Numerical modelling of a PV-T collector working as evaporator in a solar assisted heat pump

E Zanetti¹*, D Del Col¹

¹ Department of Industrial Engineering, University of Padova, via Venezia 1, Padova, Italy
*Corresponding author, e-mail: emanuele.zanetti.1@phd.unipd.it

Abstract. A numerical study of a PV-T collector working as an evaporator for a solar assisted heat pump is presented. The work has been realized in the framework of the SolairHP project for the study of a reversible dual source heat pump. The main objective is to increase the seasonal coefficient of performance of the heat pump compared to an air source heat pump and reduce the overall electrical consumption by including a solar collector-evaporator. A mathematical steady state model of the solar evaporator device has been realized in Matlab environment: it considers the energy and mass fluxes in three dimensions by working a discretization of the main surfaces of the PV sandwich and of the absorber plate (roll-bond plate). The input of the model are the ambient conditions (irradiance components and air temperature), the thermodynamic conditions of the refrigerant at the inlet (enthalpy and mass flowrate) and the circuitation. The main results of the model are the electric and thermal efficiency, the conditions of the refrigerant all along the channels and the temperature distribution on the PV-T surfaces. The numerical model has been used in the design procedure of the solar evaporators to be coupled to a 8 kW heat pump working with R32. A preliminary seasonal analysis on the PV-T evaporator has been realized in order to evaluate the advantage of the dual source system.

1. Introduction

Solar hybrid PV-T collectors are photovoltaic systems where both electrical and thermal energy are generated. They are generally composed of PV cells in contact with channels where a coolant fluid flows and absorbs the generated heat [1]. Hybrid PV-T collectors are already widespread in the market, especially for hot water production applications, and several studies are available in literature concerning this technology [2], [3]. Another application of PV-T collectors is the solar assisted heat pump (SAHP) where the solar radiation is employed as heat source. SAHPs, whose idea was early introduced by [4], can be subdivided in “direct-expansion SAHP” where the refrigerant flows and evaporates inside the solar collector and “indirect-expansion SAHP” [5] where the heat is carried to the heat pump through a water loop. The use of a direct-expansion SAHP can provide a reduction of the number of components and also an improvement of the efficiency compared to other hybrid PV-T systems [6]. In [7] an experimental and numerical study on a direct-expansion SAHP working with R22 is presented: the solar system, based on a six PV modules array coupled with an aluminium plate and copper pipes, works as evaporator on a heat pump producing hot water in a storage tank. The simulations and the experimental tests showed a mean COP value of 3.41 and a maximum value of 7.3 under a global irradiance higher than 600 W/m² and air temperature between 7 and 13 °C. In [8] the effect of irradiance, temperature and mass flow rate of water at the condenser in a solar assisted heat pump has been experimentally investigated. The collector consists of six evacuated tubes containing U-shaped aluminium pipes and solar cells. It was found that the mass flow rate and temperature at the condenser have a low effect on the electrical performance of the system. A solar assisted heat pump prototype working with R22 was realized in [9]. The refrigerant evaporates inside three hybrid PV-T solar collectors composed of roll-bond plates. The two studied geometries showed high refrigerant pressure losses; a good agreement was found between the experimental performance of the system and the numerical results obtained by calculating an efficiency factor as proposed by [10]. Further approaches on the modelling of PV-T evaporators have been presented by [11], which validated their results with the experimental measurements made on a 1.5 kW heat pump working with R134a, and by [12] which developed a transient simulator of the solar collector using CO2 as the refrigerant, focusing on the effect of the
electrical load variations. The performance of a SAHP can be improved by adopting a dual source approach, as reported by [13] due to the inefficiency of the solar evaporator when the irradiance is reduced.

In the present paper, the numerical model of a PV-T evaporator installed in a dual source SAHP working with R32 is presented. The model has been employed to study two possible geometries of the evaporator and to estimate the performance of the system to be installed in Padova, Italy.

