

	Amb	Ambient
	Ad	Adhesive
	Env	Environnement
	Con	Convection
	Cond	Conduction
	C	Collector
	Elec	Electrical
	is	Insulation
	in	Inlet
	PV	Photovoltaic module
	Pab	Absorber plate
	Pabu	Upper half of absorber
	Pabh	Lower half of absorber
	Out	Outlet
	Th	Thermal
68	wi	Wind
69	<i>RMS</i>	mean root square deviation
70	FVM	finite volume method
71		
72		
73		

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75 **1. INTRODUCTION**

76

77 Solar thermal collectors and photovoltaic panels are recognized as favourable solutions for collecting and
78 transforming solar energy to useful energy due to the abundance and inexhaustibility of the solar resource.
79 A solar thermal (ST) collector is a device that converts solar radiation energy into heat. Due to the photo-
80 conversion effect, the photovoltaic module is converting a part of sunlight into electrical energy. Around
81 80% of the solar radiation captured by the PV module is not converted, it consequently causes elevation
82 of the operating temperature, which induces a decrease of their efficiency and ageing. Indeed, the PV
83 cells that operate under strong sunlight and high temperature (i.e. in tropical or desert regions or in
84 summer seasons in temperate areas) generally suffer from more efficiency loss [1]. The thermal and
85 electrical cogeneration combined in a photovoltaic thermal collector is an effective way to reduce the loss
86 of collected solar radiation through heat dissipation and to control the PV module temperature. The
87 temperature regulation of photovoltaic panels can be achieved by several methods such as air-cooling,
88 water-cooling, use of heat pipe, phase change materials and thermoelectric cooling. For a configuration of
89 air forced cooling the back side temperature of a PV module is higher than at the front side. Garg et al. [2]
90 studied the influence of cooling the photovoltaic module using natural or forced air. The maximum
91 photovoltaic module obtained is around 49 ° C for the natural circulation mode against 45 ° C found by
92 the forced circulation mode. The water has obviously a better ability to transport heat than air. As a
93 classical configuration, the water flows through a heat exchanger which it placed at the rear surface of the
94 PV module. Several configurations can be found mostly depending on thermal contact between PV and
95 Thermal functions. Such PVT could be uncovered or covered with a glass placed on the front of the
96 collector. A glass cover reduces the thermal losses of the collector but increases the optical losses
97 (reflection, absorption) and consequently reduces the electrical efficiency of the photovoltaic module. A
98 conventional the PVT sheet and tube glazed collector provides a lowest electrical and thermal
99 performance for the same area with respect to the separate conventional PV and Thermal panels. A
100 comparative investigation between a PVT with sheet and tube absorber, a standard photovoltaic module
101 and a traditional ST collector with a sheet and tube absorber, has been performed by Rejeb et al. [3].
102 43.51% and 65.31% thermal efficiencies were reported, for the covered PVT collector and the covered ST
103 collector both with sheet and tube absorber, respectively. Bergene et al. [4] developed a mathematical
104 model to study the behaviour of a solar photovoltaic thermal sheet and tube collector. They investigated

105 the effect of fin size, flow rate and the inlet fluid temperature on the electrical and thermal performance.
106 They noted that for optimized values of these parameters, the overall (thermal + electrical) efficiencies
107 can be reached at 80%. Taoufek et al. [5] carried out a numerical study and experimental validation of an
108 unglazed PVT with a galvanized steel sheet and tube absorber. They reported that the maximal thermal
109 useful heat and the electrical power generated by the PVT collector were 290 W and 48 W, respectively.
110 Their work was performed under the meteorological conditions of Ghardaia (latitude of 32.29°N;
111 longitude of 3.41°E) in Algeria. The solar radiation varied between 100– 1020 W/m² from 7:00AM to
112 19:0PM. Thus, several works focused on the improvement of the performance of PVT. The PVT sheet
113 and tube configuration has to date been the most extensively studied in literature [3-12], despite a
114 relatively low efficiency commonly arising from a poor thermal coupling between the photovoltaic
115 module and the thermal absorber. Furthermore, different works have been carried out to transform the
116 PVT sheet and tube configuration to a more developed PVT structure at lower cost. Sandnes et al. [13]
117 investigated a PVT collector with a polymer flat exchanger absorber. Their hybrid collector consists of c-
118 Si solar cells jointed to a polymer heat exchanger (polyphenylenoxide plastics) in order to improve the
119 heat transfer to the cooling fluid. Their results showed that the presence of solar cells on the top of the
120 absorber reduce the absorption about 10%, whereas the presence of an additional glass reduces the optical
121 efficiency by about 5%. A comparative investigation on the photovoltaic thermal collector with straight
122 absorber channels and was carried out by Salem et al. [14] for a 1.0 L/min flow rate, a 47.2% and 34.3%
123 value of thermal performance for straight channels and helical ones were obtained.

124 In addition, the integration of phase change materials to regulate their operating temperatures has
125 attracted the attention of several authors [15-18]. Yang et al. [15] have experimentally studied the
126 possibility of integrating phase change materials (PCMs) within a PVT collector. The collector consists of
127 an absorbing plate fixed at the rear of a photovoltaic module. Water tubes are welded to the absorber. A
128 layer of MCP is then placed at the rear of the tubes. A 8.16% and 6.98% value of electrical efficiency for
129 PVT with PCM and for PVT were obtained. Other studies have shown that heat transfer inside the PVT
130 collector can be improve by using a porous media [19-20]. Omer el al [19] studied experimentally the
131 performance of PVT using the porous media. Their experiments were performed under the meteorological
132 conditions of Kirkuk (latitude of 35.47°N; longitude of 44.40°E) in Iraq. The ambient temperature ranged
133 between 25-29 °C, while the solar radiation varied between 580– 1000 W/m² from 9:00AM to 16:0PM.
134 The results revealed that the maximum electrical power outputs with porous media is 98 W, while without
135 porous media was 92 W. Also it can be seen that the thermal useful heat power outputs with porous media
136 collector was 860 W, while without porous media was 710 W.

137 Other technics such as the combination of water and air as a heat transfer fluid (bi-fluid) in order to
138 extract the heat from the PV module can also lower the PV function operating temperature. A bi-fluid
139 PVT [21-26] collector was designed to generate the electrical power, heating air for buildings or drying
140 applications and for domestic hot water heating. A numerical and experimental study was carried out by
141 Jarimi et al. [21] in order to compared the electrical and thermal performances of PVT for
142 different cooling modes (air, water and combined air/water). They concluded that using bi-fluid provided
143 the best higher electrical and thermal outputs powers compared with the separate use of air or water.

144 The thermal resistance conduction between the PV cells and the heat absorber exchanger should be
145 minimized. Zondag [27] examined the effect of improving the thermal exchange through the thermal
146 conductance between the photovoltaic module and the absorber sheet varying between 40 W/m.K and 250
147 W/m.K. Its review paper showed that a small value leads to an increase of the temperature difference
148 between the PV module and the absorber (by 12°C) and therefore is reducing both, the thermal and
149 electrical efficiencies. Dupeyrat et al. [28] designed a high performance advanced PVT collector with
150 enhancement thermal conductance (700 W/m.K) between the PV module and thermal absorber (1.2 mm
151 roll bond aluminium) by laminating the PV cells to absorber directly using EVA resin (Ethylene Vinyl
152 Acetate). A maximum of 87% overall performance have been achieved.

153 Like the thermal resistance between the two functions of a PVT collector, the optical efficiency plays an
154 important role in determining the overall thermal and electrical performance. Zondag [27] noted that
155 adding the anti-reflective coating (ARC) to the transparent glazing and encapsulation area can be
156 favourable. Also, they suggest that the use of low emissivity coatings leads to enhancement of the thermal
157 efficiency and a decrease in the electrical performance. Gang [29] examined the impact of using a low
158 emissivity coating, in order to reduce the optical loss in the PVT heat pipe collector. A (270.3 W/m²,
159 178.9 W/m², 284 W/m²) on the heat gain and (34.8 W/m², 35.0 W/m², 36.5 W/ m²) on the electrical
160 power were found for solar selective coating, black pigment coating and no coating on the absorber plate,
161 respectively. Santbergen et al. [30] calculated the effect of a 300 nm SnO₂ coating on the electrical
162 performance of a PVT collector. A reduction of the emissivity from 0.8 to 0.2 was calculated with a
163 corresponding reduction of the electrical performance from 12.97 % to 11.84 %. For the developed low-
164 emissivity coating used by Lämmle et al. [31], a 60% enhancement of the thermal performance and a 3%
165 decrease of the electrical performance were observed.

166 In a methodology point of view, the majority of previous studies on heat transfer governing the PVT
167 collector performance have considered a steady state approach to assess behaviour model. Using the
168 modified Hottel-Whiller model developed by Florschuetz [32], the thermal efficiency can be presented by

169 the global thermal loss, mean fluid and ambient temperatures, solar radiation and heat removal factor.
170 This model neglects the thermal mass and thus provides accurate results only with data averaged over a
171 long period. Hence it is unsuitable for studies of thermal and electrical performance under rapidly varying
172 inlet water temperature, wind speed, ambient temperature, or solar radiation fluctuations. Similarly, it is
173 not suitable for achieving control criteria for instantaneous regulation systems. Chow [33] noted that the
174 operation of a PV/T collector is inherently dynamic: the variation of climatic conditions (solar irradiance,
175 wind speed) are transient in nature. A dynamical model is particularly useful in order to investigate in-use
176 behavioural thermal response and energy performances of various collector components. Lumped
177 capacitance models presented the simplest dynamical models that take into account the thermal mass by
178 considering aggregated thermal capacitances. This model can reproduce the observed variations with
179 satisfactory accuracy in the limit of low to moderate thermal inertia.

180 The mesh-based models [34] (the finite-element, finite-difference, and finite-volume methods) introduce
181 the discretization of the solar collector along the fluid direction. It is then possible to reproduce the fluid
182 temperature profile and, therefore, to calculate the outlet fluid temperature with a higher accuracy.
183 Compared to other discretization methods, the advantage of this method is to ensure the conservation of
184 flows and thus avoid the generation of parasitic sources. It has the advantage of strictly respecting the
185 conservation equations, not only in a global way (compared to finite-difference), but also for each volume
186 of control, this advantage is fully expressed in the case of strongly coupled and nonlinear equations
187 (compared to finite-element).

188 Although a 1D discretized model can predict the behaviour of collector with a good accuracy, it is not
189 able to predict the spatial distribution of temperature across the whole collector surface. However, a non-
190 homogeneity of the temperature distribution on a solar PV panel leads to a performance degradation and
191 ageing acceleration. It is therefore very important to understand the temperature distribution inside the
192 photovoltaic module. Moreover, the electrical performance decreases due to the non-uniform solar cell
193 temperature distribution which leads to a reverse saturation current [35] and to problems of mismatch
194 [36]. Furthermore, thermal expansion depends on the temperature of the local cell and the non-uniformity
195 of the cell temperature causes mechanical stress and decreases the lifespan of solar cells [37]. To
196 overcome the limitations associated with one-dimensional models of the PVT systems, the two-
197 dimensional models are then essential.

198 In the above-mentioned literature, a PVT collector that contains all the possible enhancements (anti-
199 reflective and low-emissivity coating, thermal resistance between the absorber plate and the cooling fluid,
200 the exchange surface area between the flat plat exchanger and the cooling fluid), within a single

201 component, has not yet been investigated and examined. The innovation and originality of this study is to
202 develop a two-dimensional approach using the finite volume method (FVM), in order to study the heat
203 transfer mechanisms (heat diffusion, thermal radiation, energy transfer) and the photo-conversion. The
204 choice of a 2D modelling approach allows to tackle the distribution of temperature in the photovoltaic
205 module and to increase the accuracy of the behavioural description in order to guide design decisions. The
206 advanced PVT collector is developed in order to: reduce the optical and thermal losses, maximize the
207 fluid temperature and to minimize the PV module operating temperature effect. Moreover, the impact of
208 the arrangements such as anti-reflective and low-emissivity coating, thermal resistance thermal between
209 the absorber plate and the cooling fluid, enhanced exchange surface area between the flat plat exchanger
210 and the cooling fluid are studied and analysed. Moreover, a comparison between the proposed PVT
211 collector and basic ones is performed in terms of temperature distribution and thermal and electrical
212 performances. In the section 2 the developed mathematical model is introduced. The numerical solution
213 and model validation are presented in section 3 and 4 and the results are presented in section 5.

