EXPERIMENTAL STUDY ON HEAT TRANSFER FROM AMBIENT AIR IN ROW-INSTALLED PHOTOVOLTAIC-THERMAL (PVT) SOLAR COLLECTORS ON A FLAT ROOF

Summary

With climate change urging the transition to renewable energy sources, Photovoltaic-Thermal (PVT) solar collectors emerge as a promising decarbonized solution producing both electricity and heat. Our study evaluates the performance of PVT panels designed to be as the sole heat source of a brine to water Heat Pump (HP) system for buildings thermal needs. By focusing on the impact of the number of PVT panels in a single row on ambient air heat gain, experimental data reveal a significant influence of panel configuration on the heat exchange coefficient, with a noticeable decrease in heat loss as the number of panels increases, highlighting the critical role of PVT panel arrangement in system performance enhancement.

Keywords: Photovoltaic-Thermal (PVT), solar collector, Heat Pump (HP), heat transfer

1. Introduction

With the urgent need for decarbonized thermal solutions to reduce the scale of ongoing climate change, Photovoltaic-Thermal (PVT) solar collectors stand out for their dual role of producing renewable electricity and heat within a single component. They present a performant alternative as the sole heat source of a brine/water Heat Pump (HP) for thermal needs of buildings including mainly space heating (SH) and Domestic Hot Water (DHW). Indeed, in (Chhugani et al., 2023), it has been highlighted that this combination of PVT and HP with a well sized hot buffer storage and a floor heating can achieve higher overall energy performance than a reference system made up with the same area of PV panels and an air/water heat pump. In the terminology of solar-assisted heat pumps as reminded in (Jonas, 2023), this system configuration is classified as an “indirect” (the refrigerant fluid does not circulate in the panels) and “serial” (the PVT panels provide heat to the heat pump evaporator) combination.

The most critical feature in such system is the capability of PVT panels to recover heat from the ambient air. In (Jaafar et al., 2022), it has been shown that the higher this capability is, the more performant the system is. Indeed, when there is no irradiation (very cloudy day or night), PVT panels act solely as air/water heat exchangers. Then, numerous parameters, such as the distance from the panel to the roof, the roof pitch, the mounting system as well as the orientation of the installation in relation to the prevailing winds in the region, affect this capability.

This paper focuses on the impact of the number of PVT panels as a single row on the *ambient air heat gain* performance of the field. The end goal of this study is to provide valuable insights which can be used for optimizing the PVT field design to enhance the performance of a PVT-HP system.

2. Methodology

2.1. PVT panel prototype

Each of the PVT panels tested is made up with a standard PV panel with a mini-channel flow distribution heat exchanger mounted behind it (see Fig. 1). They were installed in a row in Z configuration (see Fig. 3).

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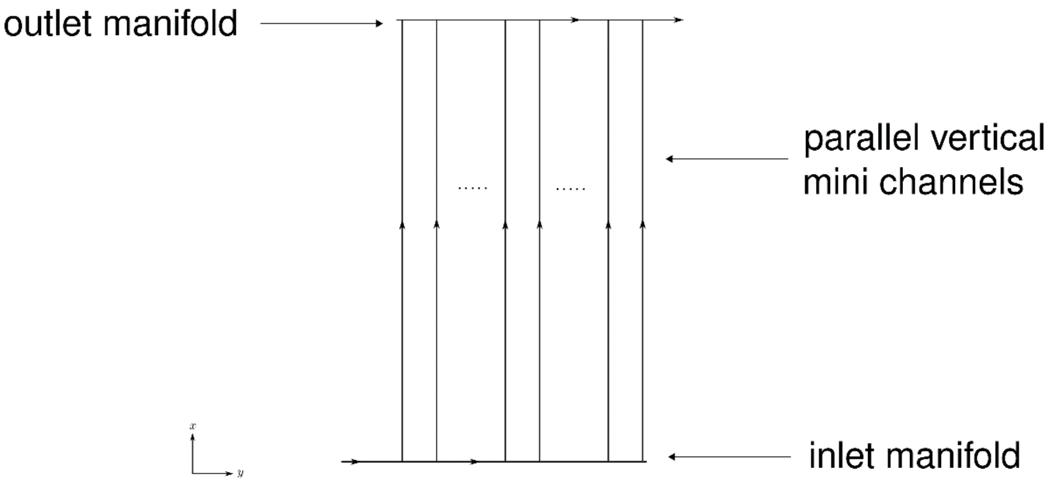