2. System description

The objective of the numerical model presented in this work is to simulate the operation of a PV-T collector working as an evaporator in a dual source heat pump whose simplified layout is displayed in Figure 1. In the refrigerant loop, the compressor sends the fluid to the condenser where hot water is produced and sent to a thermal storage. After the throttling process inside the expansion valve, the refrigerant enters the evaporator and it finally returns to the compressor. In this case, two evaporators are arranged in parallel after the expansion valve: a finned coil evaporator and the hybrid PV-T collector. The 3-way-valve enables the refrigerant to flow into a single device, thus the control logic of the system should be based on the performance comparison of the heat pump working with the finned coil or with the solar collector as the evaporator. It is expected that when the irradiance is high, the solar collector permits the refrigerant to evaporate at higher evaporation temperature than that in the finned coil, leading to higher heat pump performance. When the irradiance is low or absent, the PV-T performance drops and the finned coil must be selected as evaporator. The correct choice of the heat source is an important aspect of this type of systems as it will be discussed in the next sections.

Figure 1. Schematic of the dual source SAHP working with a finned coil and a hybrid PV-T evaporator whose layout is displayed on the right.

The PV-T collector studied and modelled here is composed of a commercially available photovoltaic module consisting of ten strings of six cells in poly-crystalline silicon (peak power of 280 Wp) embedded in a EVA/tedlar structure and a roll-bond absorber applied on the backsheet. An additional insulation layer can be positioned on the bottom of the absorber (Figure 1). The roll-bond is a widely used technology that has been proven to be competitive in solar thermal applications [14]. Furthermore, roll bond absorbers can also be used for hybrid PV-T collectors because of their flexibility on the choice of channels arrangement [2] and this is a key feature for the study of the temperature distribution on the PV cells surface, especially with phase-change fluid flowing inside the collector. The main disadvantage of roll-bond absorbers is the limit to the system pressure: following the guidelines by the roll-bond manufacturers, 17 bar was selected as maximum operative pressure of the refrigerant inside the channels.
For R32, the corresponding saturation temperature is about 25°C. The area of the absorber is 1.46 m², the thickness of the aluminium plates is 0.75 mm and the channels section is “One Side Flat” meaning that only one of the roll-bond sheets has been inflated (the channel width is about 8 mm).

3. Numerical model of the PV-T evaporator

3.1. Discretization

A numerical model of the PV-T evaporator has been developed in Matlab® environment. The objective is to study the behaviour of different absorber circulations, therefore steady state conditions have been here considered. During the two-phase flow of the refrigerant, the heat transfer coefficient varies over a wide range, thus a 1D approach could not be applied for the modelling. A distributed parameter model has been realized where the various layers of the PV-T have been discretized into small elements, therefore the heat transfer has been considered along three dimensions. In addition, this approach allows calculating the temperature distribution on the PV cells. The inputs of the model are the thermodynamic conditions of the refrigerant at the inlet of the evaporator (enthalpy \( h_{\text{in}} \), temperature \( T_{\text{in}} \), and mass flow rate \( m_f \)), the external conditions (irradiance \( G \), air temperature \( T_{\text{air}} \)) and the channels arrangement. Each modelled layer (glass layer, cell layer, absorber layer) is described by a matrix whose components represent the discretized elements where the energy equation is solved. Depending on the geometry of the absorber, some of these elements are located in the area where the channels for the refrigerant are placed, in this case also mass and momentum equations are solved. In the following sections, the balance equations for each layer and the fluid flow modelling are presented more in detail.