214 **2. Mathematical MODEL**

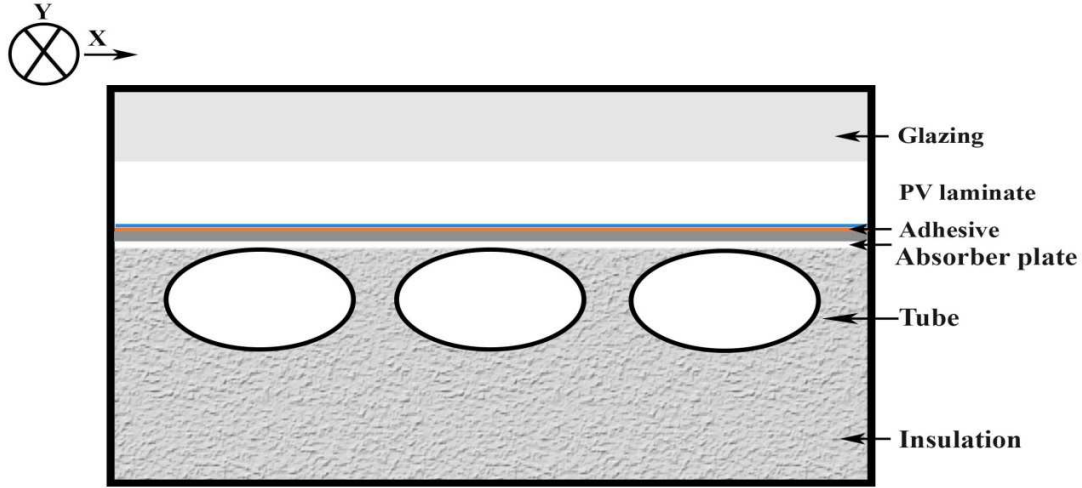
215 Theoretical models are carried out to examine and investigate the thermal and electrical behaviour for two
216 hybrid PVT collectors (sheet and tube design and advanced channelled configuration). An energy thermal
217 balance model inside the different components of two considered hybrid PVT collectors is established. It
218 is based on the geometric description of the system and the coupling between the conduction heat transfer
219 inside the different layers, convection and radiation exchange inside the collector and the heat exchanges
220 with surroundings (convective and radiation losses) and solar energy conversion to electrical power.
221 Various assumptions have been introduced in order to facilitate the calculations. (i) The solar radiation
222 distribution in the collector is uniform, (ii) The thermo-physical parameters of each solid layer of the
223 collector are considered constant, (iii) Thermal losses on the lateral sides are neglected. (iiii) Ohmic
224 losses in photovoltaic cells are neglected. The governing equations for the different collectors are
225 presented in the following sections.

226 **2.1. PVT Sheet and Tube Collector**

227 The proposed solar hybrid PVT sheet and tube without optical coatings is shown in Fig.1. An absorbent
228 sheet is jointed on the PV module rear area using an adhesive layer. The water flows across tubes welding
229 to the absorbent sheet in order to remove the un-useful heat from the PV module, therefore lowering the
230 running PV cells temperature and transferring heat to domestic use. In order to reduce the forward
231 thermal losses of the collector (wind convection and radiation losses) to the surrounding, a glazing is

232 placed between the PV module and the ambient air. A back thermal insulation (glass wool) is placed at
 233 the rear collector surface in order to decrease the back thermal losses collector.

234



235

236 **Figure 1: The proposed solar hybrid PVT sheet and tube (baseline) considered in the present study.**

237 ➤ Glass cover

238 The glazing thermal energy balance (Eq. 1) is estimated by considering that the stocked energy by the
 239 glazing ($m_g C_g \frac{dT_g}{dt}$) is identical to the sum of the radiation exchange flux ($Q_{ray,g \rightarrow env}$) between the collector
 240 and the surroundings (sky), the heat exchange by convection between the collector and the surroundings
 241 ($Q_{conv,g \rightarrow amb}$), the absorbed solar energy by glazing ($Q_{in,g}$), convective ($Q_{conv,g \rightarrow amb}$) and the radiation
 242 ($Q_{ray,pv \rightarrow g}$) heat transfer exchange of the air enclosed in the gap and the conductive flux within the glazing
 243 ($Q_{cond,g}$).

$$244 \quad m_g C_g \frac{dT_g}{dt} = Q_{in,g} - Q_{conv,g \rightarrow amb} - Q_{ray,g \rightarrow env} + Q_{conv,pv \rightarrow g} + Q_{ray,pv \rightarrow g} + Q_{cond,g} \quad (1)$$

245 $Q_{in,g}$ is the amount of solar radiation absorbed by the glazing, and is determined by

$$246 \quad Q_{in,g} = \alpha_g G \quad (2)$$

247 where, G is the solar radiation reached by the glazing and α_g is the effective absorptance of
 248 glazing.

249 $(Q_{\text{conv,g} \rightarrow \text{amb}})$ is the wind convection exchange from the glazing to ambient and it is calculated using the
 250 MacAdams empirical formula [38]

$$251 \quad (Q_{\text{conv,g} \rightarrow \text{amb}}) = A(5.7 + 3.8V_{\text{wi}})(T_{\text{amb}} - T_{\text{g}})$$

252 where, V_{wi} is wind speed. $T_{\text{amb}}, T_{\text{g}}$ are ambient, glazing temperatures, respectively.

253 The net infrared radiation exchange $(Q_{\text{ray,g} \rightarrow \text{env}})$ from the glazing to the sky can be determined assuming
 254 that the sky is a black body with a temperature of T_{sky} [39].

$$255 \quad Q_{\text{ray,g} \rightarrow \text{env}} = A\varepsilon_{\text{g}}\sigma(T_{\text{g}}^2 + T_{\text{sky}}^2)(T_{\text{g}} + T_{\text{sky}})(T_{\text{g}} - T_{\text{sky}}) \quad (4)$$

256 Where ε_{g} is the emissivity of glass cover, T_{sky} is the sky temperature, and σ is the Stefan Boltzmann
 257 constant ($5.6697 \times 10^{-8} \text{ W}/(\text{m}^2.\text{K}^4)$).

258 The sky (T_{sky}) temperature is given by the Swinbank formula [40].

$$259 \quad T_{\text{sky}} = 0.0522 * T_{\text{amb}}^{1.5} \quad (5)$$

260 The convective $(Q_{\text{conv,pv} \rightarrow \text{g}}$ and radiation $(Q_{\text{ray,pv} \rightarrow \text{g}})$ heat transfer exchange of the air enclosed in the
 261 gap, can be written as [3].

$$262 \quad Q_{\text{ray,pv} \rightarrow \text{g}} = A \frac{\sigma(T_{\text{g}}^2 + T_{\text{pv}}^2)(T_{\text{g}} + T_{\text{pv}})}{\frac{1}{\varepsilon_{\text{pv}}} + \frac{1}{\varepsilon_{\text{g}}} - 1} (T_{\text{g}} - T_{\text{pv}}) \quad (6)$$

$$263 \quad Q_{\text{conv,pv} \rightarrow \text{g}} = A \frac{\text{Nu}_a k_a}{\delta_a} (T_{\text{g}} - T_{\text{pv}}) \quad (7)$$

264 The Nusselt number (Nu_a) used can be evaluated using Eq. 8 [41]

$$265 \quad \text{Nu}_a = 1 + 1.44 \left[1 - \frac{1708}{\text{Ra}\delta_a \cos\theta} \right] * \left[1 - \frac{1708(\sin\theta)^{1.66}}{\text{Ra}\delta_a \cos\theta} \right] + \left[\frac{(\text{Ra}\delta_a \cos\theta)^{0.33}}{5830} - 1 \right] * \quad (8)$$

266 The flux by conduction in the glazing $(Q_{\text{cond,g}})$ can be written as Eq.9

$$267 \quad Q_{\text{cond,g}} = A_{\text{g}} k_{\text{g}} \delta_{\text{g}} \left(\frac{\partial^2 T_{\text{g}}(x,y)}{\partial x^2} + \frac{\partial^2 T_{\text{g}}(x,y)}{\partial y^2} \right) \quad (9)$$

268 ➤ Photovoltaic module

269 Using thermal energy balance, we have

270 $m_{pv} C_{pv} \frac{dT_{pv}}{dt} = Q_{in,pv} - Q_{conv,pv \rightarrow g} - Q_{ray,pv \rightarrow g} + Q_{cond,pv \rightarrow pab} - Q_{elec} + Q_{cond,pv}$ (10)

271 The photovoltaic energy balance (Eq. 10) is determined by considering that the stocked energy by the
 272 module ($m_{pv} C_{pv} \frac{dT_{pv}}{dt}$) is identical to the sum of the absorbed solar energy amount by module ($Q_{in,pv}$), heat
 273 flux by conduction inside the photovoltaic layer ($Q_{cond,pv}$), flux conduction transferred from module to
 274 absorber sheet ($Q_{cond,pv \rightarrow pab}$) minus the sum of the radiation exchange flux ($Q_{ray,pv \rightarrow g}$) and convective
 275 ($Q_{conv,pv \rightarrow g}$) between the module and the glazing, and the electrical energy generated by the solar cells
 276 (Q_{elec})

277 The absorbed solar energy amount by module ($Q_{in,pv}$) is described by Eq.11

278 $Q_{in,pv} = \alpha_{pv} G \tau_g$ (11)

279 Where α_{pv} solar cell absorption coefficient.

280 The electrical energy generated by the solar cells (Q_{elec}) is evaluated using the following empirical
 281 formula [42]:

282 $Q_{elec} = G \tau_g \eta_{elec} Pac \eta_0 [1 - \beta(T_{pv} - T_{ref})]$ (12)

283 Where β is the solar cell temperature coefficient (0.0045), η_0 is the reference efficiency for a reference
 284 temperature (25 °C), Pac is the packing factor and T_{pv} is the operating photovoltaic module temperature.

285 The conduction flux transferred from module to absorber sheet ($Q_{cond,pv \rightarrow pab}$) is described as follows:

286 $Q_{cond,pv \rightarrow pab} = R_{pv,pab} \Delta T$ (13)

287 Where the thermal conductance between the photovoltaic module and the absorber sheet is describes as:

288 $R_{pv,pab} = \frac{k_{ad}}{\delta_{ad}}$ (14)

289 Where k_{ad} and δ_{ad} are thermal conductivity and thickness adhesive layer, respectively.

290 The conduction flux inside PV module ($Q_{cond,pv}$) is defined as

291 $Q_{cond,pv} = A_{pv} k_{pv} \delta_{pv} \left(\frac{\partial^2 T_{pv}(x,y)}{\partial x^2} + \frac{\partial^2 T_{pv}(x,y)}{\partial y^2} \right)$ (15)

292 ➤ Absorber sheet

293 Using thermal energy balance for the absorber sheet, we have

$$294 \quad m_{\text{pab}} C_{\text{pab}} \frac{dT_{\text{pab}}}{dt} = Q_{\text{cond,pv} \rightarrow \text{g}} - Q_{\text{cond,pab} \rightarrow \text{tu}} - Q_{\text{cond,pab} \rightarrow \text{is}} + Q_{\text{cond,pab}} \quad (16)$$

295 The absorber sheet energy balance (Eq. 1) is determined by considering that the stocked energy by the
 296 module ($m_{\text{pab}} C_{\text{pab}} \frac{dT_{\text{pab}}}{dt}$) is identical to the sum of heat flux by conduction inside the absorber sheet
 297 ($Q_{\text{cond,pab}}$), and the flux conduction transferred from module to the absorber sheet ($Q_{\text{cond,pv} \rightarrow \text{pab}}$) minus the
 298 flux conduction transferred from absorber sheet to tube ($Q_{\text{cond,pab} \rightarrow \text{tu}}$) and from absorber to insulation
 299 $Q_{\text{cond,pab} \rightarrow \text{is}}$

300 The flux by conduction inside the absorber sheet ($Q_{\text{cond,pab}}$) is defined as Eq.17:

$$301 \quad Q_{\text{cond,pab}} = A_{\text{pab}} k_{\text{pab}} \delta_{\text{pab}} \left(\frac{\partial^2 T_{\text{pab}}(x,y)}{\partial x^2} + \frac{\partial^2 T_{\text{pab}}(x,y)}{\partial y^2} \right) \quad (17)$$

302 The conduction flux transferred from the absorber sheet to the tube ($Q_{\text{cond,pv} \rightarrow \text{pab}}$) is described as:

$$303 \quad Q_{\text{cond,pab} \rightarrow \text{tu}} = A_{\text{pab,tu}} h_{\text{cond,pab} \rightarrow \text{tu}} (T_{\text{tu}} - T_{\text{pab}}) \quad (18)$$

304 Where $A_{\text{pab,tu}}$ is the contact area between the absorber sheet and the tube [3]:

$$305 \quad A_{\text{pab,tu}} = \frac{\pi R dy}{2} \quad (19)$$

306 ➤ For the tube

307 Using thermal energy balance for the tube, we have

$$308 \quad \rho_t \delta_t A_t dy C_t \frac{dT_t}{dt} = A_{\text{pab,t}} h_{\text{cond,pab} \rightarrow \text{t}} (T_{\text{pab}} - T_t) + P e h_{\text{conv,t} \rightarrow \text{f}} dy (T_f - T_t) \\ + A_{\text{i,t}} h_{\text{cond,t} \rightarrow \text{i}} (T_i - T_t) + k_t \delta_t \left(\frac{\partial^2 T_t}{\partial^2 y} \right) \quad (20)$$

309 The insulation energy balance (Eq. 21) is determined by considering that the stocked energy by the tube
 310 ($\rho_t \delta_t A_t dy C_t \frac{dT_t}{dt}$) is identical to the sum of heat flux by conduction inside the tube $k_t \delta_t \left(\frac{\partial^2 T_t}{\partial^2 y} \right)$, and the flux
 311 conduction transferred from the absorber sheet to the tube $A_{\text{pab,t}} h_{\text{cond,pab} \rightarrow \text{t}} (T_{\text{pab}} - T_t)$ minus the heat
 312 exchange by convection between the tube and the fluid cooling $P e h_{\text{conv,t} \rightarrow \text{f}} dy (T_f - T_t)$ and the flux
 313 conduction transferred from the tube to the insulation $Q_{\text{cond,tu} \rightarrow \text{is}}$

314 Bejan [43] suggested a correlation for the Nusselt number, it is utilized in our work

315 $Re < 2300 \Rightarrow Nu_t = 4.364$ (21)

316 $Re > 2300 \Rightarrow Nu_t = 0.023Re^{0.8}Pr^{0.4}$ (22)

317 ➤ For the water

318 The energy balance of the fluid leads to

319 $\rho_f A_f dy C_f \frac{dT_f}{dt} = Peh_{conv,t \rightarrow f} dy (T_t - T_f) - \dot{m} C_f \Delta T_f$ (23)

320 The fluid energy balance (Eq. 23) is estimated by considering that the stored energy by the fluid
 321 ($\rho_f A_f dy C_f \frac{dT_f}{dt}$) is identical to the heat flux by convection between the tube and the fluid
 322 $Peh_{conv,t \rightarrow f} dy (T_t - T_f)$ minus the heat extracted by the water flow rate ($\dot{m} C_f \Delta T_f$).

