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Numerical model for the thermal yield estimation of unglazed photovoltaic-thermal collectors using indoor solar simulator testing

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Abstract

It is a common practice to test solar thermal and photovoltaic-thermal (PVT) collectors outdoors. This requires testing over several weeks to account for different weather conditions encountered throughout the year, which is costly and time consuming. The outcome of these tests is an estimation of the thermal performance characteristics of the collector. Collector performance parameters can be derived with less effort by indoor testing under a solar simulator. However, in case of unglazed PVT collectors the thermal and the electrical performance is affected by two phenomena- additional long wave radiation (3000 nm and greater) due to emissions and reflections from the high temperature artificial sky, and an energy content of the PV spectrum (300-1100 nm) that differs from the global solar spectrum (300-2500 nm). These differences from the reference AM 1.5 solar spectrum lead to errors in the estimation of collector thermal and electrical performance. Therefore, results of indoor performance tests must be corrected to obtain the output of an unglazed PVT collector in real outdoor environment.

In this paper a method is proposed to estimate the real thermal performance of unglazed PVT collectors, by using a compact indoor solar simulator testing in combination with a detailed steady state numerical PVT collector model.

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The numerical model takes into account the physical and spectral attributes of the solar simulator and is used to correct for the unwanted phenomena to derive the actual outdoor collector performance. The resulting numerical model also offers detailed understanding of the collector and can therefore be used to optimize the design of the collector. Furthermore, this model is used to derive thermal performance characteristics of the unglazed PVT collector as defined by solar thermal testing standards, which can be used in system simulation tools (E.g. TRNSYS models) to obtain annual collector and system yields.

Keywords: Unglazed, Collector, Numerical, Simulator, Performance

Nomenclature

Symbol	Description	Unit
A	gross surface area of the collector	m^2
A_s	area of a collector segment	m^2
b_1	heat loss coefficient at zero reduced temperature	$\text{W}/\text{m}^2\text{-K}$
b_2	wind dependence of heat loss coefficient	$\text{W.s}/\text{m}^3\text{-K}$
b_u	collector efficiency coefficient (wind dependence)	s/m
C_p	specific heat of circulating fluid	$\text{J}/\text{kg-K}$
D	outer diameter of the tube	m
d_{cavity}	diameter of the U-shaped cavity	m
D_h	hydraulic diameter of the channel	m
F_G	function describing the dependency of the PV laminate output on the incident irradiance	-
F_{ij}	view factor from surface i to surface j	-
F_T	function describing the dependency of the PV laminate output on the temperature of solar cells	-
G	global irradiance normal to the collector surface	W/m^2
G^*	Net irradiance normal to the collector surface, corrected for the long wave radiation	W/m^2
Gr	Grashof number	-
I	output current of the PVT collector	A
J	radiosity of the surface	W/m^2
k	heat conductivity of a material	$\text{W}/\text{m-K}$
k_{air}	heat conductivity of air	$\text{W}/\text{m-K}$

l	length of the segment	m
$\dot{m}$	mass flow rate of the circulating fluid	kg/s
Nu	Nusselt number	-
Pr	Prandtl number	-
Q	energy flow	W
Ra	Rayleigh number	-
Re	Reynolds number	-
T	temperature	K
T_{red}	reduced temperature	K-m ² /W
U	heat loss coefficient	W/m ² -K
V	output voltage of the PVT collector	V
$\dot{V}$	volume flow rate of the circulating fluid	m ³ /s
W	centre to centre distance between the tubes in the collector	m
β	temperature coefficient for maximum electrical power at a given cell temperature	-
δ	thickness of a material layer in the PVT collector	m
ϵ	emissivity of the surface	-
η_0	zero loss thermal efficiency at $T_{red}=0$	-
$\eta_{0,b}$	peak collector efficiency	-
η_E	electrical efficiency of the PVT collector	-
η_{STC}	electrical efficiency of the PV laminate under standard test conditions	-
η_T	thermal efficiency of the PVT collector	-
ν	kinematic viscosity of a fluid	m ² /s
σ	Stefan Boltzmanns constant	W-m ⁻² K ⁻⁴
$\langle \tau \alpha \rangle$	transmission-absorption factor for the PVT collector	-

Subscripts

abs	flat absorber section
a	ambient air
back	back insulation
bot	bottom glass surface
cell	solar cells
con	conduction

conv	convection
curv	U-shaped curved absorber section
el	electricity
f	circulating fluid
gr	ground
g	glass surface
ins	back insulation
ir,net	net absorberd radiation
m	mean
mpp	maximum power point
rad	radiation exchange between the panel and the surroundings
room	refers to the experiment room
sky	outdoor/artificial sky
top	glass top surface
tube,in	inner surface of the tube carrying the circulating fluid

1. Introduction

Use of solar energy in the building sector is still limited due to several challenges, namely —limited roof area; high costs of solar thermal collectors when compared with conventional alternatives; and the lack of aesthetic integration into building roofs (Stryi-Hipp et al., 2012). PVT (abbreviated for combined photovoltaic-thermal) collectors can tackle these issues - by providing aesthetic homogeneity to the roof and using the roof area more effectively; and by reducing balance of system and installation costs, as a result of combining a PV panel and a solar thermal collector into a single module. A PVT panel/collector is a panel in which either PV cells are directly laminated on to a thermal absorber; or a PV module is placed on top of a thermal absorber, to produce heat and electricity from the same irradiated area. PVT collectors promise clear advantages over the combination of PV panels and solar thermal collectors.

Zondag (2008), Hasan and Sumathy (2010), Tyagi et al. (2012), Aste et al. (2014), Al-Waeli et al. (2017), etc., have reviewed the historical evolution of PVT collectors, by studying experimental work, field studies, qualitative performance evaluation, numerical models and analytical studies carried out by different researchers in past decades.