Fig. 1: Scheme of the tested PVT panel prototype

2.2. Test bench

The experimental set-up used for this study is shown in photo (see Fig. 2) and described on the diagram below (see Fig. 3). It consists mainly of a water tank filled with a mixture of water and glycol, whom temperature profile is estimated using two temperature probes installed in two immersion sleeves at the top and the bottom. For the PVT thermal power measurements, a flowmeter ($\dot{V}_{PVT}$) and two temperature probes are installed at the inlet ($T_{PVT,in}$) and the outlet ($T_{PVT,out}$). The overall pipes are insulated. The fluid circulation is insured using a water circulator which is controlled by a regulator through a PWM signal to ensure volume flowrate within a given range in the row of 1 to 8 PVT panels installed on a flat roof (see Fig. 1b). Of course, for each configuration, the flow rate in the row was set so that the flow rate per panel remained roughly the same. The regulator also controls an electrical heating resistance installed in the middle of the water tank with a hysteresis control strategy based on the temperature on its top side. Thus, the fluid temperature entering the PVT field is higher than the ambient air temperature, so the measured heat output is negative.

The assumption made here is that the heat loss (measured here) is equivalent to the heat gain (needed for HP combination) for the same absolute temperature difference between the PVT panel heat exchangers (in average) and the ambient air ($T_{PVT,m} - T_{amb}$). In theory, when free convection dominates, the movement of air around the panels is not symmetrical from one situation to another as it is described in (Incropera et al., 2013) for a hot flat plate and a cold flat plate. However, we assume that the potential variations of heat loss/gain by installing several panels in a row are similar. Moreover, it is important to note that the radiative heat transfers, from the sky and from behind the panel, are considered invariable from one panel to another so that the measured variations are only linked to the variations of convective heat transfer with ambient air (or of temperature distribution over the heat exchangers).

Fig. 1: (a) Photo of the test bench technical room (b) Photo of the row of PVT panels on the flat roof

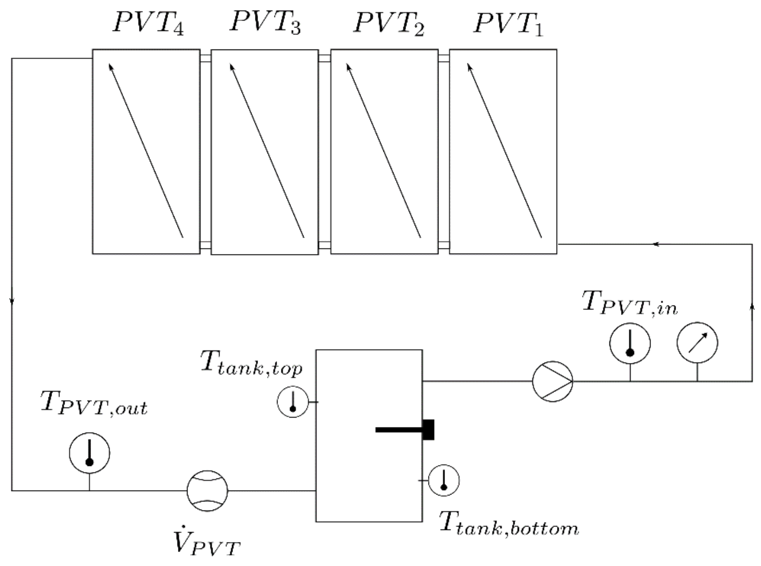

Fig. 3: Test bench diagram

2.3. Data filtering

When the electrical heating resistance of constant power (set before the test between 400 W and 2 kW depending on expected power in the coming conditions) switches off, inlet temperature drops drown then the resistance switches on again. We removed the data corresponding to this phenomenon with a 20-minute window around the local minimum inlet temperature (see grey boxes on Fig. 3). We also removed the first 10 minutes of each test sequence to avoid transient effects. Then we selected data with $G < 1\ W/m^2$ (night) and with flow rate in expected range. Finally, we resampled the data at a time step of 5 minutes and selected those that verify $T_{PVT,m} - T_{amb} \in [6\ ;\ 12]\ K$ and $u \in [0\ ;\ 3]\ m/s$.