323 ➤ For the insulation layer

324 The insulation energy balance (Eq. 24) is determined by considering that the stocked energy by the
 325 insulation layer ($\rho_{is} \delta_{is} C_{is} \frac{dT_{is}}{dt}$) is identical to the sum of heat flux by conduction inside the insulation sheet
 326 ($Q_{cond,in}$), and the flux conduction transferred from absorber sheet to insulation ($Q_{cond,pab \rightarrow is}$) and from the
 327 tube to the insulation ($Q_{cond,tu \rightarrow is}$) minus the heat exchange by convection between the insulation and
 328 the surroundings ($Q_{conv,is \rightarrow amb}$)

329 $\rho_{is} \delta_{is} C_{is} \frac{dT_{is}}{dt} = -Q_{conv,is \rightarrow amb} + Q_{cond,is} + Q_{cond,tu \rightarrow is} + Q_{cond,pab \rightarrow is} + Q_{cond,is}$ (24)

330 ($Q_{conv,is \rightarrow amb}$) is the wind convection exchange from the insulation to ambient and it is calculated using
 331 the MacAdams empirical formula [3]:

332 $(Q_{conv,is \rightarrow amb}) = A(5.7 + 3.8V_{wi})(T_{amb} - T_{is})$

333 The flux by conduction inside the insulation ($Q_{cond,is}$) is defined as:

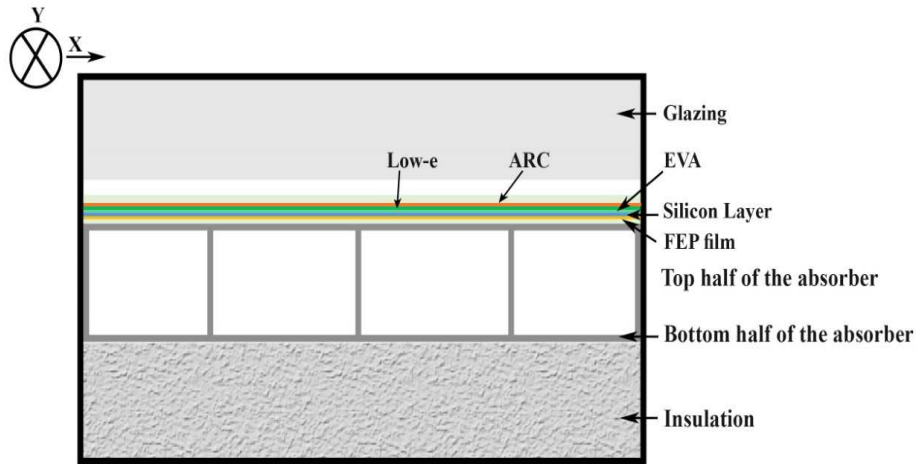
334 $Q_{cond,is} = A_{is} k_{is} \delta_{is} \left(\frac{\partial^2 T_{is}(x,y)}{\partial x^2} + \frac{\partial^2 T_{is}(x,y)}{\partial y^2} \right)$ (25)

335

336 **2.2. Proposed PVT Collector with Channel**

337 The advanced PVT channelled collector (Fig. 2) was arranged to perform at smaller solar cell temperature
 338 and larger water temperature than the first PVT sheet and tube configuration. The main distinguishing

339 component of the second collector. An anti-reflective coating (ARC) is applied to transparent covers and
 340 encapsulation surfaces and can be beneficial for both the thermal and electrical efficiencies. A low-
 341 emissivity coating is applied and can improve the thermal efficiency for collectors operating at high
 342 temperatures, but their inherent transmission losses reduce the electrical efficiency. A channel heat
 343 exchanger is used to decrease the thermal losses between the solar cells and the cooling fluid.



344
 345 **Figure 2: The advanced PVT channelled collector considered in the present study**

346 For the upper half of Absorber

347 The thermal energy balance of the upper half of the absorber leads to

$$348 \quad m_{pabu} C_{pabu} \frac{dT_{pabu}}{dt} = h_{cond,pv \rightarrow pabu} (T_{pv} - T_{pabu}) + \frac{A_{pabu,f}}{A_{pbu}} h_{conv,pabu \rightarrow f} (T_f - T_{pabu})$$

$$+ \frac{A_{pabu,pabh}}{A_{pbu}} h_{cond,pabu \rightarrow pabh} (T_{pabh} - T_{pabu}) + k_{pabu} \delta_{pabu} \left(\frac{\partial^2 T_{pabu}(x,y)}{\partial x^2} + \frac{\partial^2 T_{pabu}(x,y)}{\partial y^2} \right) \quad (26)$$

349 ➤ For the lower half of Absorber

350 The thermal energy balance of the lower half of the absorber leads to

$$351 \quad m_{pabh} C_{pabh} \frac{dT_{pabh}}{dt} = h_{cond,pabh \rightarrow is} (T_{is} - T_{pabh}) + \frac{A_{pabh,f}}{A_{pbh}} h_{conv,pabh \rightarrow f} (T_f - T_{pabh})$$

$$+ \frac{A_{pabu,pabh}}{A_{pbh}} h_{cond,pabu \rightarrow pabh} (T_{pabu} - T_{pabh}) + k_{pabh} \delta_{pabh} \left(\frac{\partial^2 T_{pabh}(x,y)}{\partial x^2} + \frac{\partial^2 T_{pabh}(x,y)}{\partial y^2} \right) \quad (27)$$

352 ➤ For the heat carrier fluid

353 The thermal energy balance of the heat carrier fluid leads to

354
$$\rho_f A_f dy C_f \frac{dT_f}{dt} = -\dot{m} C_f \Delta T_f + A_{pabu,f} h_{conv,pabu \rightarrow f} (T_{pabu} - T_f) + A_{pabh,f} h_{conv,pabh \rightarrow f} (T_{pabu} - T_f) \quad (28)$$

355 ➤ For the insulation layer

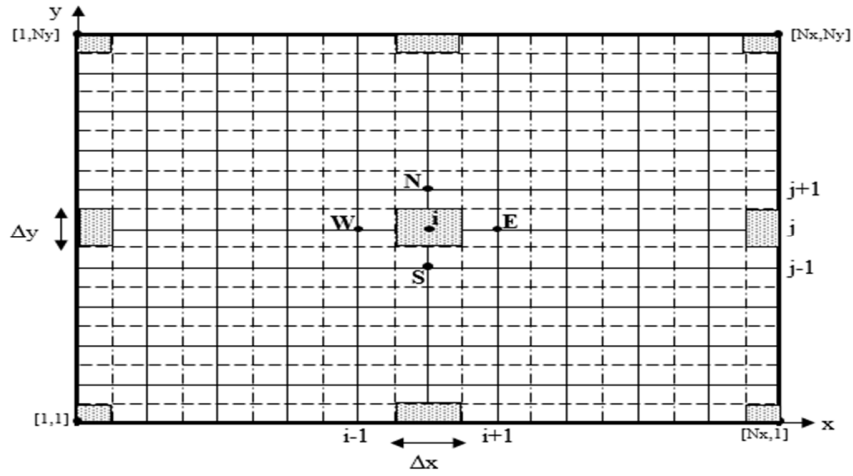
356 The thermal energy balance of the insulation leads to

357
$$m_{is} C_{is} \frac{dT_{is}}{dt} = h_{wind} (T_{amb} - T_{is}) + h_{cond,pabh \rightarrow is} (T_{is} - T_{pabh}) + k_{is} \delta_{is} \left(\frac{\partial^2 T_{is}(x,y)}{\partial x^2} + \frac{\partial^2 T_{is}(x,y)}{\partial y^2} \right) \quad (29)$$

358 **3. Numerical Solution and validation**

359 **3.1. Numerical Solution**

360 The system of the differential equations described above is nonlinear. Therefore, rather than attempting to
361 obtain an analytical treatment, it is numerically discretized and solved using the finite volume method
362 (FVM). The general approach of the FVM involves the definition of the control volumes to contain the
363 nodes associated with the state variables of the model. The governing heat balance equations for the two
364 different collectors presented in the above sections were solved by the classical two-dimensional FVM
365 and an implicit method. The entire computational domain is discretised using a uniform structured
366 130*100 mesh. This grid value is based on the mesh independency analyses for a 2m x 1m surface PVT
367 conducted with an uniform mesh of constant control volume dimensions Δx and Δy for internal cells, and
368 half and quarter volumes for edges and corners of each body (Fig.3). The mathematical established
369 model is implemented in a Fortran platform in accordance to the simulation algorithm procedural
370 summarized in Fig. 4. In order to numerically the discretized equations system, an iterative Gaussian
371 elimination method is used. The simulation starts by initiating arbitrary temperatures of glass, PV,
372 absorber, tube and insulation equal to the ambient temperature, as well as the temperature of working
373 fluid is equal to the inlet water temperature and then solved iteratively until the convergence criteria is
374 obtained. This criteria is defined by a threshold in the relative discrepancy (10^{-9}). The calculation is
375 repeated for all nodes before proceeding to the next time step ($\Delta t=1s$), initiated using the latest solution.



376

377 **Figure 3: Mesh describing the spatial discretisation of the system**

378 The physical size value (T) in the node $[i,j]$, at time $t + \Delta t$ is considered to be $T_{i,j}(t + \Delta t)$.

379 For the temporal discretization, an unsteady state implicit scheme was adopted as follows:

380
$$\int_t^{t+\Delta t} T(t)dt = (T(t + \Delta t) - T(t))\Delta t = (T - T_0)\Delta t \quad (30)$$

381 where the accumulation and the source terms are assumed to be constant inside each control volume. The
 382 first derivatives in space on the faces of the control domain are evaluated assuming that the variations
 383 between two neighbouring nodes are linear (Eq. 31 to 34).

384
$$\left(\frac{\partial T}{\partial x}\right)_{i+\frac{1}{2},j} = \frac{T[i+1,j] - T[i,j]}{\Delta x} \quad (31)$$

385
$$\left(\frac{\partial T}{\partial x}\right)_{i-\frac{1}{2},j} = \frac{T[i,j] - T[i-1,j]}{\Delta x} \quad (32)$$

386
$$\left(\frac{\partial T}{\partial x}\right)_{i,j+\frac{1}{2}} = \frac{T[i,j+1] - T[i,j]}{\Delta y} \quad (33)$$

387
$$\left(\frac{\partial T}{\partial x}\right)_{i,j-\frac{1}{2}} = \frac{T[i,j] - T[i,j-1]}{\Delta y} \quad (34)$$

388 Using these hypotheses, the discretized equations can be written as follows:

389
$$a_0 T[i,j] = a_s T[i,j-1] + a_n T[i,j+1] + a_w T[i-1,j] + a_e T[i+1,j] + b \quad (35)$$

390 $a_0, a_s, a_n, a_w, a_e, b$ are coefficients that are depending on the selected solutions profiles, physical
 391 properties and calculation steps.

392 Using these hypotheses, the discretized equations can be written as follows:

$$393 \quad a_0 T[i,j] = a_s T[i,j-1] + a_n T[i,j+1] + a_w T[i-1,j] + a_e T[i+1,j] + b \quad (36)$$

394 $a_0, a_s, a_n, a_w, a_e, b$ are the coefficients that are depending on the selected solutions profiles, physical
395 properties and calculation steps.