Over the years, a number of field testing and experimental studies on PVT collectors have been performed across the world (Suzuki and Kitamura, 1979), (Vries et al., 1997), (Tripanagnostopoulos et al., 2001), (Huang et al., 2001), (Bakker et al., 2002), (Sadamoto et al., 2003), (Bakker et al., 2004), (He et al., 2006), (Robles-Ocampo et al., 2007), (Assoa et al., 2007), (Chow et al., 2009), (Dupeyrat et al., 2011b), (Ceylan et al., 2014), (Rommel et al., 2014), (Cremers et al., 2015), (Aste et al., 2015), (Rommel et al., 2015), etc. In addition to the field testing, in recent years a number of studies involved indoor performance testing of PVT collectors under a solar simulator, testing both air and liquid PVT collectors (Solanki et al., 2009), (Dupeyrat et al., 2011a), (Agrawal et al., 2012), (Dupeyrat et al., 2014), (Fudholi et al., 2014), etc. In each of these studies, the solar simulator either consisted of halogen lamps, or measurement requirements were based on the ISO 9806-1 (ISO 9806, 2013) or EN 12975-1 (BS EN 12975-1, 2006) testing procedure, which only specify the maximum non-uniformity (below 15%) of irradiance over the test surface. Details of the spectral distribution of these solar simulators have not been mentioned in many of these studies. Currently, there are a few manufacturers of solar simulators complying with both PV and solar thermal testing standards.

On the modelling of PVT collectors, various numerical models have evolved in time. Bergene and Løvvik (1995) created a detailed physical model of a sheet and tube PVT collector, incorporating the heat transfer in the collector through conduction, convection and radiation, as well as temperature dependent power generation. This model was based on the model for solar thermal collectors by Duffie and Beckman (1991). Vries (1998) presented dynamic as well as steady state numerical models for sheet and tube PVT collectors. He showed that the steady state model while not accurately representing the transient behaviour of a PVT collector, can closely estimate the total yield over a day or over a year, and it is less computation intensive. Further improving this steady state model, Zondag et al. (2003) presented theoretical models for 9 different PVT designs and evaluated them with respect to each other. Santbergen et al. (2010) used the numerical model presented by Zondag to simulate a domestic hot water system for a single family house and estimated the annual yield of the system. Recent work on PVT modelling has been done by Rejeb et al. (2015), investigating the effect of meteorological, design and optical parameters on the performance of the PVT collector using a numerical dynamic simulation model. Hocine et al. (2015) also developed a numerical simulation model by adapting the Hottel-Whiller model (Hottel

and Willier, 1958) and making corrections to the heat loss coefficients. Aste et al. (2015) used a dynamic simulation model similar to the one proposed by Zondag, where they made some further corrections to the PV efficiency equation; and modelled the PV sandwich (PV laminate bonded with the roll-bond absorber) as a single temperature node.

Numerical models combined with physical testing are crucial to the development of PVT collectors. Measurement of electrical and thermal efficiencies of a PVT collector is required to characterize the collector performance, and to estimate monthly and annual yield. Outdoor testing is a common practice to test PVT collectors. However, it is weather dependent and may take a long period and many measurements over different typical days to derive the performance characteristics of a PVT collector. Furthermore, a suitable roof is required for such tests, where there are no surrounding trees or buildings to obstruct the sunlight and the wind. Therefore, performance testing of a thermal or a PVT collector costs a significant amount of time and money in terms of installation and measurements. These issues can be avoided with indoor solar simulator testing, which provides a much more controlled and steady environment to estimate the performance characteristics of a PVT collector. Solar simulators for testing of thermal panels are often quite large and the panel is placed at a distance from the simulator. A glass screen is placed between the simulator and the collector to block infrared radiation. Simultaneously, the simulator can be ventilated to keep the surfaces cool. However, for testing PV panels either flash test or compact solar simulators combining pre-cooling or surface cooling of PV panel are used. Such simulators are more suitable for PVT testing due to a better spectral match with solar radiation. However, due to its small size the test panel is kept very close to the simulator. This causes significant long wave radiations being reflected back to the collector by the artificial sky. Unlike glazed collectors, unglazed collectors are sensitive to long wave radiation.

Secondly, the solar simulator used in the testing may have a different proportion of energy in the PV spectrum compared to the solar radiation. This will affect the electrical efficiency and as a result the thermal efficiency of the PVT collector. These two unwanted effects of the simulator need to be corrected for estimating accurate thermal performance of PVT collectors. To our knowledge there is a lack of studies evaluating indoor testing with respect to the outdoor tests for PVT collectors. The method described in this paper takes into account these considerations to estimate the true performance of an unglazed PVT collector from indoor test results.

The aim of this paper is to discuss a steady state numerical model for a sheet and tube type unglazed liquid PVT collector and an unglazed liquid PVT collector with roll-bond¹ absorber, which considers the influence of the artificial sky on the thermal performance of the collectors.. This model has been validated with both –indoor and outdoor measurements on identical collectors. Using this numerical model in conjunction with the indoor testing, the performance of PVT collectors can be characterized accurately and in short time.

In the following sections of this paper, first the indoor and outdoor test facilities will be introduced. Then the numerical model will be explained. Finally, the accuracy of the numerical model with respect to the indoor and the outdoor tests will be discussed.

2. Test facilities

To carry out testing of thermal and electrical performance of PVT collectors, an indoor test setup consisting of a solar simulator was built in the lab, and an outdoor testing facility was built at the Eindhoven University of Technology (TUE).

2.1. Indoor test setup

The indoor test setup is shown in Fig.1. This test setup consists of two test loops —the thermal test loop and the electrical test circuit, shown schematically in Fig.2 and Fig.3 respectively. The setup includes a compact area (test surface 2m x 1.1m) AAA rated Eternal Sun solar simulator which simulates the spectrum of the sun in compliance with the international standard for solar simulators IEC 60904-9 (2007). The simulator produces a global irradiance of 1000 W/m² in the place of the collector at a distance of 10 cm from the artificial sky. The irradiance produced by this solar simulator has a spectrum distribution such that the incident radiation in the PV spectrum range (300-1100 nm; the range in which silicon cells operate) is only 63% of the radiation coming from the sun. This percentage was measured and provided by the manufacturer. This factor has been taken into account while estimating the electrical efficiencies during indoor testing (eq. 5 and eq.22). The spectral distribution of the solar simulator in the range 300-1100

¹The manufacturing process of a roll-bond absorber is given by Del Col et al. (2013).

nm, referred to as ST-1000, is shown in Fig.4. Since the panel must be placed very close to the simulator, the setup is forced ventilated to maintain a cold artificial sky, which creates a draft of 4.5 m/s average wind speed on the collector surface.

