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

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Summary

With climate change urging the transition to renewable energy sources, Photovoltaic-Thermal (PVT) solar collectors emerge as a promising solution producing both renewable electricity and heat for buildings. Our study evaluates the performance of PVT panels designed to be as the sole heat source of a brine/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 that the row's configuration can have a significant impact on their thermal performance.

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 slightly 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: it is called PVT-IDX-SAHP.

The most critical feature in such system is the capability of PVT panels to collect 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, PVT panels act mainly as air/water heat exchangers. Then, night thermal performance of PVT panels depends not only on their individual design but also on where and how they are installed: 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. When panels are connected in a row, the airflow dynamics around them may change significantly compared to individual panels and then affect overall convective heat transfer of the PVT field. Moreover, when panels are connected in parallel in a row, uneven flow distribution can reduce performance. It is difficult to separate the two causes, so this paper is also based on previous work on the distribution of flow rates in the panels.

This paper focuses on the impact of the number of PVT panels, of a specific design, as a single row, on the *night 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-IDX-SAHP system.

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2. Methodology

2.1. PVT panel prototype

Each of the PVT panels tested was 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 parallel in shared-consecutive manifold Z-configuration (see Fig. 3).

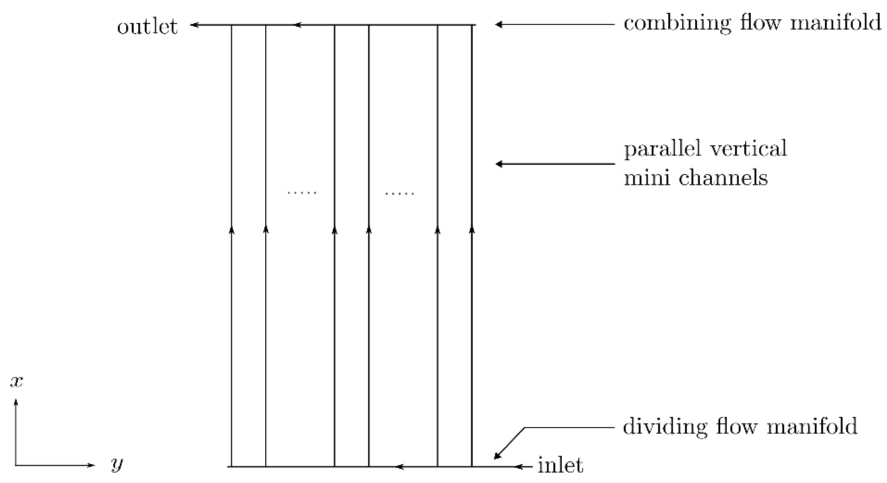

Fig. 1: Scheme of the tested PVT panel prototype

2.2. Experimental set-up

The experimental set-up was designed to study thermal performance of PVT panels during night. The experimental tests have been carried out with an inlet temperature higher than the ambient one using a buffer tank fitted with a heating resistance element. It was known that the PVT operation accordingly (PVT cooling down the fluid) is not exactly the opposite of the expected one in a PVT-IDW-SAHP system (PVT heating up the fluid), but it was the only way to study and evaluate the heat transfer with ambient air (the main topic of the present paper) since there was no available equipment to cool down the buffer tank fluid. The set-up is shown in photos (see Fig. 2) and described on the diagram below (see Fig. 3). It consisted mainly of a 100 L insulated cylindrical water tank (on the right of the Fig. 2 (a)), piping and a row of PVT collectors. This hydraulic circuit was filled with a mix of 60\% water and 40\% Mono-Propylene Glycol in volume, called MPG40%. The water tank temperature profile was estimated using two temperature probes (PT1000) installed in two immersion sleeves at its top and its bottom. As Fig. 2 (b) shows, the PVT panels were installed on a flat roof as a row in Z configuration with "shared-consecutive manifolds", i.e. their input and output manifolds were connected by fittings. For the PVT thermal power measurements, a flowmeter ($\dot{V}_{PVT}$) and two temperature probes (PT1000 Class A) were installed at the inlet ($T_{PVT,in}$) and the outlet ($T_{PVT,out}$). The heat transfer through the overall insulated pipes between the water tank and the PVT was estimated and considered in the calculation since that the inlet and outlet temperature probes are not installed directly next to PVT. The outdoor ambient temperature T_{amb} was measured a few meters away from the row of PVT panels (PT1000 Class A). The wind speed u was measured in the horizontal plane just above the PVT row. The fluid circulation was insured using a water circulator which was controlled by a controller via a PWM signal to ensure total volume flow rate within a given range in the row. The controller also controlled the activation of a 2-kW electrical heating resistance installed in the middle of the water tank with a hysteresis control strategy based on the temperature at the top of the hot buffer tank. Additionally, a solid-state relay was installed to be able to modify the operation power of the heating resistance via a PWM signal with a minimum of 800 W (40%) and a maximum of 2 kW (100%). Technical details of the equipment and instrumentation are respectively summarised in Table 1 and 2.