396 As an example, if we consider the case for the inside face of the photovoltaic module, by grouping the
397 equations (Eq.10-15 and Eq. 43-47), the resultant becomes (Eq. 37 to 42):

$$398 \quad a_0 = [(\rho_{pv} C_{pv} \delta_{pv} \Delta x \Delta y) + h_{ray,pv \rightarrow g} \Delta t \Delta x \Delta y + h_{conv,pv \rightarrow g} \Delta t \Delta x \Delta y + h_{cond,pab \rightarrow pv} \Delta t \Delta x \Delta y + k_{pv} \delta_{pv} \frac{\Delta t \Delta y}{\Delta x} \\ + k_{pv} \delta_{pv} \frac{\Delta t \Delta y}{\Delta x} + k_{pv} \delta_{pv} \frac{\Delta t \Delta x}{\Delta y} + k_{pv} \delta_{pv} \frac{\Delta t \Delta x}{\Delta y}] \quad (37)$$

$$399 \quad a_w = k_{pv} \delta_{pv} \frac{\Delta t \Delta y}{\Delta x} \quad (38)$$

$$400 \quad a_e = k_{pv} \delta_{pv} \frac{\Delta t \Delta y}{\Delta x} \quad (39)$$

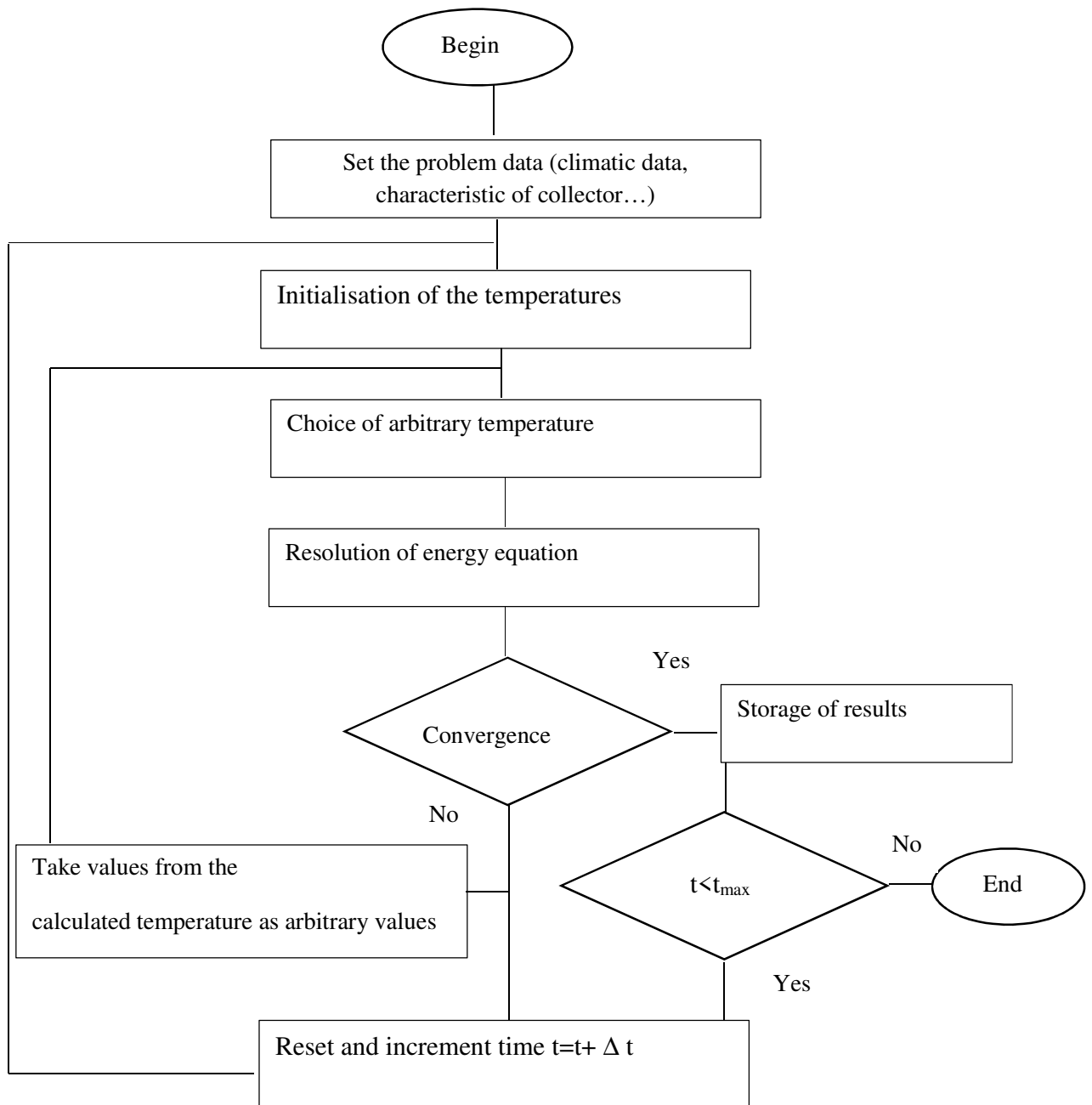
$$401 \quad a_s = k_{pv} \delta_{pv} \frac{\Delta t \Delta x}{\Delta y} \quad (40)$$

$$402 \quad a_n = k_{pv} \delta_{pv} \frac{\Delta t \Delta x}{\Delta y} \quad (41)$$

$$403 \quad b = (\rho_{pv} C_{pv} \delta_{pv} \Delta x \Delta y T_{pv(i,j)}^t) + h_{ray,pv \rightarrow g} T_{g(i,j)}^{t+dt} \Delta t \Delta x \Delta y + h_{conv,pv \rightarrow g} T_{g(i,j)}^{t+dt} \Delta t \Delta x \Delta y \\ + h_{cond,pab \rightarrow pv} T_{pab(i,j)}^{t+dt} \Delta t \Delta x \Delta y \quad (42)$$

404

405



406
407

408 **Figure 4: Simulation procedural algorithm**

409

410

411

412

413 **3.2. Model Validation**

414 The collector performance is characterised by the thermal and electrical efficiencies as a function of the
 415 reduced temperature. The reduced temperature corresponds to the temperature difference between the
 416 fluid and the ambient divided by the solar radiation. The thermal efficiency is defined by

$$417 \quad \eta_{th} = \dot{m}_w C_p \frac{(T_m - T_{amb})}{GA} \quad (43)$$

418 where T_m is the average of inlet and outlet temperatures, $T_{out} - T_{in}$, and \dot{m}_w is the mass flow rate.

419 According to the EN 12975 standard, the thermal efficiency is given by the following equation and it is
 420 related to the thermal conversion factor for the no loss (η_0), linear heat loss coefficient (C1) and
 421 Temperature dependence of heat losses (C2)

$$422 \quad \eta_{th} = \eta_0 - C_1 \left[\frac{(T_{in} - T_{amb})}{G} \right] - C_2 \left[\frac{(T_{in} - T_{amb})^2}{G} \right]$$

423 (44)

$$424 \quad \eta_{elec} = \eta_{elec,0} - C_3 \left[\frac{(T_{in} - T_{amb})}{G} \right] \quad (45)$$

425 The validation of the numerical model associated with the PVT sheet and tube collector is achieved by the
 426 comparison with the fittings results obtained experimentally on the EN 12975 standard steady state
 427 efficiencies by Bhattarai [44] (see Fig.5). Table 1 summarizes the parameters of the collector. We kept the
 428 same physical properties of the collector components, the surface of the PVT collector and the
 429 arrangement of the tubes used by Bhattarai [44] in our numerical code. A good concurrence was obtained
 430 for this first model.

431 To assess the extent of agreement our numerical model, the mean root square deviation (RMS) is
 432 performed. It is defined as follows:

$$433 \quad RMS = \sqrt{\frac{\sum (100 * (X_{exp,i} - X_{num,i})^2)}{n_{exp} * (X_{exp,i})^2}} \quad (46)$$

434 where $X_{exp,i}$ and $X_{num,i}$ denote the experimental and numerical results, respectively.

435 The RMS of electrical and thermal efficiencies are 2.31%, 0.57%, respectively. Therefore, we have
 436 obtained a satisfactory conformance. The numerical model reproduces the dynamics observed in the
 437 experiment, and the response of the collector to varying environmental conditions. The curves in Fig. 5a

438 describe the correlation of thermal efficiency to reduced temperature, obtained by fitting experimental
 439 data (eq.47) from [44], and the correlation resulting from the simulation (eq.48).

$$440 \quad \eta_{th, \text{exp linear fitting}} = 0.5887 - 7.0524 \left[\frac{(T_m - T_{amb})}{G} \right] - 0.269 \left[\frac{(T_m - T_{amb})^2}{G} \right]$$

441 (47)

$$442 \quad \eta_{th, \text{modeling, present work}} = 0.58 - 6.8 \left[\frac{(T_m - T_{amb})}{G} \right] - 0.025 \left[\frac{(T_m - T_{amb})^2}{G} \right]$$

443 (48)

444 The curves in Fig. 5b describe the correlation of electrical efficiency to reduced temperature, obtained by
 445 fitting experimental data (eq.49) from [44], and the correlation resulting from the simulation (eq.50).

$$446 \quad \eta_{elec, \text{modeling, present work}} = 0.137 - 0.5 \left[\frac{(T_{in} - T_{amb})}{G} \right] \quad (49)$$

$$447 \quad \eta_{elec, \text{exp linear fitting}} = 0.1369 - 0.477 \left[\frac{(T_m - T_{amb})}{G} \right]$$

448 (50)

449 A good agreement is observed, especially for thermal efficiency versus a values of reduced temperature
 450 inferior to 0.075 Km²/W, and for electrical efficiency versus a values of reduced temperature inferior to
 451 0.06 Km²/W. A gradual divergence is evident towards higher ambient temperature and solar radiation
 452 flux. Considering that uncertainties in material properties and parameters were small, the discrepancy was
 453 attributed to an underestimation of losses by convection to ambient air and long wave radiation by the
 454 model compared to the experiment. The deviation amounts to an overestimation of thermal efficiency by
 455 0.05% for a reduced temperature of 0.08 Km²/W, which can be considered the difference in performance
 456 between an ideal collector and a real system. For the parametric analysis presented in the following
 457 sections, this systematic error is assumed invariant for all the configurations.

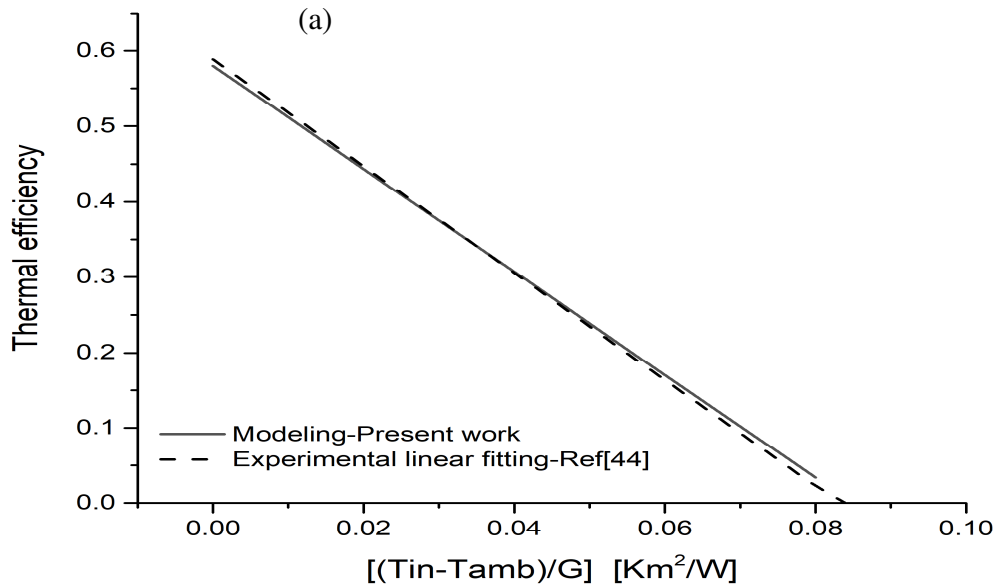
458 A second validation was undertaken using the experimental results obtained by Sarradarbadi et al [45],
 459 who evaluated the efficiencies of an uncovered PVT-water collector under Mashad (Iran) climatic
 460 conditions. The numerical simulation was conducted under the experimental conditions given in [45].
 461 Taking into account the difference in experimental set up, some changes were applied to the model. In
 462 particular, the glass layer was removed, and the balance energy equation was substituted as follows (Eq.
 463 51):

464

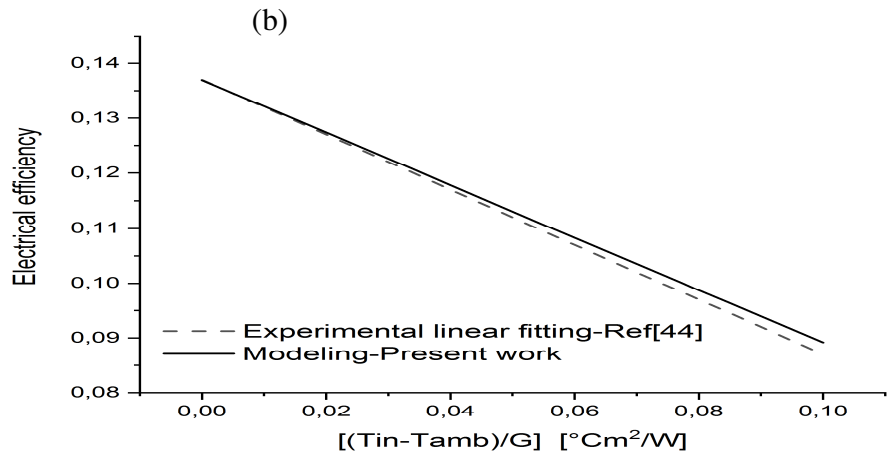
$$m_{pv}C_{pv} \frac{dT_{pv}}{dt} = h_{cond,pv \rightarrow pabu}(T_{pab} - T_{pv}) + \alpha_{pv}G + (5.7 + 3.8V_{wi})(T_{amb} - T_{pv}) - Q_{elec} \quad (52)$$

$$\varepsilon_{pv}\sigma(T_{pv}^2 + T_{sky}^2)(T_{pv} + T_{sky})(T_{pv} - T_{sky}) + k_{pv}\delta_{pv}\left(\frac{\partial^2 T_{pv}(x,y)}{\partial x^2} + \frac{\partial^2 T_{pv}(x,y)}{\partial y^2}\right)$$

465 The predicted results provided by our code and experimental data of outlet water temperature are plotted
 466 in Fig. 5-c. It is noted that the simulation results agree well with the experimental data carried out by
 467 Sarradarbadi et al. [45]. The obtained RMS of outlet water temperature is 1.5%. Therefore, we have
 468 obtained a satisfactory conformance. It can be concluded that our numerical model can be used to predict
 469 the behaviour of PV T collector with low relative errors.