3.2. Equations for each layer

The implemented equation for a glass layer of area \( dx \cdot dy \) can be written as:

\[
G \tau - \left( \frac{h_{\text{loss,top}}}{(T_g - T_{\text{air}})} \right) \left( \frac{T_g - T_{\text{pv}}}{\lambda_g} + \frac{T_{\text{pv}}}{\lambda_{\text{eva}}} + \frac{1}{\varepsilon_{\text{pv}}} \right) \left( \frac{\partial^2 T_g}{\partial x^2} + \frac{\partial^2 T_g}{\partial y^2} \right) = 0
\]

(1)

where the subscripts “g”, “eva”, “pv” refer to the glass, PV cell and EVA layers, \( l \) is the width, \( \lambda \) is the thermal conductivity, \( \varepsilon \) is the emittance and \( \sigma \) is the Stefan Boltzmann constant. The first term represents the radiation absorbed by the glass, considering the different irradiance components following the procedure reported in [15]. The second term represents the thermal losses, where:

\[
h_{\text{loss,top}} = h_{\text{conv,top}} + h_{\text{rad,top}}
\]

(2)

In equation (2) \( h_{\text{conv,top}} \) is the convective heat transfer coefficient calculated with the Watmuff equation [16] and \( h_{\text{rad,top}} \) is the radiative heat transfer coefficient depending on the glass emittance, the tilt angle of the collector and the equivalent sky temperature calculated with the Whillier equation. The last two terms of equation (1) represent respectively the heat transferred to the PV layer \( (q_{\text{trans}}) \) and exchanged with the adjacent glass elements.

The equation implemented for a cell layer of area \( dx \cdot dy \) is:

\[
G \tau + q_{\text{trans}} - \frac{T_{\text{pv}} - T_{\text{abs}}}{\lambda_{\text{pv}}} + \left( \frac{q_{\text{trans}}}{\lambda_{\text{pv}}} \frac{\partial^2 T_{\text{pv}}}{\partial x^2} + \frac{q_{\text{trans}}}{\lambda_{\text{pv}}} \frac{\partial^2 T_{\text{pv}}}{\partial y^2} \right) - P_{\text{el}} = 0
\]

(3)

where the subscripts “tedlar” and "abs" refer to the tedlar and roll-bond absorber layers. The first term represents the radiation transmitted by the glass considering the different irradiance components following the procedure reported in [15]. The third and fourth term represent respectively the heat absorbed by the roll-bond plate \( (q_{\text{abs}}) \) and exchanged with the adjacent cell elements. Finally, \( P_{\text{el}} \) is the electric power generated by the cell element. For each PV cell (identified in the model by a certain number of discretizations), it is possible to calculate the I-V curve by the application of a single diode
modelling approach ([15]) depending on the irradiance and on the maximum cell temperature. Therefore, voltage, current and power depend on the cell arrangement (series, parallel) and on the electric load (fixed resistance or MPPT system).

The equation implemented for a roll-bond layer discretization of area $dx \cdot dy$ is:

$$
q_{ab} - h_{loss,bottom}(T_{abs} - T_{air}) + \left( \lambda_{abs} l_{abs} \frac{\partial^2 T_{abs}}{\partial x^2} + \lambda_{abs} l_{abs} \frac{\partial^2 T_{abs}}{\partial x^2} \right) = 0
$$

(4)

where the second term represent the heat losses: the heat transfer coefficient $h_{loss,bottom}$ can be written as the contribution of a radiative term and convective term (as in equation (2)) or as a conductive term in the case of an insulating layer on the bottom. The last term is the heat exchanged with the adjacent roll-bond elements.

3.3. Fluid flow modelling

In the case of discretized elements associated to the refrigerant channels, an additional term ($q_f$) is added to equation (3), representing the heat transferred to the fluid:

$$
q_f = K(T_{abs} - T_f)
$$

(5)

where $K$ is the global heat transfer coefficient and $T_f$ is the refrigerant temperature inside the channel element. The fluid temperature depends on the heat transferred $q_f$ in the preceding circuit element and on the pressure (thermodynamic conditions of the refrigerant calculated with Refprop[17]). The refrigerant heat transfer coefficient during evaporation is calculated with the Liu and Winterton correlation [18] and for the single-phase flow the Gnielinski correlation [19] has been implemented. Regarding the pressure drop, for each channel element, the contribution of gravity losses, bend losses and friction losses are considered: the gravity term takes into account the tilt angle of the module and the flow direction (upwards or downwards), and the bend losses have been modelled following the description of Paliwoda [20]. The friction factor for the two-phase flow has been calculated with Friedel equation [21] (Rohuani equation for the void fraction [22]). As already mentioned, the circuit elements can be detected by a numerical value and thus even complex channel patterns can be easily reproduced and simulated by the model. Furthermore, this approach allowed to apply an electric–hydraulic analogy to calculate the mass flow rate in every element depending on the pressure change.