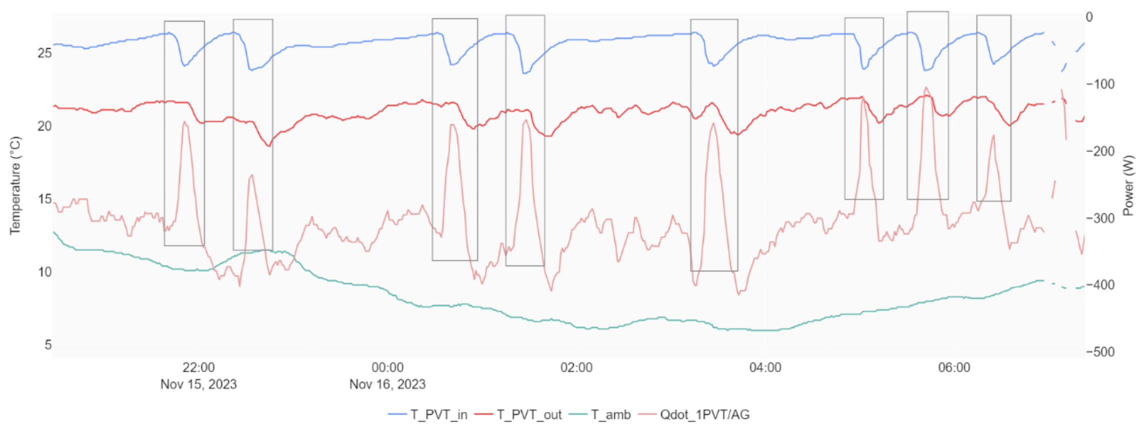

Fig. 4: Time series graph of a test night

2.4. Data analysis

As we study PVT panels heat gain/loss with their environment, the experiments were performed during the night. Then, for unglazed PVT technology, it is relevant to linearize the heat output of the panels in relation to the difference between the average temperature of the heat exchanger and the ambient temperature:

$$\dot{Q} = (\rho \dot{V}_{PVT}) C_p (T_{PVT,out} - T_{PVT,in}) = \mathrm{A}_1(u)\,(T_{PVT,m} - T_{amb}) \qquad \text{(eq. 1)}$$

ρ stands for the density of the heat transfer fluid and C_p for its calorific capacity (here MPG 40%). By disregarding the influence of second and fourth order terms in $(T_{PVT,m} - T_{amb})$, by including the radiative heat transfer in A_1, so that we set the a_4 and a_7 coefficients relating to $G' = E_L - \sigma {T_{amb}}^4$ to 0, and by defining $A_1(u)$ as $a_1 + a_3(u-3)$, we achieve the simplified version of the ISO 9806:2017 standard equation described in ("ISO/DIS 9806:2017 Solar energy — Solar thermal — Test methods," 2017).

In the following results section, we will present the results with two different approaches:

- Linearize $A_1(u)$ in this form: $a_1 + a_3(u-3)$ and get a_1 and a_3 coefficients with a linear least squares method (LLSQ).
- Find $A_1(u_m)$ for each wind range $[u_m - 0.1\ ;\ u_m + 0.1]$ for $u_m \in [0.1, 0.3, 0.5, 0.7\ldots]$

3. Results

3.1. Least squares method

We implemented a classic linear least squares method with the *linalg.lstsq* function of *NumPy* Python library to get a_1 and a_3 coefficients. We also performed 10,000 bootstrap resampling. For each bootstrap sample, we fitted the $a_1$and a_3 coefficients. By resampling our dataset with replacement, we generate a comprehensive distribution of coefficients estimates, allowing us to provide a quantitative measure of uncertainty: the standard deviation of $A_1(u)$ calculated with equation 2 and represented by the coloured band on Fig. 3.

$$\sigma_{A_1} = \sqrt{{\sigma_{a_1}}^2 + (u-3)^2{\sigma_{a_3}}^2} \qquad \text{(eq. 2)}$$

As it is shown in figures 5 and 6 below, this multiple linear regression works well to estimate the thermal output of the PVT panels row.