Tab. 1: Test bench equipment

Component	Manufacturer	Model	Details
System controller	Resol	MX	
Hot buffer tank	Thermador	BMEL100SK	100 L volume, 25 mm of insulation (rigid expanded polyurethane)
Electric heating resistance	Self	50.206.413	Maximum power 2 kW, length 290 mm
Circulation pump	Wilo	Para ST 15/7-50/IPWM2	130 mm long, HMT 69 kPa at 500 L/h

Tab. 2: Test bench instrumentation

Variable	Manufa-cturer	Model	Measurement range	Condi-tions	Tolerance / precision
$\dot{V}_{PVT}$	Resol	V40-25	[50 L/h, 5000 L/h] with nominal at 2500 L/h	[0°C, 120°C]	
$T_{PVT,in}$	Resol	FKP4 PT1000 Class A (2-wire)	[-50°C, 250°C]		(T) $\Delta T = \pm(0.15 + 0.002\lvert T\rvert)$
$T_{PVT,out}$	Resol	FKP4 PT1000 Class A (2-wire)	[-50°C, 250°C]		(T) $\Delta T = \pm(0.15 + 0.002\lvert T\rvert)$
T_{amb}	Resol	FAP13 PT1000 Class A	[-50°C, 250°C]		(T) $\Delta T = \pm(0.15 + 0.002\lvert T\rvert)$
u	SES Automation	RB-WT1000_4…20mA	[0 m/s, 35 m/s]		(P) +/- 3 %

Fig. 2: (a) Photo of the test bench technical room (b) Photo of the row of PVT panels on the flat roof

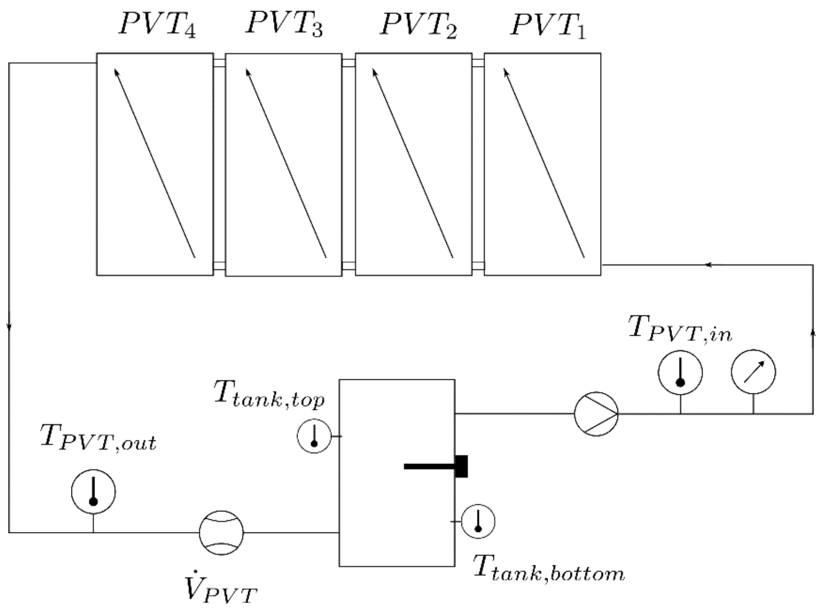

Fig. 3: Test bench diagram

2.3. Experimental protocol

A given number of panels, say N_{panels} in {2, 4, 6} in number, was installed in parallel with shared-consecutive manifolds, defining what is referred to as a "configuration". A test sequence for a given configuration consisted of multiple nights during which a total flow rate was imposed, ensuring that the average flow rate per panel remains approximately constant across configurations. The total flow rate in the row $\dot{V}_{PVT}$ was set between $N_{panels} \cdot 100\ L/h$ and $N_{panels} \cdot 120\ L/h$. In

A setpoint temperature at the top of the hot buffer tank, for hysteresis control of the electrical heating resistance, was selected based on the expected ambient temperature during the night. Given that the mean temperature of a PVT panel field operating as the sole heat source for a heat pump is between 6 and 12 K lower than the ambient temperature, the objective was to obtain data with a mean heat exchanger temperature 6 to 12 K higher than the ambient temperature. Thus, this setpoint temperature was set around 20 K for night ambient temperature around 10 K.