The system of nonlinear equations is solved for all the elements of the collector and an iterative cycle is implemented in order to obtain the temperature distribution over all the surfaces and the mass flow rate. The cycle stops when the sum of the heat/reflection losses, the heat transferred to the fluid and the generated electrical power minus the total incident radiant flux is equal or less than a tolerance value.

4. Simulation results

The model is applied here to compare different circuit geometries. The circuit arrangements (Geometry#1 and Geometry#2) are displayed in Figure 2. In the simulations the insulation layer has not been considered, because in the summer period, when the collector is not supposed to work as an evaporator, the cell temperature can be lower without insulation and thus the electric efficiency can benefit.
Figure 2. Channels arrangement of the two simulated configurations (courtesy of CGA Technologies): Geometry#1 (left) and Geometry#2 (right). Sections A-A are marked as reference positions for Figure 4.

4.1. Performance comparison

In each of the two cases of Figure 2, the layout is composed by six main circuits that evolve from the inlet channel and are arranged in parallel in a serpentine-based configuration. Close to the outlet channel, another branching zone is created. Additionally, across the entire circuit, some by-pass channels are placed in order to uniform the pressure field of the absorber.

In Figure 3 the temperature distribution over the PV cells obtained from the model (Sec. 3) is shown for the two geometries when the global irradiance on the collector surface is equal to 800 W/m$^2$, air temperature is 5°C and the evaporation temperature is 5°C. The refrigerant mass flow rate is 2.8 g/s in both cases and the quality at the outlet is found to be approximatively 0.95. The results show that the cell temperature is significantly affected by the channels arrangement.

Figure 3. Temperature distribution on the surface of the PV module calculated by the model in case of Geometry#1 (left) and Geometry#2 (right). Simulation conditions: $G = 800$ W/m$^2$, $T_{\text{air}} = 5$ °C, $T_e = 5$ °C.
For the Geometry#1, the zone covered by the channels is maintained at a uniform temperature whereas for Geometry#2 where the channels are more spaced, some hot spots appear. For both geometries, the cell temperature over the more external area, close to the branched zone, is at higher temperature. This is due to the fact that the refrigerant in the external circuits fully evaporates before than in the others: this phenomenon is very pronounced in Geometry#1 where the cell temperature close to the right border of the module is higher. This is confirmed by Figure 4 which shows the vapour quality values evaluated in the six circuits of the section A-A of Figure 2, located at the last passage of the serpentine. Geometry#2 circuit arrangement is better balanced than that of Geometry#1 where the mass flow rate at the external circuits (higher pressure drop) is lower and thus complete evaporation is reached earlier.

\[ T^* = \frac{(T_{mf} - T_{air})}{G} \]  

(6)

Here \( T_{mf} \) is the mean fluid temperature inside the collector. In the case of a PV-T evaporator, \( T_{mf} \) can be replaced by the evaporation temperature \( T_e \); the efficiency curve for Geometry#2 has been numerically derived and it is depicted in Figure 5. The thermal efficiency curve can be a useful tool because it allows to estimate the evaporation temperature of the refrigerant that is necessary to obtain a desired output when the inlet conditions (mass flow rate and enthalpy) and the environmental conditions \((T_{air}, G)\) are fixed. The following equation must be solved:

\[ \frac{\dot{m}_f (h_{in} - h_{out})}{GA} = f \left( \frac{(T_e - T_{air})}{G} \right) \]  

(7)