The thermal test loop is based on the NEN-EN ISO 9806 testing norm for solar thermal collectors. It consists of a Hüber UniChiller thermostatic bath through which the temperature of the liquid (water in this case) going into the collector can be controlled within 0.1 K. The flow is regulated via a needle valve and measured with an Omega FLR1011ST flow meter. The electrical test circuit contains a voltage bridge to measure the voltage of the PV laminate and a LEM current sensor for measuring the current. The PV circuit is connected to a micro-inverter which ensures the maximum power point operation of the solar module. During the measurements, fluid inlet and outlet temperatures, ambient temperature, wind speed, irradiance, DC current and DC voltage were measured for varying flow rates and varying fluid inlet temperature.

Based upon the error analysis of the test setup, temperatures can be measured within an uncertainty of ± 0.5 K (for a single thermocouple) and ± 0.25 K (for the differential temperatures measurement). The flow can be measured with an uncertainty of less than ± 0.65 L/hr. Overall, this leads to an uncertainty of ± 4.0 W in the estimation of thermal power and a relative error of less than ± 2 % in the measurement of optical efficiency. On the electrical side, the voltage is measured with an uncertainty of ± 0.15 V. The current is measured with an uncertainty of ± 0.04 A. The resulting electrical power is estimated within an uncertainty of ± 3.7 W or a relative error of ± 2.65 % in the measurement of electrical efficiency at STC.

Figure 1: Solar simulator indoor test setup

Figure 2: Schematic of solar simulator thermal test loop

Figure 3: Schematic of solar simulator electrical test loop

Figure 4: Eternal Sun solar simulator spectral distribution (van Domburg and Roest, 2015)

2.2. Outdoor test setup

The outdoor test setup, shown in Fig.5, was built by Solar Energy Application Centre (de Keizer et al., 2016) in collaboration with the TUE. This

setup is able to test three different PVT circuits simultaneously and independently. During the measurements, the global tilted radiation in the plane of the collectors was measured by Kipp & Zonen CMP11 pyranometer, while the long wave radiation was measured by Kipp & Zonen pyrgeometer CGR3. Wind speed and direction was also measured in the plane of the collectors using Gill WindSonic. The ambient, input and output temperatures to PVT collectors were measured by class 1/3B Pt100 sensors which have a 0.1 K tolerance of measurement. Flow was measured for each circuit separately by using the electromagnetic flow sensors Krohne Optiflux 1100C with a standard accuracy of 0.5 %. The resulting thermal power can be measured within an uncertainty of ± 13.5 W.

Figure 5: Outdoor PVT collector test setup

Simultaneously, electrical measurements were done under maximum power point (MPP) operation. MPP operation was achieved by using SolarEdge DC/DC power optimizers connected to SolarEdge inverter. The DC voltage was measured directly, while the DC current was measured via Murata shunt (0.25% accuracy) for each PVT collector. The resulting electrical power can be measured within an uncertainty of ± 6.1 W.

All data was logged using Yokogawa MW100 data logger. The heat produced by PVT collectors was delivered to the heating and cooling system of TUE,

which is connected to underground aquifers. A combination of heaters and chillers ensured that water was delivered back to the collectors at specified desired temperatures.

2.3. Description of prototype PVT collectors

Two unglazed PVT liquid collectors with different absorber designs have been investigated. In both collectors, a glass-glass PV laminate is combined with an aluminium absorber which is either a sheet and tube type –manufactured by SolarTech (SolarTech, 2014), or a roll-bond absorber with serpentine flow channels –manufactured by DimarkSolar (DimarkSolar, 2014). Henceforth, these PVT collectors will be referred to as the PVT-1 and the PVT-2. The PVT-1 uses CIGS solar cells, while the PVT-2 uses polycrystalline silicon solar cells. The PV laminate and the absorber are pressed mechanically against each other to maintain good thermal contact. The basic design of these PVT collectors is shown in Fig.6.

Figure 6: Cross section of the tested PVT collectors along with energy flows (left) PVT-1, (right) PVT-2

Fig.6 also shows the energy flows within and around each PVT collector. The solar irradiation falls on the solar cells. A part of it is converted into electricity and the rest is absorbed as heat. Part of this absorbed energy is lost to the environment through convection and radiation. The remaining energy is harvested as useful heat. The useful heat is transported to the absorber and then conducted to the tubes (PVT-1) or the flow channels (PVT-2). Eventually, this heat is carried away by the circulating fluid via convection.

It is noteworthy that most of the heat loss occurs on the front side of the collector, as the back side is insulated (Expanded polystyrene, 26 mm thick insulation for PVT-1; and PET foam, 13 mm thick insulation for PVT-2).

3. Measurements

Each PVT collector has been tested indoors under the solar simulator and outdoors on the roof.

3.1. Indoor performance measurements

Each collector was tested for two different flow rates (60 and 75 kg/hr); and 5 different fluid inlet temperatures (10, 20, 30, 40 and 50 °C). These flow rates and temperatures were chosen keeping in mind low temperature applications for unglazed PVT collectors.

The solar simulator produces a uniform global irradiance of 1000 W/m². Long wave radiation was calculated in accordance with ISO 9806 testing norm. Throughout the measurements, all measured parameters were logged using the SignalExpressTM program from National Instruments. For each measurement, the thermal efficiency, the electrical efficiency, the mean fluid temperature and the reduced temperature were calculated. Only the steady state values of the measurements were used in the calculations. Steady state refers to the state when there is no discernible change in fluid inlet and outlet temperatures, flow rate, voltage and current with respect to time. Steady ambient conditions are automatically guaranteed in the indoor test setup. The thermal performance of the PVT collector is expressed by the thermal efficiency which is defined as:

$$\eta_T = \frac{m \cdot C_p (T_{out} - T_{in})}{(G^* \cdot A)} \quad (1)$$

Here G^* is the net irradiance in the collector plane, adjusted for the long wave radiation, defined by

$$G^* = G + \frac{\epsilon}{\alpha} \cdot (E_L - \sigma \cdot T_a^4) \quad (2)$$

Where, E_L is the long wave radiation in the collector plane. In case where E_L is not measured, the ISO 9806 norm suggests a method to calculate the long wave radiation from the sky temperature.

$$E_L = F_{sky} \cdot \epsilon_{sky} \cdot \sigma \cdot T_{sky}^4 \quad (3)$$

Here, F_{sky} is the view factor from the collector to the sky, which is equal to $(1 + \cos\beta)/2$; β being the collector tilt. For indoor testing with artificial sky, $\beta = 0$; and the view factor must be calculated for the test setup.