470
 471 **Figure 5-a: Comparison of predicted thermal efficiency with the experimental fit curve presented in [44].**

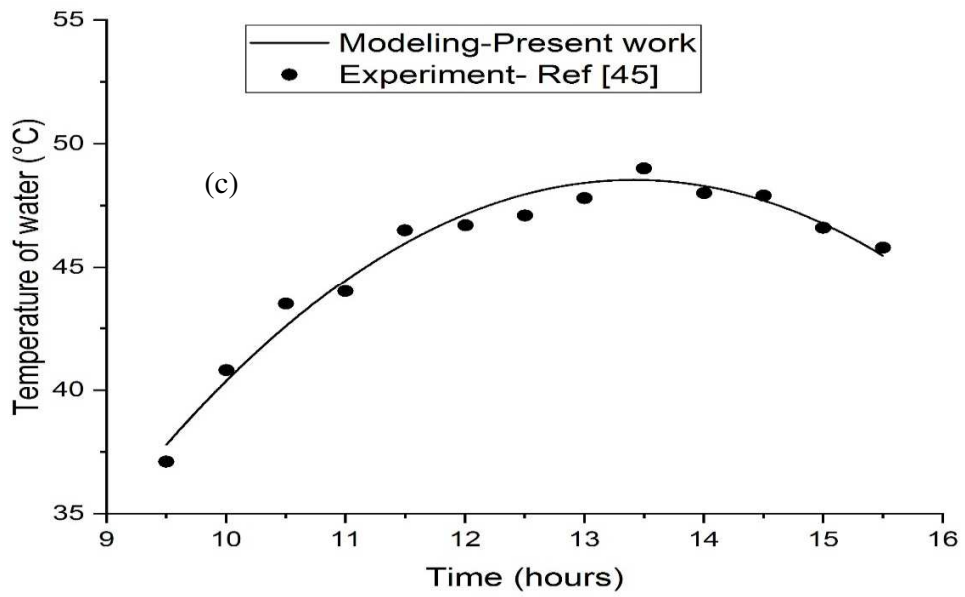


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473

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Figure 5-b: Comparison of predicted electrical efficiency with the experimental fit curve presented in [44].



475

476

477

Figure 5-c: Comparison of the outlet water temperature versus the experimental results performed by Sarradarbadi et al. [45].

Components	PVT-sheet and tube	Parameters	
Glazing	0.004m	Thickness, δ_g 478	
	670 (J/kgK)	Specific heat, C_g 479	
	2200 (kg/m ³)	Density, ρ_g 480	
	0.9 (W/mK)	Thermal conductivity, k_g 481	
PV module	900 (J/kgK)	Specific heat, C_p 482	
	140 (W/mK)	Thermal conductivity, k_{pv} 483	
	1	Packing factor, P_f 484	
	0.405 K-1	Solar cell temperature coefficient, β 485	
Absorber Plate	2702(kg/m ³)	Density, ρ_{pab} 486	
	310 (W/mK)	Thermal conductivity, k_{pab} 487	
Tube/channels	2702 (kg/m ³)	Density 488	
	310 (W/mK)	Thermal conductivity 489	
	10	Tube/channels number 490	
	0.008m	Diameter of tube / length of channel 491	
	0.004m	Radius of tube / width of channel 492	
	0.0012m	Thickness 493	
	Insulation	0.05m	Thickness, δ_i 494
		20(kg/m ³)	Density, ρ_i 495
0.030(W/mK)		Thermal conductivity 496	

Table 1: Parameter sets for the photovoltaic thermal (PVT) sheet and tube collector

4. Results and Discussion

In this section, the effect of the modification (optical coating, thermal resistance thermal between the photovoltaic and absorber plate, choice of the exchange area between the absorber exchanger and the cooling fluid) utilized to transform the basic PVT sheet and tube design to a most advantageous PVT configuration is examined and discussed. Moreover, a comparison between the advanced PVT collector and the basic ones is performed in terms of the temperature distribution.

According to the ISO 9806 standard, the thermal efficiency is given by the following equation and it is characterized by three fit-parameters namely: η_0 is the thermal conversion factor when the reduced temperature is equal to zero. C_1 and C_2 are linear and quadratic heat loss coefficients.

507 $\eta_{th} = \eta_0 - C_1 \left[\frac{(T_m - T_{amb})}{G} \right] - C_2 \left[\frac{(T_m - T_{amb})^2}{G} \right]$
 508 (53)

509 Where T_m is the average of inlet and outlet temperatures, $T_{out} - T_{in}$, and \dot{m}_w is the mass flow rate.

510 Performing linear regression of electrical efficiency η_{elec} to reduced temperature $\left(\left[\frac{(T_m - T_{amb})}{G} \right] \right)$ with the,
 511 the thermal efficiency is given by the following equation:

512 $\eta_{elec} = \eta_{elec,0} - C_3 \left[\frac{(T_m - T_{amb})}{G} \right]$ (54)

513 Where

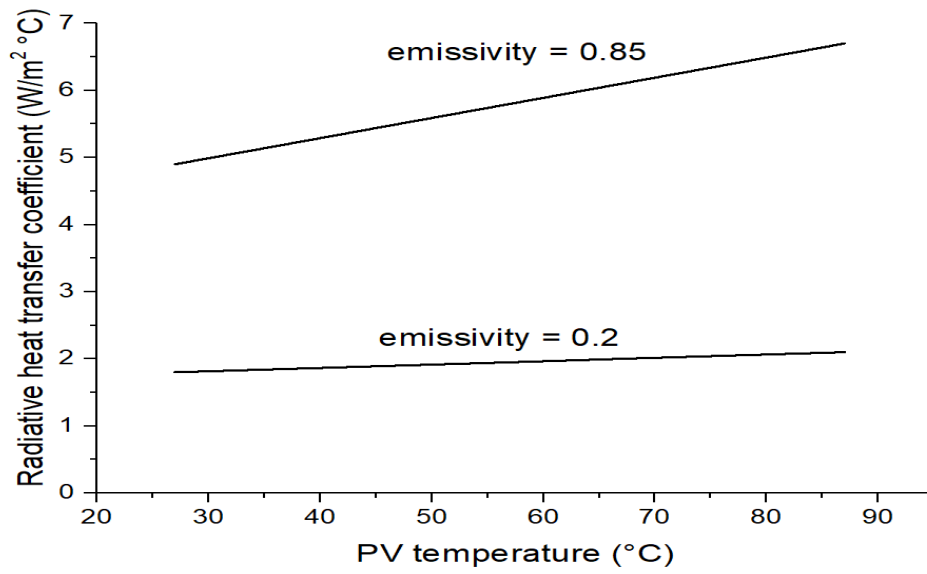
514 $\eta_{elec,0}$ is Electrical conversion factor for the no loss, C_3 is the Linear temperature dependence factor
 515

516 **4.1. Parametric study**

517 **4.1. 1. The Impact of Optical Coatings Applied to the PVT Collector**

518 **• Low-emissivity coatings**

519 As shown in Fig.6, an increase of emissivity leads to an increase of the radiation heats loss and thus a
 520 decrease of the thermal efficiency). The addition of a low-emissivity coating can overcome this drawback,
 521 however this also results in an increase in the reflectance, hence lower absorptance, and leads to a rise in
 522 collector temperature. The consequence is a fall in electrical efficiency. It is therefore interesting to
 523 choose an optimal value with regards to the electrical and thermal efficiencies. As shown in Fig.7, an
 524 emissivity of 0.45 offered the best compromise for the PVT configuration under investigation.

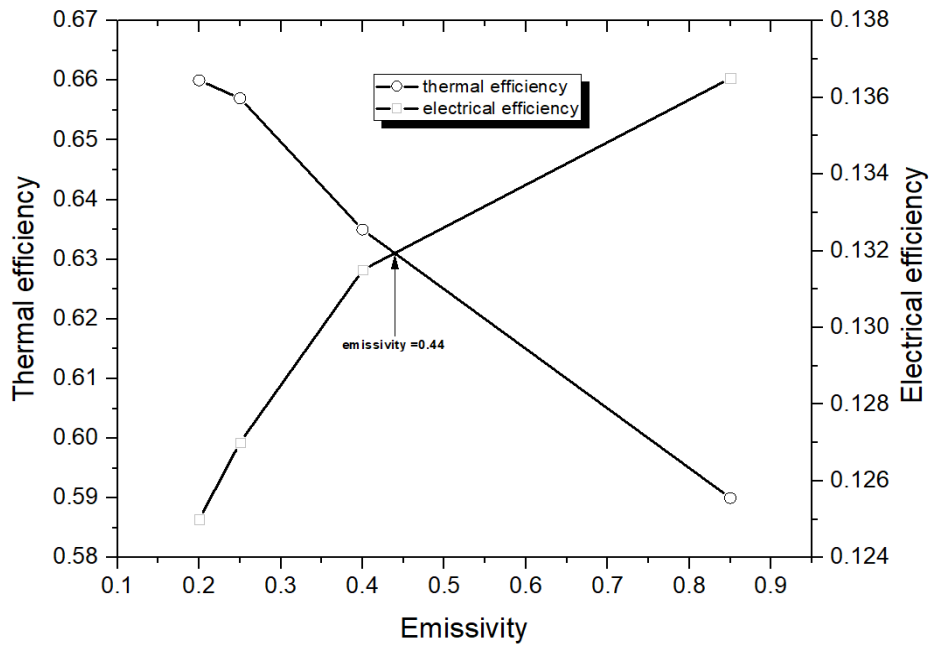


525

526

527

Figure 6: Radiative heat loss coefficient as function of photovoltaic module temperature for a range in longwave emissivity.



528

529

Figure 7: Thermal and electrical efficiencies as a function of emissivity.