Fig. 5: Timeseries graph of test sequences for 6 panels in a row with the thermal output per collector gross area "Qdot_1PVT/AG" and the modelled one "Qdot_1PVT/AG_model"

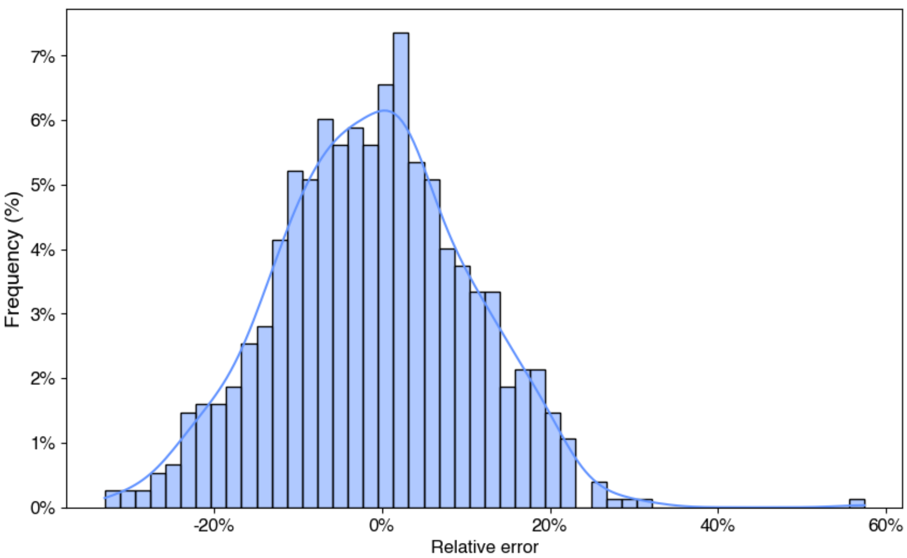

Fig. 6: Relative error between modelled and measured thermal output for 6 panels in a row

Finally, we compared the results from 1 to 8 panels in a row:

Tab. 1: Results obtained with least squares method on filtered data for 1 to 8 panels

Number of panels	Mean total flow rate	Mean flow rate per panel	Hours of data	b_1	a_3	MAE	MAE out of mean power
Unit	L/h	L/h	h	W/m^2K	W/m^2K $/(m/s)$	W	%
1	143	143	4.7	22.6	6.8	21	7%
2	220	110	26.0	19.8	5.0	22	9%
4	420	105	18.6	19.2	4.2	21	9%
6	704	117	62.3	15.3	6.0	14	9%
8	1013	127	28.1	13.9	6.4	8	7%

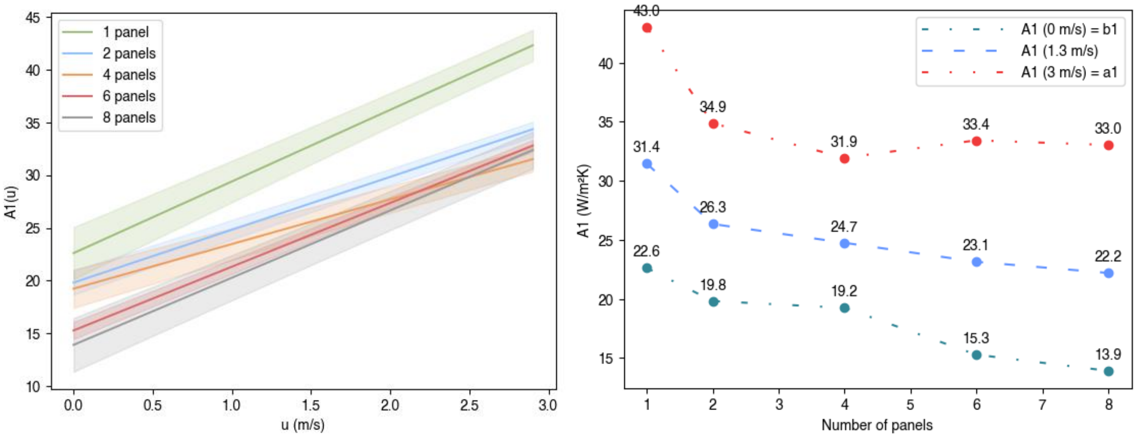

Fig. 7: Results obtained with least squares method on filtered data for 1 to 8 panels