Here \( A \) is the area of the collector, \( h_{out} \) is the outlet desired enthalpy and the right-handed side expression is a function representing the efficiency curve obtained by the model simulations. It must be noted that even if the output is fixed (i.e. superheating = 5 K), \( h_{out} \) depends on the evaporation temperature thus an iterative cycle has been implemented to solve the equation. A preliminary estimation of the evaporation temperature inside the PV-T evaporator installed on a 8 kW SAHP has been done by using the hourly environmental conditions (air temperature and solar irradiance) of the city of Padova.
(45.4°N, 11.9°E, data provided by ARPAV). The calculation has been realized assuming to obtain the entire vaporization (SH = 0 K) of R32 flowing inside a variable number of collectors (5 to 20) arranged in parallel with 40° tilt angle (solar irradiance components calculated following [24]). A maximum value of total mass flow rate (equal to 0.028 kg/s) has been fixed in order to approximate the condenser capacity (at 50°C condensation temperature and 2 K subcooling). Since the irradiance input varies during the day, it is assumed that the heat capacity can be varied in the heat pump by changing the refrigerant mass flow rate, with capacity control in the range between 30% and 100% of the nominal power. The resulting evaporation temperature was compared to the hypothetical evaporation temperature of a finned coil evaporator, which for sake of simplicity is assumed here equal to (T_{air} - 5 K).

Figure 6 displays the percentage of hours when the evaporation temperature of the PV-T evaporator can be higher than that of the finned coil evaporator and lower than 25°C (maximum admitted saturation temperature). The total amount of hours refers to the heating season, when the solar irradiance on the collectors’ plane allows to completely evaporate the minimum refrigerant mass flow rate. The results show that the percentage of hours when it is possible to work with the PV-T module as the evaporator increases with the number of panels and the maximum value is reached when operating with 9 panels. A further increase of number of collectors leads to the increase of the time periods with evaporation temperature higher than the imposed limit of 25°C, thus a decrease of the possible operating hours of the collector.

**Figure 5.** Efficiency curve of the PV-T evaporator depending on the reduced temperature calculated by the simulations on Geometry #2.

**Figure 6.** Operating time of the PV-T evaporator. Percentage of hours when the evaporation temperature inside the PV-T is (T_e > T_{air} -5 K) and (T_e<25°C).

5. Conclusions

The numerical model of a PV-T solar collector working as an evaporator in a heat pump has been developed. In the model a distributed parameter approach has been applied in order to find the temperature distribution on the PV surface taking into account the positions of the refrigerant channels. The simulations allowed to compare two possible circuit geometries, based on roll-bond technology, showing the effect of channels arrangement on the cell temperature: the numerical results proved that a further optimization work must be realized in order to balance the system. Finally a preliminary seasonal analysis on the number of PV-T collectors to be installed on a dual source SAHP has been realized based on the efficiency curve of the panel obtained by the simulations. It can be shown that installing more than 9 collectors on a 8 kW heat pump in Padova working with R32 is not convenient due to the high resulting evaporation temperature and thus high refrigerant pressure in the evaporator.
6. References

[1] Zondag HA, de Vries DW, van Helden WGJ, van Zolingen RJC, van Steenhoven AA. The yield of different combined PV-thermal collector designs. *Solar Energy* 2003;74:253–69.

[2] Aste N, del Pero C, Leonforte F. Water flat plate PV-thermal collectors: A review. *Solar Energy* 2014;102:98–115.

[3] Besheer AH, Smyth M, Zacharopoulos A, Mondol J, Pugliese A. Review on recent approaches for hybrid PV/T solar technology. *International Journal of Energy Research* 2016;40:2038–53.

[4] Sporn P, Ambrose ER. The heat pump and solar energy. Proc. world Symp. Appl. Sol. energy, Phoenix: 1955, p. 1–5.

[5] Chow TT, Pei G, Fong KF, Lin Z, Chan ALS, He M. Modeling and application of direct-expansion solar-assisted heat pump for water heating in subtropical Hong Kong. *Applied Energy* 2010;87:643–9.