Tests conducted with the electric resistance at maximum power revealed significant transient behaviour. This was primarily caused by two factors: (1) the complete disruption of thermal stratification within the 100 L storage tank due to the incoming flow at the bottom, which was amplified by the tank's compact size and design, and (2) the tank's outdoor placement and limited insulation, which increase heat losses. It was therefore decided to adjust the power of the electric heating resistance according to the number of panels. For 2 panels, the power was set to the minimum possible, i.e. about 800 W. For 4 panels, a power of 2 kW meant that the resistance was always switched on and the inlet temperature was fairly stable. As it was not possible to get a more powerful heating resistance, the test with 6 panels was also carried out with this power of 2 kW.

As a result of these experimental set-up and protocol, the panels were in a situation (TB) (for "test bench") which is opposite to that of working as the heat source of a heat pump (HP). In (HP), mean PVT heat exchangers temperature is below ambient temperature and the measured heat output is positive. In (TB), mean PVT heat exchanger temperature is above ambient temperature and the measured heat output is negative. With ρ standing for the density of the heat transfer fluid and C_p for its specific heat capacity, the heat output of the PVT field can be written as:

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

At night, the PVT panels behave like an air/water heat exchanger and interact radiatively with the sky and the environment behind them. The latter may be considered at ambient temperature so that this contribution may be included in the convective heat gain/loss factor A_1. As it was detailed in (Kramer and Kramer, 2020), thermal power output of the PVT panels can be written as:

$$\dot{Q}/A_G = \begin{cases} -A_1^{(\mathbb{C},+)}(u) \cdot (T_{PVT,m} - T_{amb}) - A_{4*}^{(\mathbb{C},+)}(T_{PVT,m} - T_{amb}, u) \cdot (T_{PVT,m}^4 - T_{sky}^4) \; in \; (HP) \\ -A_1^{(\mathbb{C},-)}(u) \cdot (T_{PVT,m} - T_{amb}) - A_{4*}^{(\mathbb{C},-)}(T_{PVT,m} - T_{amb}, u) \cdot (T_{PVT,m}^4 - T_{sky}^4) \; in \; (TB) \end{cases} \quad \text{(eq. 2)}$$

Considering an average contribution of sky radiative heat transfer during test sequence, the thermal power output may be linearised as:

$$\dot{Q}/A_G = \begin{cases} -A_{1,sky}^{(\mathbb{C},+)}(u) \cdot (T_{PVT,m} - T_{amb}) \; in \; (HP) \\ -A_{1,sky}^{(\mathbb{C},-)}(u) \cdot (T_{PVT,m} - T_{amb}) \; in \; (TB) \end{cases} \quad \text{(eq. 3)}$$

Concerning the convective component, when there is little or no wind and therefore natural convection dominates, the air movement around the row of panels is likely not to be symmetrical from situation (HP) ($T_{PVT,m} < T_{amb}$) to situation (TB) ($T_{PVT,m} > T_{amb}$) as described for a hot flat plate and a cold flat plate in (Incropera et al., 2013). The corresponding thermal behaviours should correspond to different convective heat transfer coefficients. With wind, forced convection dominates, so it is possible to consider that the convective exchange coefficients are similar. Regarding radiative heat transfer from/to environment behind the panels, situations (TB) and (HP) are symmetrical because this radiative environment is considered at ambient temperature: radiative heat transfer from behind the panels in (TB) is the opposite of the one in (HP). Consequently, $A_1^{(\mathbb{C},+)}(u)$ and $A_1^{(\mathbb{C},-)}(u)$ can be considered similar for wind speed over 0.5 m/s (forced convection dominates free convection via Reynolds and Rayleigh calculation).

The main difference lies in the radiative interaction with the sky. The temperature of the sky is generally lower than the ambient temperature. In (HP), this interaction corresponds to a slightly positive, zero or negative heat transfer, whereas in (TB), it corresponds to a negative and often significant heat transfer. Consequently, $A_{1,sky}^{(\mathbb{C},+)}(u)$ and $A_{1,sky}^{(\mathbb{C},-)}(u)$ are not similar. Meanwhile, the average contribution of sky radiative heat transfer during the test sequence is equivalent for all configurations because PVT panels have the same view, both front and rear, and sky and ambient temperature difference distribution are equivalent. Therefore, the variation of $A_{1,sky}^{(\mathbb{C},-)}(u)$ from one configuration to another is only linked to the variation of the convection contribution. Thus, this variation is a relevant indicator of the variation of $A_{1,sky}^{(\mathbb{C},+)}(u)$ for different number of panels in the row linked to variations in air flow and temperature around the panels.