530

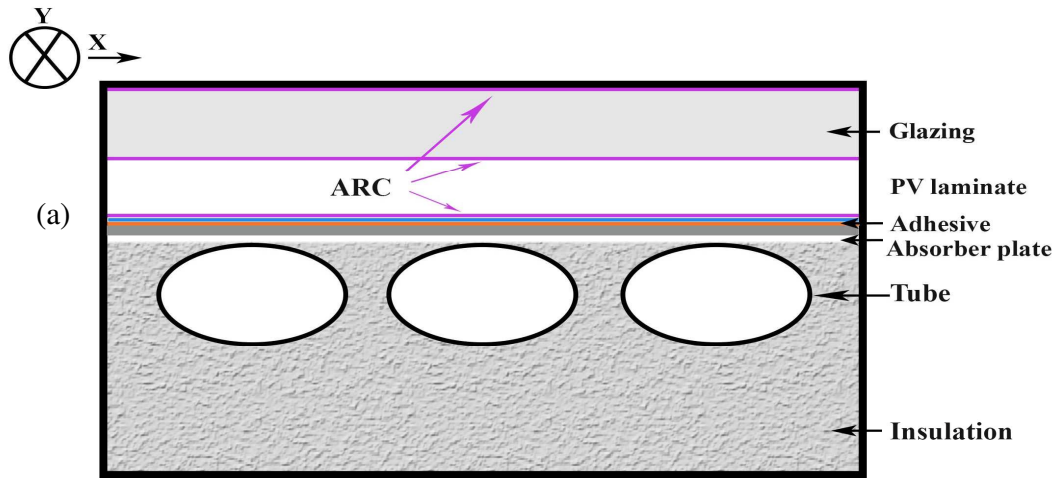
- **Anti-reflective coating**

531 The thermal and electrical efficiency for the PVT sheet and tube without a specific optical coatings
532 (baseline) and for PVT with anti-reflective coatings are shown in table 2 and Fig.8. From these results, we
533 can observe that the enhancement in the solar transmission improves both thermal and electrical
534 efficiencies. The radiation transmission of the glass cover can be improved by applying an anti-reflective
535 coatings (ARC), which provide a better optical matching of the refractive indices between the glazing and
536 the static air layer, and between the glass cover and the surrounding air. On the other hand, a higher
537 reflective (without coating) leads to the increase in the thermal losses of the collector. As shown in Table
538 2 and Fig.8, without ARC and no loss (i.e. temperature of fluid is equal to the ambient temperature) the
539 PVT collector exhibited 13.70% electrical efficiency and 58% thermal efficiency whereas those with
540 ARC exhibit 14.7% electrical and 61% thermal performances. The first order heat loss coefficient C1
541 decrease from 6.8 W/m²K to 4.2 W/m²K. This is a due that the use of anti-reflective coating can be
542 minimized the radiative heat loss

Parameter	PVT sheet and tube without optical coatings (baseline)	PVT sheet and tube with Anti-reflective coating	PVT sheet and tube with anti-reflective and low-emissivity coatings	PVT sheet and tube absorber with anti-reflective and low-e coating, assuming a simple adhesive or whole package lamination manufacturing process	Advanced channelled PVT
Thermal conversion factor for the no loss (η_0)	58%	61%	67%	71%	73%
linear heat loss coefficient (C1) W/m ² K	6.8	4.2	3.9	3.9	3.9
Temperature dependence of heat losses (C2) W/m ² K	0.025	0.026	0.028	0.028	0.028
Electrical conversion factor for the no loss (η_{elec})	13.70%	14.70%	14.25%	15.10%	15.40%
Linear temperature dependence factor W/m ² K	0.5	0.45	0.45	0.45	0.45

543 **Table 1: Thermal and electrical efficiencies curves parameters for different considered PV-T collectors**

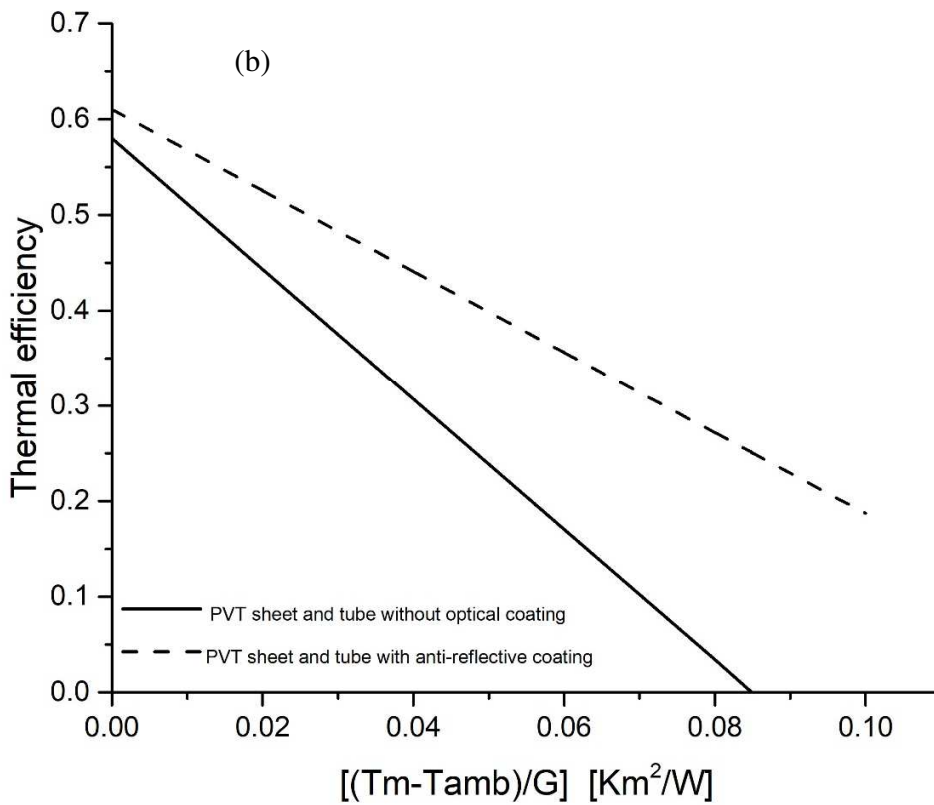
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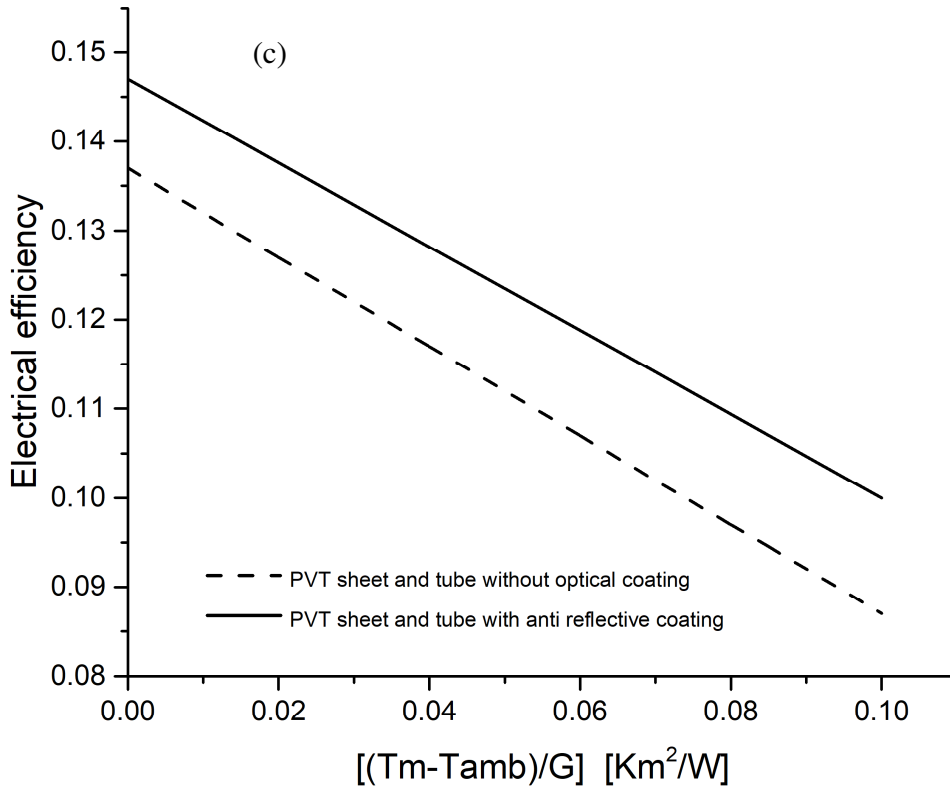
546 **Figure 8-a: PVT sheet and tube collector with anti-reflective coatings**

547



548

549 **Figure 8-b: Thermal efficiency for a PV/T sheet-absorber without optical coatings, and for a PV/T sheet and tube**
 550 **absorber with anti-reflective coating.**

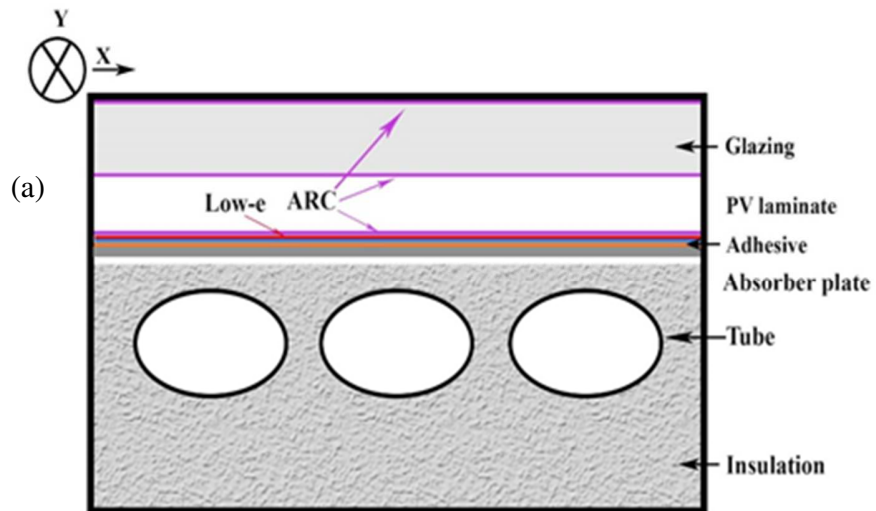


551
 552 **Figure 8-c: Electrical efficiency for a PV/T sheet-absorber without optical coatings, and for a PV/T sheet and tube**
 553 **absorber with anti-reflective coating.**

554 **• Anti-reflective and low-emissivity coating**

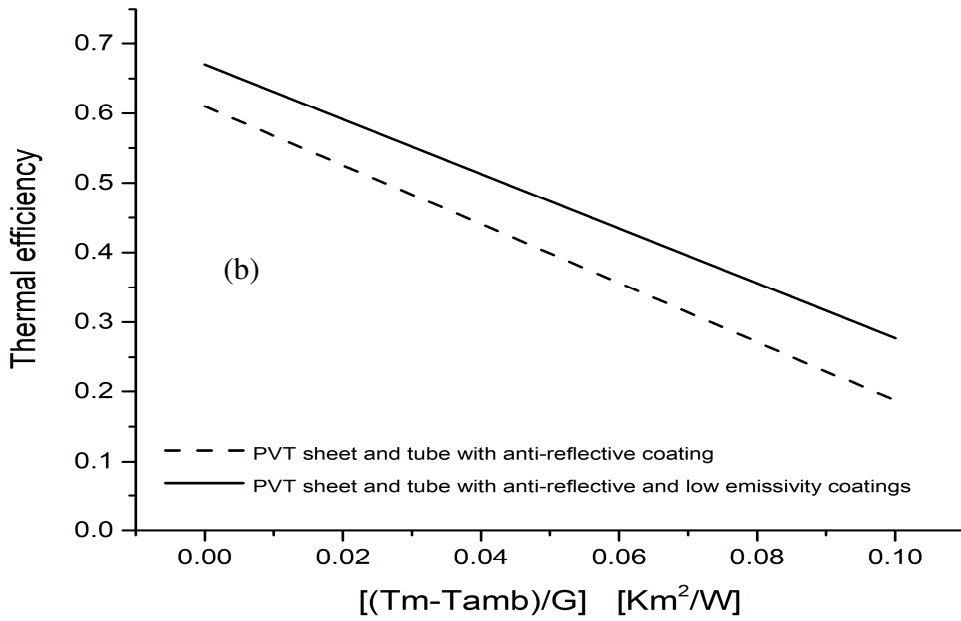
555 The thermal and electrical efficiency for the PVT sheet and tube without optical coatings (baseline) and
 556 for PVT with both anti-reflective and with a low emissivity coatings are shown in table 2 and Fig.9. From
 557 these results, we can note that for zero loss the PVT sheet and tube absorber with both anti-reflective and
 558 with a low emissivity provides the best thermal efficiency (67%) among side-by- side PVT with anti-
 559 reflective (61%) and PVT without optical coating (58%). This is due to the fact that the infrared radiation
 560 losses can be reduced by applying a selective coating (low-e coating). As shown in in table 2 and Fig.9,
 561 without ARC and low emissivity, for the zero loss, the baseline PVT collector exhibits 13.7% electrical
 562 efficiency whereas those with ARC and low emissivity exhibit 14.25% performance. On the other hand,

563 the electrical efficiency for the PVT sheet and tube absorber with both anti-reflective and with low
564 emissivity (14.25%) were lower than for PVT sheet and tube with ARC and without low emissivity
565 (14.7%). The use of low emissivity induces the augmentation in temperature of collector resulting the
566 drop in electrical efficiency.