Reduced temperature is expressed as:

$$T_{red} = \frac{T_m - T_a}{G^*} \quad (4)$$

The electrical performance of the PVT collector is expressed by the electrical efficiency defined as:

$$\eta_E = \frac{V_{mpp} \cdot I_{mpp}}{G \cdot A} \quad (5)$$

PVT collectors investigated in this study are unglazed PVT liquid collectors. As per the ISO 9806 norm, the steady state thermal efficiency of the unglazed solar thermal collector can be represented by the following equation:

$$\eta_T = (1 - b_u \cdot u) \cdot \eta_{0,b} - (b_1 + b_2 \cdot u) \cdot T_{red} \quad (6)$$

Henceforth, the coefficients in eq.6 will be referred to as the collector thermal performance parameters throughout this paper.

3.2. Outdoor performance measurements

Outdoor measurements were carried out at TUE at the outdoor testing facility of SEAC from May 2015 to May 2016. Steady state outdoor performance of the PVT collectors was measured in accordance with the NEN ISO 9806 test norm. Only the data points for which the following conditions were satisfied (over the measurement period of 15 minutes) were used in the analysis:

- Global solar irradiance is above 650 W/m²
- Average air speed over the collector is between 0-3 m/s
- Global solar irradiance and long wave irradiance do not change beyond ± 50 W/m² of the mean value over the measurement period
- Ambient temperature does not change beyond ± 1.5 K of the mean value over the measurement period
- Flow rate does not change beyond $\pm 5\%$ of the mean value over the measurement period
- Inlet temperature does not change beyond ± 0.2 K of the mean value over the measurement period

- Outlet temperature does not change beyond ± 0.5 K of the mean value over the measurement period
- Surrounding air speed does not change beyond ± 0.5 m/s of the mean value over the measurement period but it may change by ± 1.0 m/s up to 10 % of the measurement period

These data points were used to derive thermal performance parameters for the tested PVT collectors, in accordance with eq.6.

4. Numerical model

A steady state numerical model was developed for the investigated unglazed PVT collectors described earlier. This model is based on the second thermal model for flat plate covered/glazed PVT collectors, proposed by Vries (1998), which Zondag et al. (2003) used to evaluate the thermal and electrical efficiencies of different PVT design concepts. Santbergen et al. (2010) incorporated more accurate solar cell absorption factors in this model for better estimation of electrical yield for different PV technologies.

The steady state thermal model is based on the assumption that each layer of the collector is in thermal equilibrium at each moment and dynamic effects associated with the heat capacity of the collector have been neglected. De Vries showed that although dynamic effects exist, they cancel out over a day. For example, during the early hours of the day calculated fluid outlet temperatures are higher than measured because the solar heat is being used to heat up the collector, while in the evening hours calculated fluid outlet temperatures are lower than measured due to the discharging of heat stored in PVT collector over the day. De Vries compared the results obtained from the dynamic model and the steady state model, and concluded that the average daily yield differed only by less than 0.7 %. For the estimation of the annual yield, this difference is even more insignificant.

In this paper, the aforementioned improved numerical model from Santbergen has been adapted to unglazed PVT collectors. A radiation heat exchange model has been developed and added to the collector model to account for the additional thermal heat gain due to long wave emissions and reflections from the artificial sky. Furthermore, a heat conduction model for two-dimensional heat conduction in the absorber has been included to the collector model. Inputs to this model are ambient conditions such as radiation, ambient temperature, wind speed, etc. Moreover, other inputs to this model are the

dimensions of the artificial sky, the distance between the simulator and the panel, the spectral distribution of the simulator, the temperature of the hot artificial sky and the ventilation airflow between the simulator and the collector.

A simplification of the model by straightening the serpentine tube is shown in Fig.7. The numerical model analyses the collector by dividing it into number of segments along the flow direction. Each layer of the segment is treated as a separate temperature node. The heat balance for each layer is solved simultaneously to obtain the temperatures at all the nodes in each segment along the length of the tube/channel. The liquid output temperature of a segment is input to the next segment. All the energy flows have been shown in Fig.6 for both PVT collectors. The heat balance for different layers in the sheet and tube type PVT collector is represented by the following set of equations (7-12).

Figure 7: A simplification of the model by straightening the tube

$$Q_{con,top} - Q_{conv,air} - Q_{rad} = 0 \quad (7)$$

$$Q_{ir,net} - Q_{el} - Q_{con,top} - Q_{con,bot} = 0 \quad (8)$$

$$Q_{con,bot} - Q_{con,abs} - Q_{back1} - Q_{back2} = 0 \quad (9)$$

$$Q_{con,abs} - Q_{conv,f} = 0 \quad (10)$$

$$Q_{conv,f} - Q_T = 0 \quad (11)$$

$$Q_T - \dot{m} \cdot C_p \cdot (T_{f,out} - T_{f,in}) = 0 \quad (12)$$

In the next section only the radiation heat exchange and the heat conduction in the absorber will be described in detail. Other heat flows have been described in literature as mentioned in section 4.3.