In our test facility configuration, the obtained results show:

- For $u = 0\ m/s$, the heat loss coefficient ($b_1$in W/m^2K) decreases from 1 to 8 panels installed in the row. 2 and 4 panels have a performance decrease of about 15% compared to 1 panel.
- For $u = 1.3\ m/s$, there is a significant reduction from 1 to 2 panels (-16%) and then the reduction is roughly linear with a slope of $-0.8\ W/m^2K/panel$.
- For $u = 3\ m/s$, there is a reduction between 2 panels and 1 panel and then the heat loss coefficient looks stable until 8 panels.

3.2. Clustering by wind range

In this section, we cluster data by wind ranges $[u_m - 0.1\ ;\ u_m + 0.1]$ for $u_m \in [0.1, 0.3, 0.5, 0.7\ \dots]$ of for each cluster, a linear fit provided the corresponding $A_1(u_m)$ coefficient.

This analysis shows that we should be careful when comparing "performance" based on linearization of $A_1(u)$. It can lead to misinterpretation. Thus, we find that the A_1 values for wind speed higher than 2 m/s (resp. 1 m/s) are not relevant for the configuration with 6 panels in a row (resp. 8 panels in a row) because of a lack of data.

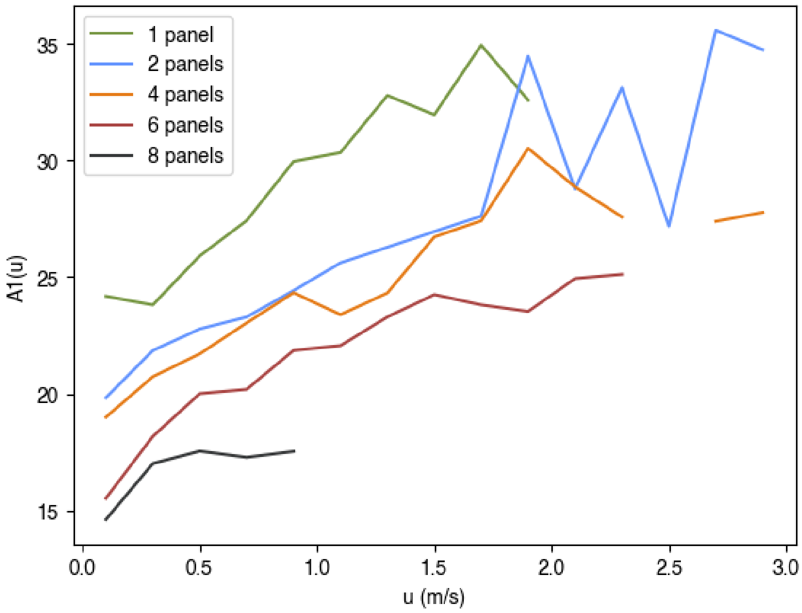

Fig. 8: $A_1(u_m)$ in each configuration

4. Discussion

4.1. Experimental bias

We should keep in mind that, due to the experimental set-up, the more panels there are, the lower the difference in temperature between the average on the exchangers and the ambient air (see Fig. 9). We can't rule out the hypothesis that this is at the root of some of the degradation of the heat loss coefficient by increasing the number of panels in the row.

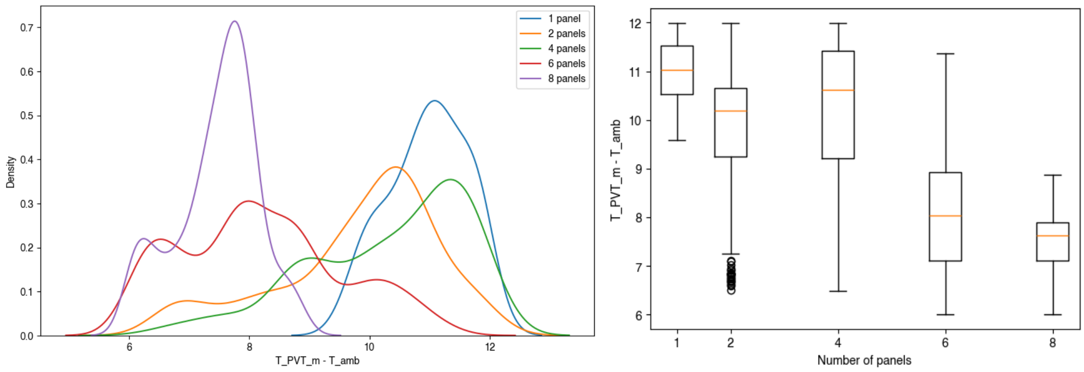

Fig. 9: (a) Distribution of T_PVT_m – T_amb in each configuration
(b) Boxplots of these distributions in each configuration