2.4. Analysis method

Data per minute was resampled at a time step of 5 minutes. Resampling data from a 1-minute to a 5-minute time step reduces measurement noise, highlights mid-term trends, and remains sufficient to capture the dynamics of PVT panel performance.

Data was filtered to keep only:

a) Nights ($G \leq 0 \; W/m^2$),

b) Total flow rate in expected range i.e. between $N_{panels} \cdot 100 \; L/h$ and $N_{panels} \cdot 120 \; L/h$,

c) $u \geq 0.5 \; m/s$

d) $T_{PVT,m} - T_{amb}$ in range [6, 12] K

For each configuration, the data was filtered to keep only the data with the same range of mean PVT heat exchanger temperature. This filtering was performed to ensure that the datasets were comparable and that the variation in the overall night heat loss factor was not due to different operating conditions.

Wind speed was clustered into intervals of 0.2 m/s and for each cluster, an ordinary least squares (OLS) regression was performed to determine the corresponding overall night heat loss factor.

3. Results and discussion

The tests were carried out on 2, 4 and 6 panels in November 2023. Each configuration was tested over 5 or 6 nights. During these tests, the heating resistance always remained on at the selected power. Figure 4 shows the thermal power points per unit gross area of the PVT row as a function of the difference between the mean temperature of the PVT heat exchangers and the ambient temperature for 2, 4, and 6 panels in the row. These data points were obtained after resampling at 5 minutes and initial filtering of the data with steps (a), (b) and (c). The first test (2 panels) corresponds to an average thermal power per unit of gross area of 244 W/m^2, i.e. a thermal power of about 950 W, which is consistent with the electric heating resistance being set at 40% of its maximum power, i.e. 800 W (the control by the solid-state relay and the PWM signal was probably not that precise). The second and third tests (4 and 6 panels) correspond respectively to an average thermal power per unit of gross area of 240 W/m^2 and 157 W/m^2, equivalent to a total thermal power of about 1.86 kW. This was also consistent with the heating resistance being set to 100% of its maximum power, i.e. 2 kW. However, during the third test, there was not the same range of $T_{PVT,m} - T_{amb}$, which means that it was not possible to compare the results of this test with the other two, as figure 5 shows. Therefore, only the tests with 2 and 4 panels in the row were selected for comparison, with an additional filtering $T_{PVT,m} - T_{amb} \in [8\,K, 12\,K]$.