4.1. Radiation heat exchange

The PVT collector loses heat to the environment via radiation. In outdoor installations, the collector exchanges radiative heat with the sky as well as the surroundings. However, in this study during the indoor tests the collector is directly facing the hot artificial sky at a small distance of 10 cm from it with the view factors 98% or above. During testing, the artificial sky was measured to be at a temperature close to 60 °C. Due to high view factor, the collector receives higher thermal radiation due to emission and the reflection from the artificial sky. Performance of unglazed collectors is sensitive to long wave radiation and therefore a precise estimation of long wave radiation is required. When long wave radiation is not measured, the ISO 9806 norm suggests a formula (eq.3) to calculate the long wave radiation. However, it only considers the long wave radiation emitted by the artificial sky. This is true for large simulators where the collector is placed away from the solar simulator and view factors are relatively small. In this case only the long wave radiation emitted by the artificial sky arrives at the collector. However, in case of the compact solar simulator used in this work, reflected long wave radiation from the artificial sky needs to be added in eq.3 to calculate the total long wave radiation in the collector plane.

To model this radiation heat exchange between the two surfaces, the radiation network approach for opaque diffused grey surfaces, described by Incropera et al. (2013) has been used. This model is applicable to any solar simulator, large or compact, used for solar thermal testing.

The radiation network model considers the radiation heat exchange between the collector, the solar simulator and the room in which the solar simulator

is located. The room has much bigger surfaces compared to the collector and the simulator. Hence, the room surfaces are considered to behave like a black body. The net radiation heat loss from the collector to the ambient can be written using the radiation network approach.

$$Q_{rad} = \frac{\sigma (T_{top,g}^4 - J_{top,g})}{(1 - \epsilon_{top,g}) / \epsilon_{top,g} \cdot A_s} \quad (13)$$

Here, $J_{top,g}$ is the radiosity of the collector top surface defined as the sum of the irradiance emitted and the irradiance reflected by the collector top surface. $J_{top,g}$ is a function of all the surface temperatures, the emissivity of the artificial sky (measured to be 0.55 ± 0.01 with FLIR T650sc thermal imaging camera), the emissivity (0.85 for the laminate surface) and absorptivity $\langle \tau \alpha \rangle$ of the PVT collector and the view factors. J can be obtained by solving the equations described in the radiation network approach model.

In case of outdoor installations, the collector loses heat via radiation to the cold sky and to the surroundings which are assumed to be the ground in this case. Radiation heat loss by the collector to the ambient can be written in a simplified manner as following (Zondag et al., 2002):

$$Q_{rad} = A_s \cdot \sigma \cdot \epsilon_{top,g} \cdot (F_{sky} (T_{top,g}^4 - T_{sky}^4) + F_{gr} (T_{top,g}^4 - T_{gr}^4)) \quad (14)$$

For indoor test, the long wave radiation in the collector plane, which is required for plotting the thermal efficiency curve (eq.6) is calculated as:

$$E_L = F_{sky} \cdot J_{sky} + (1 - F_{sky}) \cdot \sigma \cdot T_a^4 \quad (15)$$

Where, F is the view factor from the collector to the sky. View factor from the collector to the artificial sky can be calculated from the size of the two surfaces and the distance between them. J is calculated from the radiation exchange model.

For outdoor measurements eq.3 can be used. Estimation of E_L is required to plot precise thermal efficiency curves as per eq.6. Henceforth, E_L calculated by eq.15 will be referred to as the sky reflection correction throughout this paper.

4.2. Heat conduction in the absorber

The heat flows from the absorber to the tube in the direction perpendicular to the tube as shown in Fig.8 for PVT-1. Here, arrows show the direction of heat conduction. The absorber has a U-shaped cavity in which the tube is placed. The tube acts as a heat sink for the absorber. Therefore, a temperature gradient exists in the absorber in between the tubes as shown

in this figure. In the absorber, heat travels in the direction perpendicular to the tubes and is then transported to the tube via physical contact between the U-section of the absorber and the tube. By visual inspection, it was seen that there was no direct contact between the tube and the PV laminate on top. Only a very small amount of heat flows downwards through the insulated back side of the collector. But a very significant heat loss is upwards to the ambient.

For PVT-2, a similar phenomenon is described by Aste et al. (2016) for a PVT collector with roll bond absorber.

Figure 8: Heat conduction in absorber (left), Heat balance in absorber(right)

To estimate the heat conduction from the absorber to the tubes, the heat conduction model for the temperature distribution between the tubes and the absorber, described by Duffie and Beckman (2013), has been adopted to this case. The proposed model differs from the Duffie and Beckman model as follows: 1) there is no absorber on top of the tube as seen in Fig.8. Instead the tube is enclosed within the U-shaped cavity that extends from the aluminium absorber; 2) in this case, the absorber exchanges heat with the PV laminate on top, with the ambient through the insulation below, and with the tube carrying the circulating fluid.

The heat balance of the U-shaped cavity in the aluminium absorber shown in Fig.8, is given by eq.16 -18.

$$Q_{tube} = Q_{con,abs} - Q_{back2} \quad (16)$$

$$Q_{con,abs} = 2 \cdot k_{abs} \cdot \delta_{abs} \cdot l \cdot \left(\frac{dT}{dx} \right)_{x=(W-D)/2} \quad (17)$$

$$Q_{back2} = \pi \cdot d_{cavity} \cdot l \cdot U_{abs \rightarrow a} \cdot (T_{curv} - T_a) / 2 \quad (18)$$

Here T_{curv} is the temperature of the U-shaped absorber section. As depicted in Fig.8, focusing on a small section of the absorber the following heat balance equation can be written as:

$$-U_{abs \rightarrow cells} \cdot \Delta x \cdot (T_{abs} - T_{cells}) - U_{abs \rightarrow a} \cdot \Delta x \cdot (T_{abs} - T_a) + \left(-k_{abs} \cdot \delta_{abs} \cdot \frac{dT_{abs}}{dx} \right)_x - \left(-k_{abs} \cdot \delta_{abs} \cdot \frac{dT_{abs}}{dx} \right)_{x+\Delta x} = 0 \quad (19)$$

Here, $\left(\frac{dT_{abs}}{dx} \right)_{x=(W-D)/2}$ can be obtained by solving eq.19, following the approach described by Duffie and Beckman (2013).