[6] Ji J, He H, Chow T, Pei G, He W, Liu K. Distributed dynamic modeling and experimental study of PV evaporator in a PV/T solar-assisted heat pump. *International Journal of Heat and Mass Transfer* 2009;52:1365–73.

[7] Ji J, He H, Pei G, He W, Liu K. Distributed dynamic modelling with experimental validation on a photovoltaic solar-assisted heat pump. *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy* 2008;222:443–54.

[8] Chen H, Riffat SB, Fu Y. Experimental study on a hybrid photovoltaic/heat pump system. *Applied Thermal Engineering* 2011;31:4132–8.

[9] Ito S, Miura N, Takano Y. Studies of heat pumps using direct expansion type solar collectors. *Journal of Solar Energy Engineering, Transactions of the ASME* 2005;127:60–4.

[10] Ito S, Miura N, Wang JQ, Nishikawa M. Heat pump using a solar collector with photovoltaic modules on the surface. *Journal of Solar Energy Engineering, Transactions of the ASME* 1997;119:147–51.

[11] Tsai HL. Modeling and validation of refrigerant-based PVT-assisted heat pump water heating (PVTA-HPWH) system. *Solar Energy* 2015;122:36–47.

[12] Paradis PL, Rousse DR, Lamarche L, Nesreddine H. A hybrid PV/T solar evaporator using CO2: Numerical heat transfer model and simulation results. *Solar Energy* 2018;170:1118–29.

[13] Deng W, Yu J. Simulation analysis on dynamic performance of a combined solar/air dual source heat pump water heater. *Energy Conversion and Management* 2016;120:378–87.

[14] Del Col D, Padovan A, Bortolato M, Dai Prè M, Zambolin E. Thermal performance of flat plate solar collectors with sheet-and-tube and roll-bond absorbers. *Energy* 2013.

[15] Duffie JA, Beckman WA. Solar Engineering of Thermal Processes: Fourth Edition. 2013.

[16] Watmuff J, Proctor D. Solar and wind induced external coefficients - Solar collectors. *Cooperation Mediterraneenne Pour l'Energie Solaire, Revue Internationale d'Heliotechnique, 2nd Quarter* 1987.

[17] Lemmon EW, Huber ML, McLinden MO. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties (REFPROP), Version 9.0. *Physical and Chemical Properties* 2010.

[18] Liu Z, Winterton RHS. A general correlation for saturated and subcooled flow boiling in tubes and annuli, based on a nucleate pool boiling equation. *International Journal of Heat and Mass Transfer* 1991;34:2759–66.

[19] Gnielinski, V. New Equations for Heat and Mass Transfer in Turbulent Flow Through Pipes and Ducts. *Forschung Im Ingenieurwesen* 1975;41:1975.

[20] Paliwoda A. Generalized method of pressure drop calculation across pipe components containing two-phase flow of refrigerants. *International Journal of Refrigeration* 1992;15:119–25.

[21] Friedel L. Improved friction pressure drop correlations for horizontal and vertical two phase pipe flow. *Rohre - Rohrleitungsbau - Rohrleitungstransport* 1979;18, July:485–91.

[22] Rouhani SZ, Axelsson E. Calculation of void volume fraction in the subcooled and quality boiling regions. *International Journal of Heat and Mass Transfer* 1970;13:383–93.

[23] Del Col D, Dai Prè M, Bortolato M, Padovan A, Industriale I. Investigation of Pv Solar Devices for Electricity Production and Heat Recovery, 67th Congr. Naz. ATI, Trieste: 2012, p. 11–4.

[24] Padovan A, Del Col D. Measurement and modeling of solar irradiance components on horizontal and tilted planes. *Solar Energy* 2010;84:2068–84.

Acknowledgments
The authors acknowledge CSEA (Cassa per i Servizi Energetici e Ambientali) for the support through the project CCSEB_00075 - SOLAIR-HP. CGA Technologies Srl is also acknowledged for the support in the design of the roll-bond geometries.