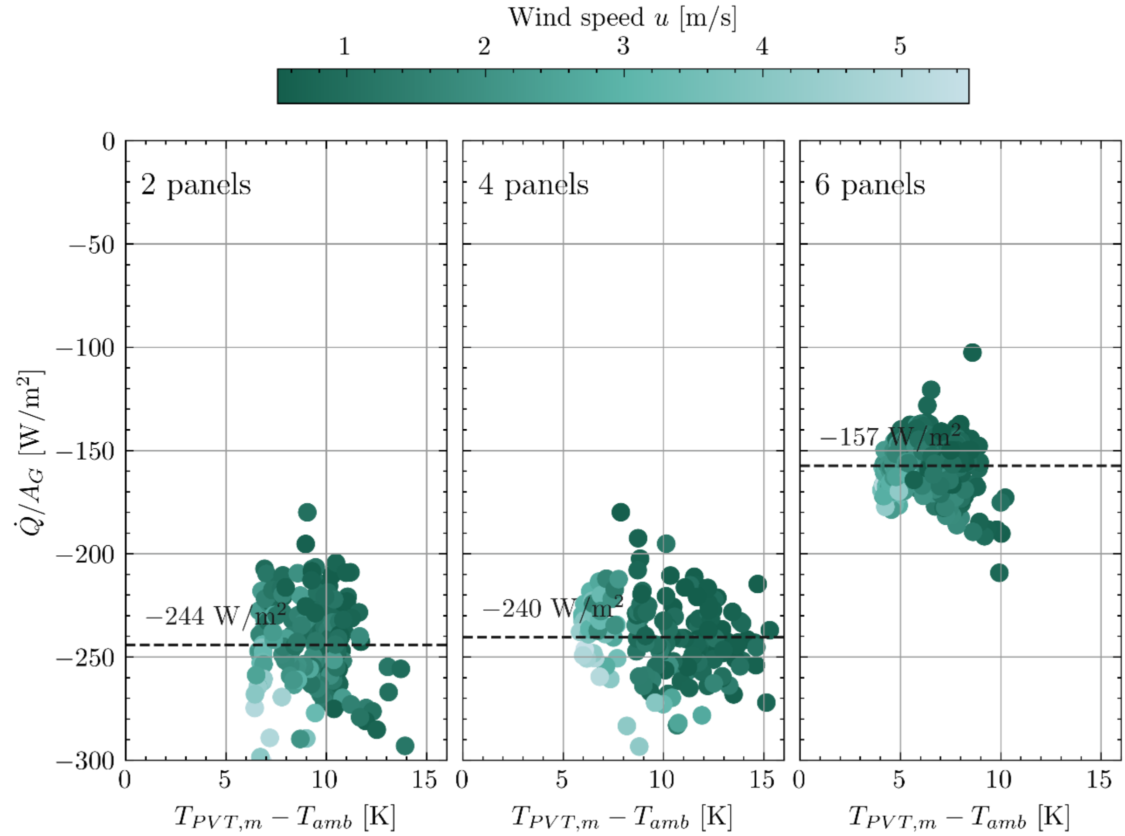

Fig. 4: Thermal power output per unit of gross area of the PVT row as a function of the difference between the mean temperature of the PVT heat exchangers and the ambient temperature for 2, 4, and 6 panels in the row

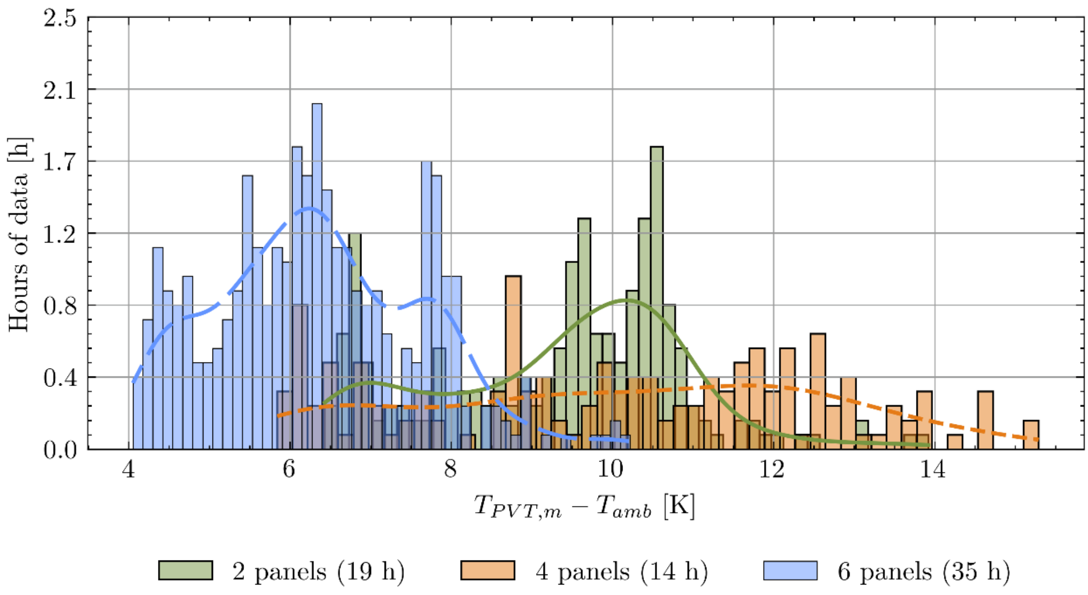

Fig. 5: Distribution of difference between the mean temperature of the PVT heat exchangers and the ambient temperature for the tests of 2, 4, and 6 panels in the row

As shown in table 3, after filtering, the quantities of data collected for 2 and 4 panels were 19 and 14 hours respectively. The segmented linear regression model is shown in Fig. 6. This model corresponds respectively to a Mean Absolute Error (MAE) of 20.5 W/m^2 or 9.8% of the mean value, and 24.5 W/m^2 or 10.2% of the mean value. Figure 7 shows the consistency between measured and modelled thermal output per collector gross area.