Combining eq.16 - 18 the heat transfer from the absorber to the tube can be expressed as:

$$Q_{tube} = -(W - D) \cdot F \cdot U_l \cdot l \cdot (T_{curv} - T_{eff}) - \pi \cdot d_{cavity} \cdot l \cdot U_b \cdot (T_{curv} - T_a) / 2 \quad (20)$$

Where F is given by $\tanh(m \cdot (W - D)/2) / (m \cdot (W - D)/2)$;

$m = \sqrt{(U_l / (k_{abs} \cdot \delta_{abs}))}$; $U_l = U_{abs \rightarrow cells} + U_{abs \rightarrow a}$;

and $T_{eff} = (U_{abs \rightarrow cells} \cdot T_{cell} + U_{abs \rightarrow a} \cdot T_a) / U_l$.

Here, T_{eff} is the effective surrounding temperature experienced by the absorber.

For PVT-2, the distance between the channels is very small and of the same order as the width of the channel. Furthermore, since the channels are blown within the aluminium absorber the heat resistance is minimal. Therefore, for PVT-2 we can safely assume that no temperature gradients exist in the absorber perpendicular to the channels.

4.3. Other heat flows

The amount of irradiation absorbed by a collector is governed by the transmission absorption factor ($\langle \tau \alpha \rangle$).

$$Q_{ir,net} = \langle \tau \alpha \rangle \cdot A_s \cdot G \quad (21)$$

$\langle \tau \alpha \rangle$ was measured to be 0.937 for the PVT-1 and 0.90 for the PVT-2. These measurements were performed at ECN (Energy Research Centre of the Netherlands) (Zondag, 2015). Electricity production can be calculated by:

$$Q_{el} = F_\lambda \cdot F_G \cdot (1 + \beta \cdot (T_{cell} - 25)) \cdot A_s \cdot G \cdot \eta_{STC} \quad (22)$$

Here, F_λ is the factor describing the global irradiation spectral match between the simulator and the solar radiation. It is 1 for outdoor installations and 0.63 for the solar simulator used for the testing.

F_G is the function describing the dependency of solar cell output on the incident radiation, and β describes the temperature dependence of PV module

efficiency.

Convective heat loss from the collector to the environment is given by:

$$Q_{conv,air} = A_s \cdot (Nu_{air} \cdot k_{air}/L_c) \cdot (T_{top,g} - T_a) \quad (23)$$

The Nusselt number depends on the ambient air flow conditions —forced vs free convection; and laminar vs turbulent convection. Equations which govern the Nu_{air} are given by Vries (1998) and Incropera et al. (2013).

To calculate the heat conduction in the collector, thermal contact resistances need to be calculated, which arise due to the lack of perfect contact between two surfaces. These resistances in the PVT-1 are the contact resistance between the PV glass and the absorber, and the resistance between the tube and the U-shaped absorber cavity. Henceforth, these resistances will be referred to as $R_{contact}$ and R_{cavity} respectively. These are depicted in Fig. 7. Similarly, the unknown resistance in the PVT-2 is the contact resistance between the PV laminate and the absorber.

Conduction heat flows in the collector are represented by Fourier's law of heat conduction.

$$Q_{con,top} = - \frac{k_{top,g} \cdot A_s \cdot (T_{top,g} - T_{cell})}{\delta_{top,g}} \quad (24)$$

$$Q_{con,bot} = - \frac{T_{abs} - T_{cell}}{\delta_{top,g}/(k_{bot,g} \cdot A_s) + R_{contact}/A_s} \quad (25)$$

$$Q_{back1} = - \frac{k_{ins} \cdot A_s \cdot (T_a - T_{abs})}{\delta_{ins}} \quad (26)$$

Here, $R_{contact}$ is the contact heat resistance between the PV glass and the absorber.

The heat transfer from the absorber to the tube is described by:

$$Q_{tube} = \frac{A_s \cdot (T_{curv} - T_{tube,in})}{R_{cavity} + R_{tube}} \quad (27)$$

Here, R_{cavity} is the heat resistance between the U-shaped absorber cavity and the tube inside the cavity (only for PVT-1; for PVT-2, $R_{cavity} = 0$), normalised to the area of the PVT segment (A_s). R_{tube} is the thermal resistance across the tube/channel, normalised to A_s .

The convective heat transfer to the circulating fluid is governed by following equation:

$$Q_{conv,f} = Nu_D \cdot (k_f/D_h) \cdot P_D \cdot l \cdot (T_{tube,in} - T_f) \quad (28)$$

Here, P_D is the perimeter of the tube/channel.

Correlations to obtain Nu_D are given by Incropera et al. (2013). Moving on to the convective heat transfer in PVT-2, fluid carrying channels are not

circular but trapezoidal in shape as shown in Fig.6. The Nusselt number for non-circular channels with laminar flow is calculated to be 1.55, from the empirical relations proposed by Shah (1975). For the turbulent flow through trapezoidal channel, the correlations for the circular tubes are used by replacing the tube diameter with the hydraulic diameter of the channel.

4.4. Estimation of unknown heat resistances

At steady state the heat transferred from PV cells to the water is equal to the heat carried away by water. Since both the panels were well insulated, the heat transfer through the back insulation is considered to be insignificant. Using this simple heat balance (eq.29), and measuring the temperatures at different nodes in the PVT collector, total heat resistance between the cells and water can be estimated. This overall heat resistance is a combination of known heat conduction and heat convection resistances, and an unknown contact resistance. Since all other heat resistances in the panel are known this method can be used to estimate unknown contact resistances.

$$(T_{cell} - T_m)/R_{cells \rightarrow water} = m \cdot C_p (T_{out} - T_{in}) / A \quad (29)$$

Tests were carried out at a flow rate of 30 lph and inlet water temperatures of 10-20 °C. Low flow rates and low inlet temperatures result in higher temperature gain by water, meaning lower relative uncertainty in the differential temperature measurement. Water was used as the circulating medium. The average temperature of PV cells was estimated using eq.30, by measuring electrical output during the measurements and at room temperature. From these set of measurements an average value of contact resistance was calculated to be used in the numerical model.

$$\frac{Q_{el,Tcell}}{Q_{el,room}} = \frac{1 + \beta \cdot (T_{cell} - 25)}{1 + \beta \cdot (T_{room} - 25)} \quad (30)$$