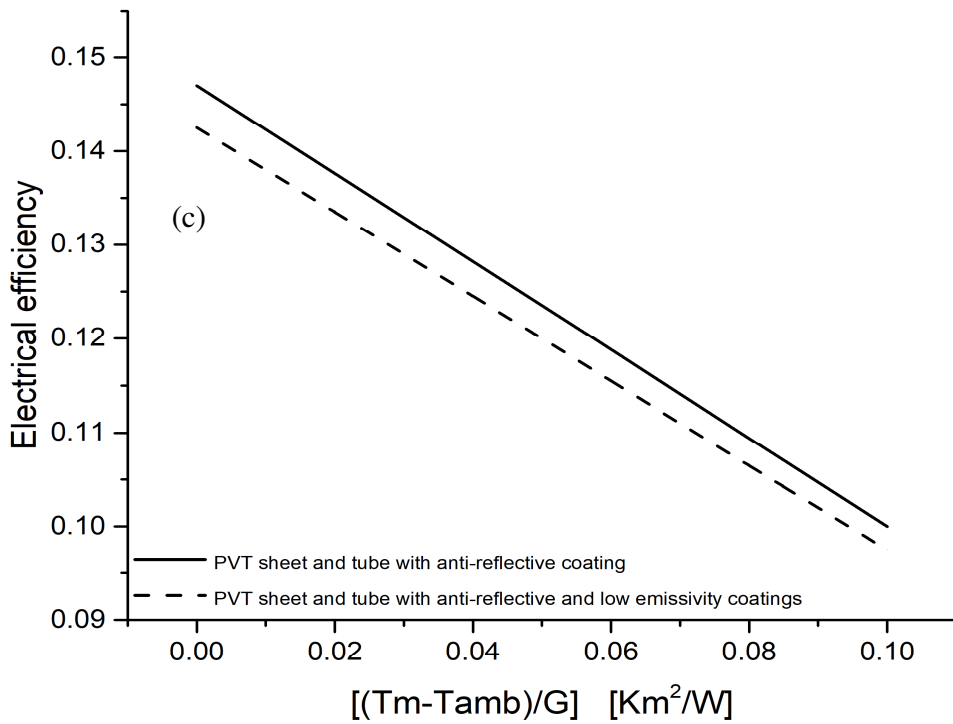


567

568 **Figure 9-a: PVT sheet and tube collector with both anti-reflective and with a low emissivity coating**



569
 570 **Figure 9-b: Thermal efficiency for a PV/T sheet-absorber with anti-reflective coatings, and for a PVT sheet and tube**
 571 **absorber with both anti-reflective and with a low emissivity coating.**



572

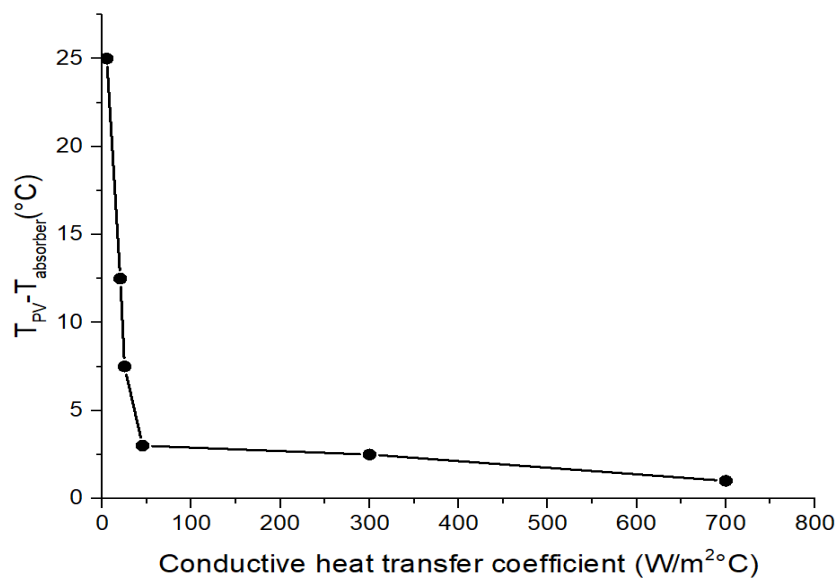
573 Figure 9-c: electrical efficiency for a PV/T sheet-absorber with anti-reflective coatings, and for a PVT sheet and tube
574 absorber with both anti-reflective and with a low emissivity coating.

575

- 576 • **The impact of thermal resistance thermal between the PV module and the absorber**
577 **plate**

578 As shown in Fig.10, the increase in conductive heat coefficient between the PV module and the absorber
579 leads to a reduction in the temperature gradient between the PV module and the absorber. As shown in
580 Table 2, with AR coatings and low emissivity PVT with a simple adhesive which corresponds to a heat
581 transfer coefficient of a 45W/m²K exhibited 14.25% electrical and 67% thermal performances for the no
582 loss whereas those PVT with advanced encapsulation of components during the lamination manufacturing
583 phase exhibited 15.10% electrical and 71% thermal performances. The advantage of this method is to
584 greatly reduce the contact resistance (700W/m²K) between the photovoltaic and thermal layers. The
585 enhancement of the conductive heat transfers between the PV module and the absorber results in a smaller
586 temperature gradient and therefore a lower PV cell temperature and higher absorber temperature. As a
587 consequence, it improved both thermal and electrical efficiencies, as shown in table 2 and Fig.11

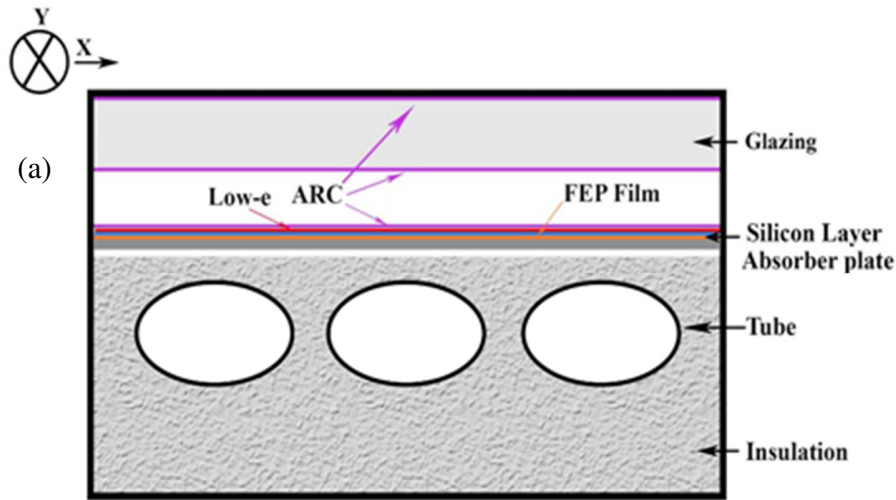
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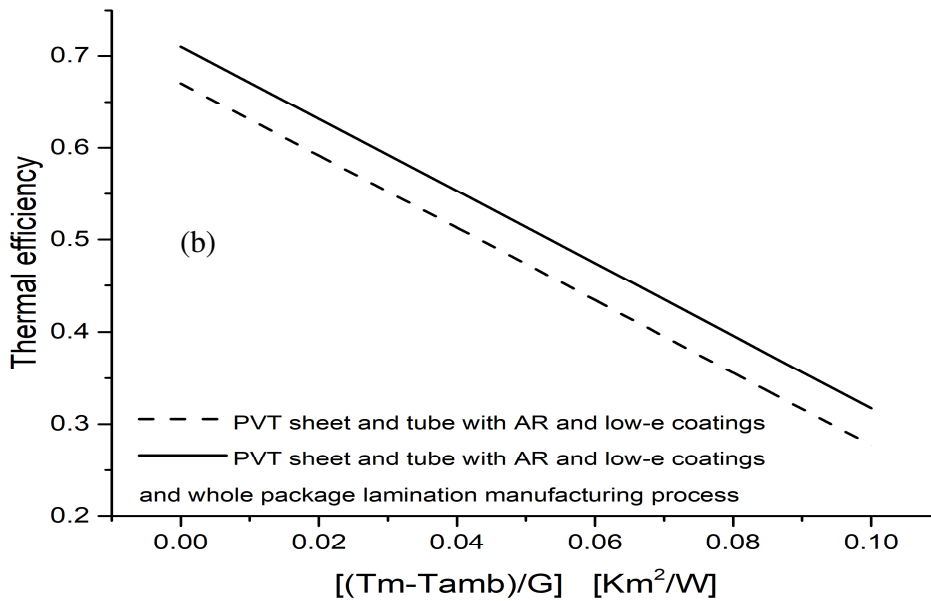
590 **Figure 10: Temperature difference between PV module and absorber layers versus conductive heat coefficient.**

591



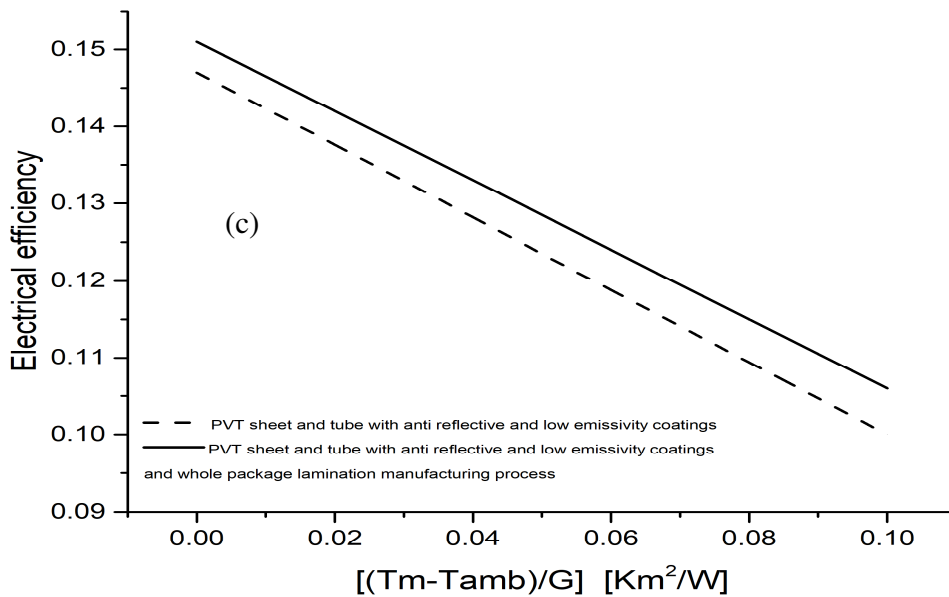
592

593 **Figure 11-a: PVT sheet and tube collector with both anti-reflective and with a low emissivity coating and whole package**
 594 **lamination manufacturing process.**



595

596 **Figure 11-b: Predicted thermal efficiencies of a PVT sheet and tube absorber with anti-reflective and low-e coating,**
 597 **assuming a simple adhesive or whole package lamination manufacturing process.**

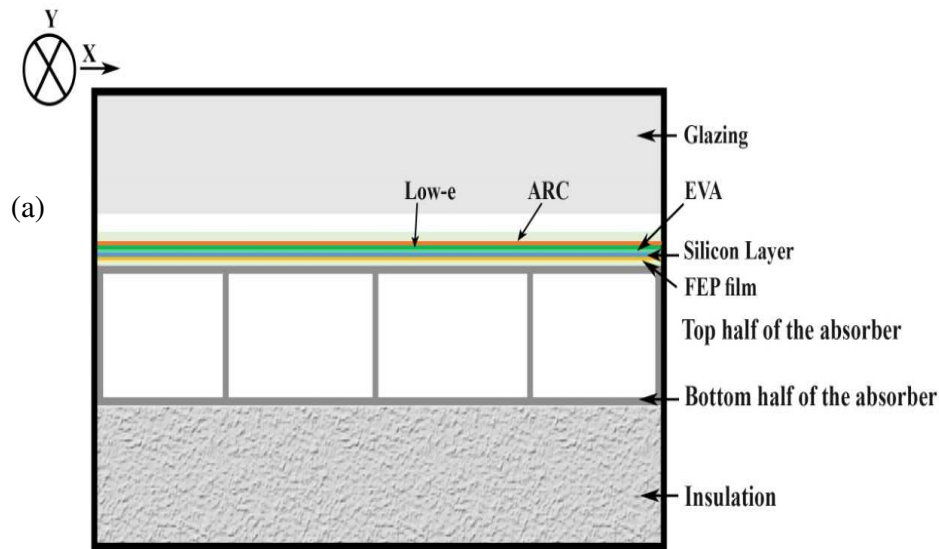


598

599 **Figure 11-c: Predicted electrical efficiency of a PVT sheet and tube absorber with anti-reflective and low-e coating,**
 600 **assuming a simple adhesive or whole package lamination manufacturing process.**

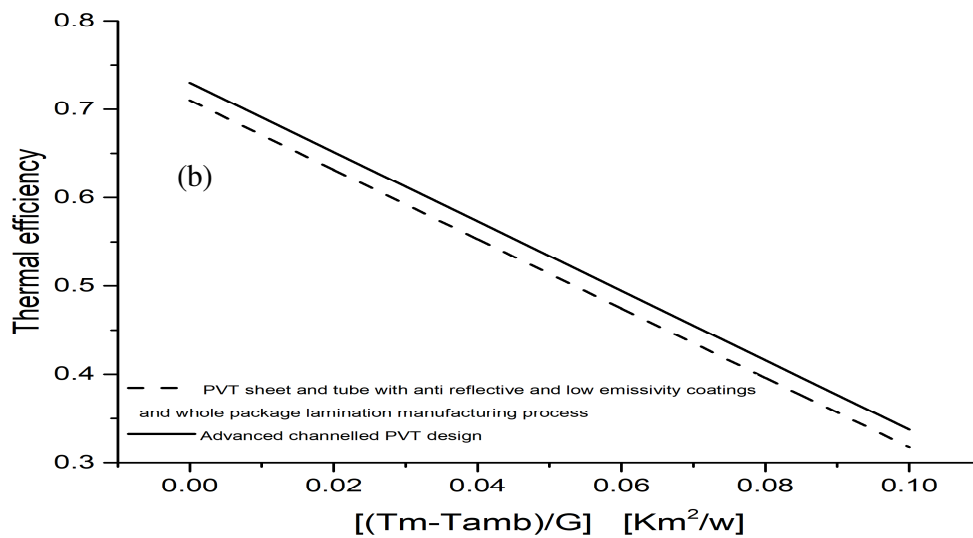
- 601 • **The impact of thermal resistance thermal between the absorber plate and the**
 602 **cooling fluid**

603 Efficient heat transfer is required between the absorber plate, absorber pipes, and then from the pipes to
 604 the cooling fluid. As shown in table 2 and Fig.12, a PVT with a sheet and tube with anti-reflective, low-e
 605 coatings and whole packing lamination manufacturing process collector exhibited 15.10 % electrical and
 606 71% thermal performances whereas those with channelled absorber exhibited 15.40% electrical and 73%
 607 thermal performances, for the no loss. This is due to enhanced tube fin efficiency and improved
 608 conduction between the plate absorber and the pipes.