As table 3 shows, the values of $A_{1,sky}^{(\mathbb{C},-)}(1)$ obtained are 24.6 W/m^{2}K for 2 panels and 21.3 W/m^{2}K for 4 panels. On average, over this range $T_{PVT,m} - T_{amb} \in [8\,K, 12\,K]$ and wind speed $u \geq 0.5\,m/s$, the night heat loss factor was measured 10% lower for a row of 4 PVT panels than for a row of 2 PVT panels. In a previous study, it was established that the distribution of flow rates could cause a reduction in performance of around 5% for 4 panels. The second possible cause is the impact of the overall row 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).

Tab. 3: Summary of flow rates, hours of data, night heat loss factor, and Mean Absolute Errors for 2 and 4 panels in the row

Number of panels	**Mean flow rate per panel**	**Hours of data**	$A_{1,sky}^{(\mathbb{C},-)}(1)$	**MAE**	**MAE out of mean power**
[-]	[L/h/panel]	[h]	$[W/m^2K]$	[W/m2]	[%]
2	110	19	24.6	20.5	9.8%
4	105	14	21.3	24.5	10.2%

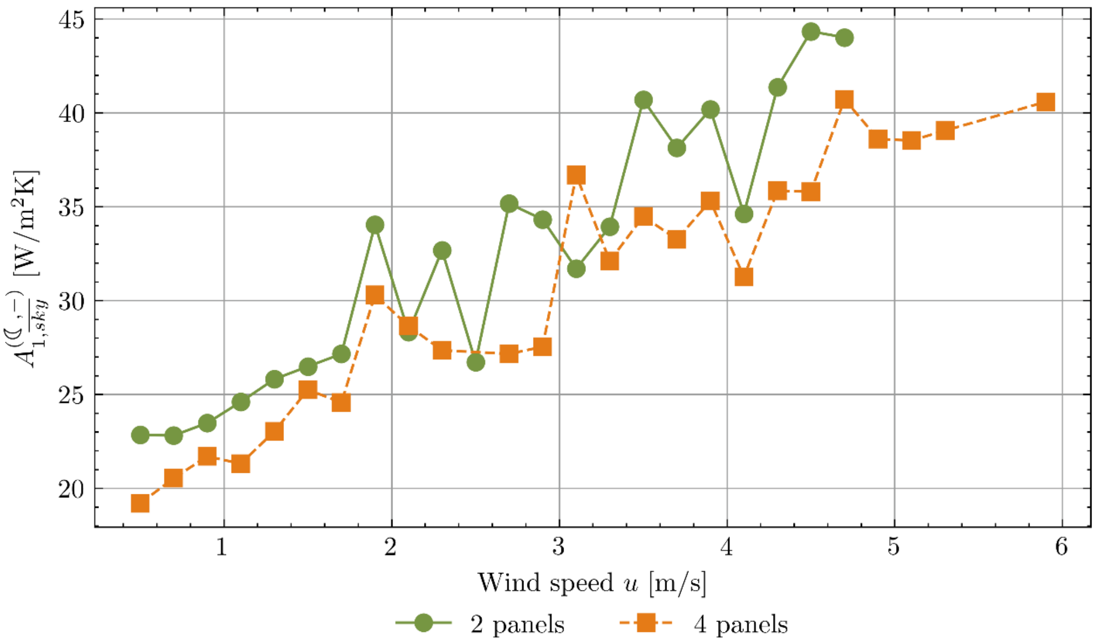

Fig. 6: Segmented linear regression model of night heat loss factor for 2 and 4 panels in the row

Fig. 7: 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"

4. Conclusion

Our research underscores the pivotal role of PVT panel configuration in enhancing the thermal performance of PVT-IDX-SAHP systems. The experimental analysis reveals that increasing the number of PVT panels in a single row may significantly reduce the night heat gain coefficient, thereby affecting overall system efficiency. A 10% reduction in night thermal performance was observed between 2 and 4 panels in the row. This may be due to uneven flow distribution in the panels, but also to the impact of the row geometry on air flow and temperature around the panels and therefore on heat transfer efficiency. This finding is crucial for the design and optimization of a PVT field to be the sole heat source of a heat pump, suggesting that careful consideration of panel arrangement can lead to substantial gains in energy performance. Future work should focus on conducting tests with perfectly stabilised inlet temperature below ambient and total mass flow rate. In this context, special attention would need to be given to condensation and frosting phenomena, as they may significantly impact thermal performance. These effects should be carefully studied to better understand their influence and to develop strategies for mitigating any adverse impacts on system efficiency.

5. References

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