4.2. Flow distribution

Due to these PVT panels prototypes design, it is highly possible that the flow distribution would be uneven between them when installed in a row (in Z configuration). In (García-Guendulain et al., 2020), this kind of flow nonuniformity and thermal imbalances in solar collectors of similar design was studied. So far, we cannot quantify this nonuniformity in our experimental set-up, but it can be a source of reduction in thermal performance. Indeed, for PVT panels in which the flow rate would be too low, the mean heat exchanger temperature would be closer to ambient air temperature so that heat transfer would be lower.

We tried to measure the flow rate in each panel in this configuration and for each number of panels. To do this, we positioned flowmeters between consecutive panels at the inlet manifold. These experimental results confirm that there may be a significant flow distribution imbalance in this PVT panels row general design (manifolds and parallel mini-channels) but do not give any information on the original set-up (see Fig. 10). Indeed, the flow meters cause sharp drops in pressure in the inlet manifold so that the profile is reversed compared with what is expected in such a Z configuration (the highest flow rates should be in the last panels): the hydraulic behaviour with flowmeters is significantly different from the configuration without.

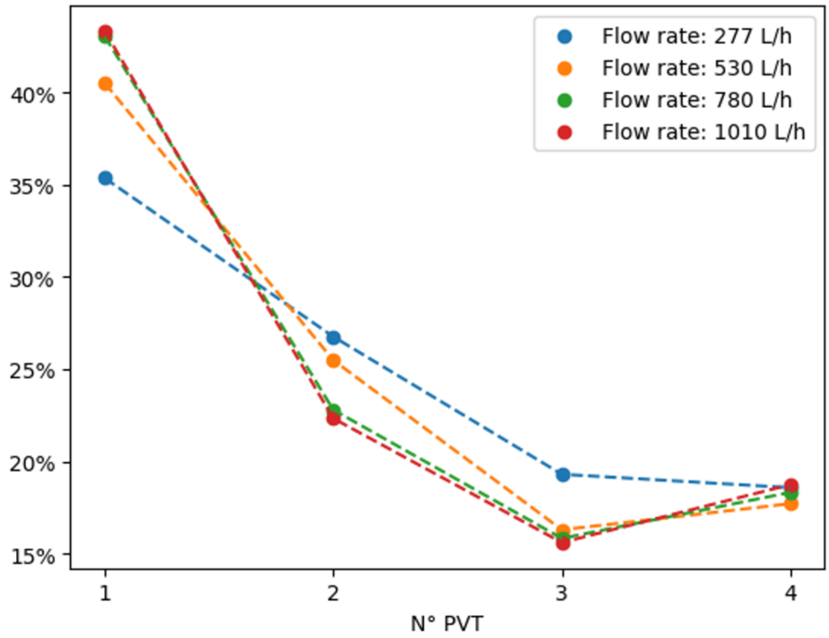

Fig. 10: measured flow distribution for 4 panels in a row in Z configuration with flowmeters between consecutive panels

4.3. Row geometry

The third possible explanation is the impact of the overall geometry on the shape and the temperature field of the airflow around the panels (including edge effects on the sides of the row corresponding to higher heat transfer).

5. Conclusion

Our research underscores the pivotal role of PVT panel configuration in enhancing the thermal performance of PVT-HP systems. The experimental analysis reveals that increasing the number of PVT panels in a single row significantly reduces the heat loss coefficient, thereby affecting overall system efficiency. This finding is crucial for the design and optimization of PVT fields, suggesting that careful consideration of panel arrangement can lead to substantial gains in energy performance. Besides, quantify the possible contribution of the heterogeneous flow distribution in the mini-channel heat exchangers is the subject of ongoing modelling and experimental work.

6. References

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