For PVT-1 there are two contact resistances which need to be estimated. This was accomplished as follows. First, $R_{contact}$ was estimated. The U-shaped cavity between the absorber and the tube was filled with heat conducting silicon paste whose thermal resistance was known. This ensured a good thermal contact between the absorber cavity and the tube. Now, the only unknown resistance in the collector was $R_{contact}$. Following the above mentioned method $R_{contact}$ was estimated. Earlier, the same process was repeated on the standard PVT-1 (without the silicon paste). The value of

$R_{contact}$ was applied to those measurements to calculate R_{cavity} . Following this procedure, values of $R_{contact}$ and R_{cavity} for PVT-1 were calculated to be $0.017 \pm 0.004 \text{ m}^2\text{K/W}$ and $0.049 \pm 0.004 \text{ m}^2\text{K/W}$ respectively. Similarly, $R_{contact}$ for PVT-2 was calculated to be $0.018 \pm 0.004 \text{ m}^2\text{K/W}$. The numerical model was completed by inserting the estimated values of these contact resistances.

4.5. Thermal efficiency curves

Generally, outdoor performance testing is carried out to derive the thermal performance curve of solar collectors. In this paper an attempt is made to derive the same performance curve without physical measurements and rather by using the numerical model. Steady state thermal efficiency curve described by eq.6 was derived for investigated PVT collectors by using the numerical model. Here, the numerical model is considered to be simulating the actual collector behaviour while being subjected to weather conditions simulated by historical weather data. Using this hypothesis, a large number of theoretical performance data was produced by calculating thermal efficiencies and corresponding reduced temperatures for outdoor conditions. The irradiance, the ambient temperature and the wind speed were taken from the Meteonorm (TMY2) weather data for the De Bilt weather station located in The Netherlands. However, any location can be used for this method since the collector performance characteristics are independent of the location and the weather. Collector performance parameters were calculated by fitting eq.6 to this data. The 'fit' function from MATLAB was used for curve fitting. Based on measurements and reported values, $\frac{\epsilon}{\alpha}$ was calculated to be 0.91 and 0.94 for PVT-1 and PVT-2 respectively. Aligning with the ISO 9806 testing procedure for steady state testing of unglazed collectors, only data points with irradiance greater than 650 W/m^2 and wind velocities up to 3.5 m/s were used to obtain the collector performance parameters for the two PVT collectors. Input fluid temperatures conditions selected in the numerical model were 10 to $50 \text{ }^\circ\text{C}$.

5. Results and discussions

The analysis of the numerical model starts with investigating the first argument made in this paper —the thermal performance of unglazed collectors

is altered by the compact solar simulator due to unwanted long wave radiation. The long wave radiation in the plane of the collector is altered when the collector is placed underneath the simulator during testing, which results in inaccurate thermal performance measurement. To show this, a comparison between the thermal performance in the indoor and the outdoor tests was made for the two collectors. Moreover, the numerical model was validated against the indoor performance tests under the solar simulator as well as against outdoor tests. Since indoor tests were carried out at an average wind speed 4.5 m/s, outdoor measurements corresponding to similar wind speed were taken for comparison against indoor measurements.

Measurement results are shown in Fig.9 and Fig.10. For both PVT-1 and PVT-2, the numerical model estimates the thermal efficiency within 0.9% (absolute) of the measured value. In both figures, the indoor measurement data and the indoor performance curve based on the numerical model are drawn in accordance with equations (1-4) defined in ISO 9806 norm. When compared against outdoor tests, indoor tests overestimate the thermal performance by 5.6% and 11.1% for the PVT-1 and the PVT-2 respectively. It can also be seen in these figures that the indoor performance can be corrected by the numerical model to arrive at the outdoor performance.

Figure 9: Difference between indoor (using a compact solar simulator) and outdoor thermal performance of PVT-1 at wind speed 4.5 m/s

Figure 10: Difference between indoor (using a compact solar simulator) and outdoor thermal performance of PVT-2 at wind speed 4.5 m/s

The first correction done by the numerical model is the precise calculation of long wave radiation coming from the artificial sky, based on the proposed radiation exchange model (eq. 15). As can be seen in Fig.9 and Fig.10, this sky reflection correction is 4.5% and 8.1% for PVT-1 and PVT-2 respectively. In absence of the collector underneath the solar simulator, the calculated E_L is $382 W/m^2$, while during the testing of the PVT collector the calculated E_L is $581 W/m^2$ (52.1% higher) and $537 W/m^2$ (40.5% higher) for PVT-1 and PVT-2 respectively. This shows that the introduction of the panel itself alters the long wave radiation arriving at the panel surface. PVT-1 being less thermally efficient than PVT-2, has higher top surface temperatures and emits more thermal radiation than PVT-2, which are then partially reflected back to the panel. From this analysis it can be concluded that during the testing with compact solar simulators, the long wave radiation received in the plane of the collector not only depends on the properties of the solar simulator but also on the performance characteristics of the collector being

tested. Therefore, the pre-measurement of E_L over the entire test surface is not sufficient. Since the reflected radiation is not uniformly distributed over the artificial sky surface, measuring the long wave radiation by placing a pyrgeometer beside the collector during the testing may also not be enough, as it will receive less long wave radiation than the collector.

In addition to the sky reflection correction, the numerical model makes corrections for the spectral mismatch to estimate the actual outdoor performance (eq. 22). As can be seen in Fig.9 and Fig.10, the PV spectrum correction is 1.1% and 3.0% for PVT-1 and PVT-2 respectively. Solar simulator used in this study has lower energy content in the PV spectrum which results in lower electrical output. This means more irradiation is available for heat generation, resulting in higher thermal efficiency. If the spectrum correction is not made then the measured electrical efficiencies are between 8.45-8.70% for PVT-1 (against 14.7% STC efficiency), and between 7.6-8.1% for the PVT-2 (against 14.2% STC efficiency). A small contributor to this drop is higher solar cell temperatures. However, the spectral mismatch between the artificial sky and the solar radiation is the major contributor (83-89%) to this efficiency drop. Therefore, a spectral correction is required for accurate estimation of electrical and thermal efficiency of a PVT collector.