609

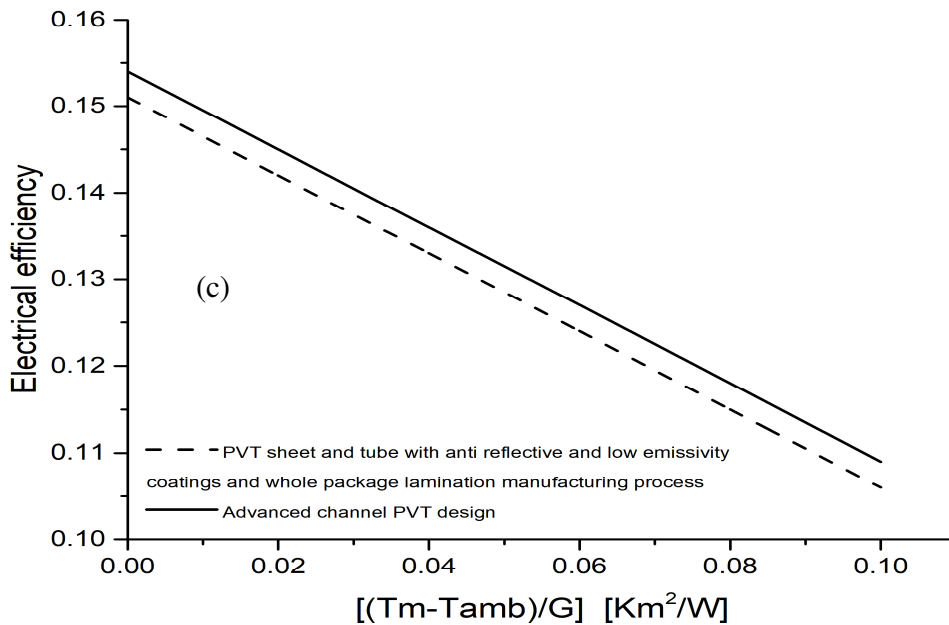
610 **Figure 12-a: The advanced PVT channelled collector considered in the present study**



611

612

613 **Figure 11-b: Predicted thermal efficiency of a PVT module with a sheet and tube absorber compared to one with a**
 614 **channelled absorber. Both configurations include anti-reflective and low-e coatings, and assume whole package**
 615 **lamination.**



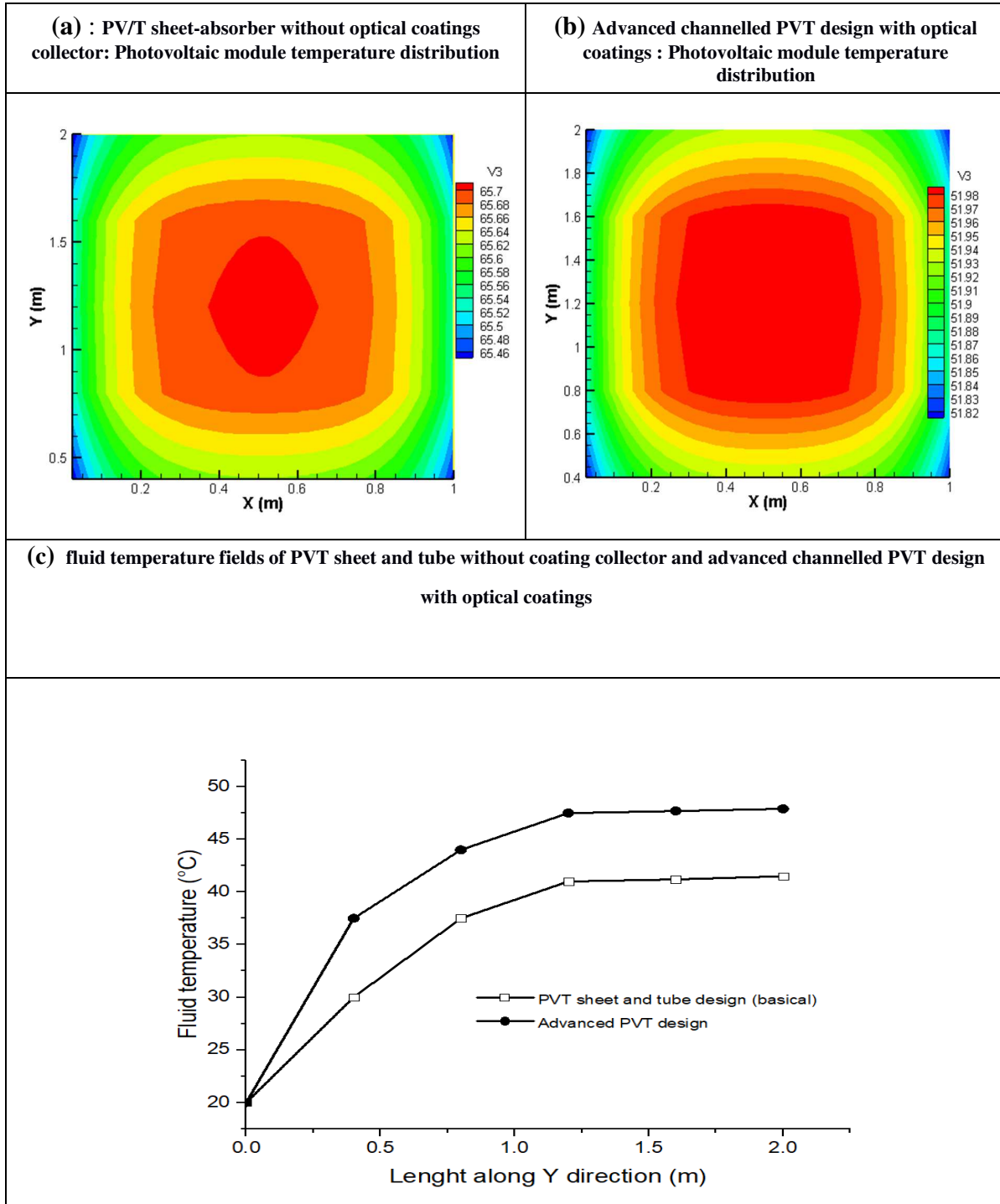
616

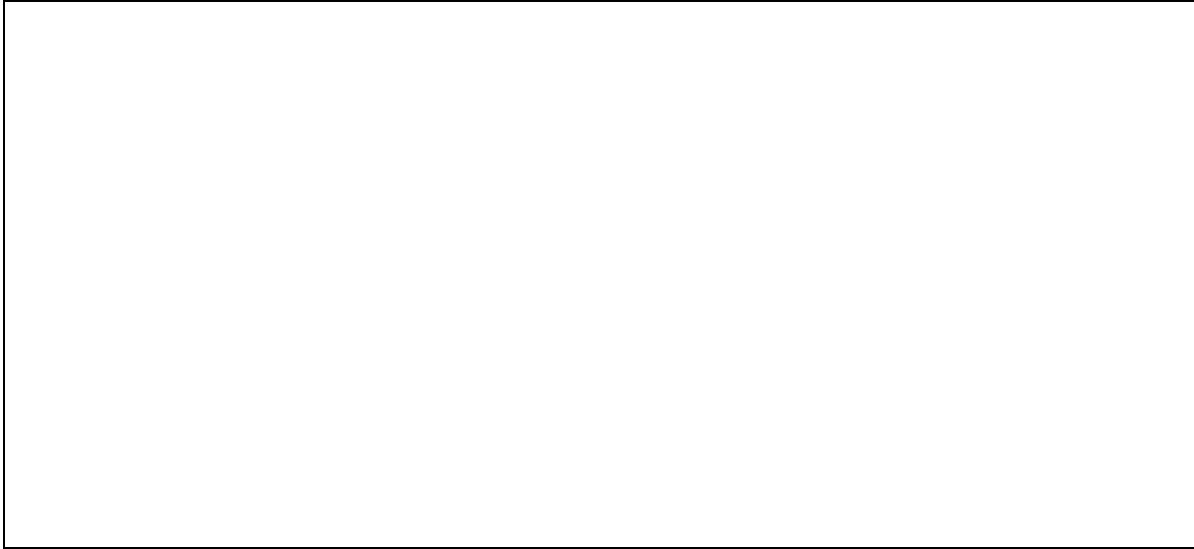
617 **Figure 11-c: Predicted electrical efficiency of a PVT module with a sheet and tube absorber compared to one with a**
 618 **channelled absorber. Both configurations include anti-reflective and low-e coatings, and assume whole package**
 619 **lamination.**

620 **5.2. Temperature distributions**

621 The steady state temperature fields are calculated under nominal environmental conditions for inlet fluid
 622 temperature (20°C), solar radiation (800W/m²), ambient temperature (20°C) and wind speed (1m/s). The
 623 photovoltaic layer and fluid temperature fields of the PVT sheet and tube without the coating collector
 624 and advanced channeled PVT design with optical coatings are presented in Fig. 12. Under the same
 625 conditions, higher PV temperatures are attained for the sheet and tube (between 65.7°C to 65.46°C) than
 626 the PVT with channel design (between 51.98°C to 51.82°C). The fluid temperatures for the two
 627 configurations increase from inlet to outlet with similar trends, although steeper gradients are observed
 628 for the channel design. From inlet to outlet, the fluid temperatures rise from 20°C to 42°C for the sheet
 629 and tube collector (C1) and from 20°C to 48°C for PVT with channels collector (C2). An enhancement of
 630 heat transfers between the photovoltaic module and the absorber is obtained due to the advanced
 631 laminated technique (C2), allowing Low-emissivity, ARC, solar cell, EVA and fluorinated ethylene
 632 propylene (FEP) lamination directly onto the absorber (C2). This results in a lower temperature gradient
 633 between the PV module and the absorber and therefore in a lower PV-cell temperature and a higher
 634 absorber temperature. Thus, good thermal contact between the absorber plate and the fluid provides

635 higher fluid temperatures. Considering the temperature gradient between the photovoltaic cells and water,
 636 it is therefore confirmed that the advanced channelled PVT design with optical coatings operates at higher
 637 fluid temperatures and lower photovoltaic cell temperature. This consequently offers better electrical and
 638 thermal efficiencies.





639

640 *Figure 12: The photovoltaic layer and fluid temperature fields of PVT sheet and tube without coating collector and*
641 *advanced channelled PVT design with optical coatings*

642 **6. Conclusion**

643

644 In this work, a novel PVT collector design is proposed as well as a 2D modeling approach. The detailed
645 PVT numerical model is developed and validated with experimental results available in the literature [44-
646 45]. It allows deepening the analysis and understanding of the impact of new proposed changes of the
647 new PVT compared to the basic PVT collector (sheet and tube heat exchanger). The effects of the optical
648 coating, the thermal resistance between the photovoltaic and absorber plate, the contact area between the
649 absorber and the cooling fluid are investigated and analyzed. The main findings are summarized as
650 follows:

- 651 • The addition of an anti-reflective coating on the photovoltaic module leads to an increase
652 of both the electrical and thermal performances.
- 653 • The thermal efficiency can be improved by the addition of a low-e coating, albeit at the
654 detriment of a reduced electrical efficiency. An optimal value of 0.44 for the emissivity
655 was determined for both the electrical and thermal efficiencies.
- 656 • The conductive heat transfers between the PV module and the thermal absorber is
657 enhanced. As a consequence, lower temperature gradient and therefore lower PV cell

658 temperature and higher absorber temperature are obtained. This leads to an improvement
659 of the thermal and electrical efficiencies.

660 Consequently, the proposed PVT collector design provides the highest fluid temperature and the lowest
661 photovoltaic module temperature compared to the basic one (PVT with sheet and tube absorber). The
662 novel PVT provides 73% and 15.4% electrical and thermal efficiencies, respectively under no loss and
663 standard test conditions (inlet fluid temperature: 20 °C; solar radiation: 800W/m²; ambient temperature:
664 20 °C; and wind speed: 1m/s).

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675

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