Figures 11, 12 and 13 have been drawn to further demonstrate the impact of the interaction between the artificial sky and the collector, showing the thermal performance of PVT-1 for different sky emissivity, temperatures and view factors, based on uncorrected long wave radiation (eq.3). Similar results were obtained for PVT-2. As can be seen in these figures, the collector performance is not significantly affected by the artificial sky temperature, but it is very sensitive to the sky emissivity and the collector view factor which depends on the size of the collector being tested.

Figure 11: Thermal performance of PVT-1 at wind speed 4.5 m/s and the sky view factor ≈ 1 - for different artificial sky emissivities

Figure 12: Thermal performance of PVT-1 at wind speed 4.5 *m/s* and sky emissivity 0.55
- for different artificial sky temperatures

Figure 13: Thermal performance of PVT-1 at wind speed 4.5 m/s and sky temperature 60 °C - for different artificial sky view factors

The numerical model takes into account this interaction between the artificial sky and the collector to estimate the long wave radiations on the collector plane. Since the true collector performance is independent of the type of simulator used, the thermal performance estimated by the numerical model must not be sensitive to the physical properties and the view factors of the artificial sky. Fig.14 shows the thermal performance of PVT-1 for different possible artificial sky emissivity (0.35-0.85), sky temperatures (10-70 °C) and view factors (0.1-1.0). PVT-2 shows similar results. It can be seen that the numerical model is able to make the necessary corrections to estimate the thermal efficiency within 1% of the actual outdoor performance. This demonstrates that irrespective of the type of solar simulator used for indoor testing, the model always corrects the collector performance to its true outdoor performance. The smaller the view factor from the collector to the simulator, the smaller is the long wave radiation correction.

Figure 14: Modelled corrected thermal efficiency of PVT-1 - for different solar simulators

Next, the numerical model was validated against the steady state outdoor measurement points. Thermodynamic parameters regarding radiation and convection, and operational parameters regarding the ambient were set to outdoor conditions in the numerical model.

Fig.15 and Fig.16 show the measured thermal efficiencies against the thermal efficiencies estimated from the numerical model for PVT-1 and the PVT-2 respectively. The closer the red dots are to the blue line (45° slope), the better the correlation between the measured thermal efficiency and the modelled thermal efficiency is. This statistical dependence can be described by Pearson correlation coefficient which shows the strength and the direction of a linear relationship between two variables. The value of the correlation coefficient is 1 for an ideal match, meaning all the red dots lie on the blue line, and the model fits the measurements perfectly. A correlation coefficient in the range 0.9-1.0 signifies a very strong relationship between the two variables. In this case, correlation coefficients are 0.90 and 0.98 for PVT-1 and PVT-2 respectively. The scatter around the blue line can be explained by

the fact that unlike the inherent assumption in the numerical model, outdoor conditions are unsteady, and due to the thermal mass of the collector it is never operating at steady state.

However, when the cumulative thermal yield from the numerical model was compared against the cumulative yield from the measured data points, the model underestimates the cumulative thermal yield by 1.9% and 2.9% for PVT-1 and PVT-2 respectively. Since these measurement points are randomly distributed over an entire year, this conclusion can be expanded to the annual yield. Therefore, the proposed numerical model can be used with a good accuracy for the annual thermal yield estimation.

Figure 15: Modelled vs measured outdoor thermal efficiency - PVT-1

Figure 16: Modelled vs measured outdoor thermal efficiency - PVT-2

Finally, collector performance parameters were calculated according to the method described in section 4.5. Table.3 shows the performance parameter values corresponding to the best fit curves for the two collectors.

Table 3: Collector performance parameters estimated by numerical model, including 95 % confidence bounds

	PVT-1	PVT-2
$\eta_{0,b}$	0.33	0.66
b_u	0.12	0.06
b_1	4.3 ± 0.2	7.4 ± 0.1
b_2	0.71 ± 0.12	1.65 ± 0.06

Thermal efficiencies calculated from these parameters (in accordance with eq.6) were compared against outdoor measurement data. Results are shown in Fig.17 and Fig.18. Correlation coefficients are 0.89 and 0.97 for PVT-1

and PVT-2 respectively. The thermal efficiency curve underestimates the cumulative thermal yield by 1.3% and 0.4% relative to the measured cumulative yield, for PVT-1 and PVT-2 respectively. Therefore, thermal efficiency curves derived from the numerical model are a good simplification to accurately estimate the thermal yield of the two collectors while requiring much less computational effort.

Figure 17: Thermal efficiency curve vs outdoor measurements - PVT-1

Figure 18: Thermal efficiency curve vs outdoor measurements - PVT-2

6. Conclusions

Two different unglazed PVT liquid collectors from different manufacturers were tested indoors under a compact solar simulator as well as outdoors. Panels were placed at a distance of 10 cm from the simulator surface. Due to the close proximity to the hot simulator surface, panels receive higher long wave radiation which cannot be estimated by the formulae provided in the testing norm ISO 9806. Moreover, the simulator has a slightly different spectral distribution than the sun which again effects the PVT collector performance. It was demonstrated that in indoor tests the thermal performance was overestimated by 5.6% and 11% for PVT-1 and PVT-2 respectively, when compared against the outdoor measurements. However, by making a long wave radiation correction (related to artificial sky reflections) and a spectral correction (related to PV yield mismatch) true thermal performance of the two PVTs was obtained. The method proposed in ISO 9806 norm, for calculation of long wave radiation in the plane of the collector, is only

suitable for large solar simulators. For compact solar simulators, in addition to the emissions from the artificial sky, reflections from the sky back to the collector must also be taken into account while calculating the long wave radiation. Because of this reflection the long wave radiation in the plane of the collector is influenced by the performance characteristics of the collector being tested. In case of PVT collectors, the spectral distribution of the solar simulator also affects the thermal performance and suitable corrections to the measured performance curves must be made.

A detailed numerical model was developed for the two PVT collectors to simulate the performance of the two PVT collectors in indoor as well as outdoor conditions. This model can be used to investigate design improvements to enhance the heat transfer or to reduce the heat loss in the two collectors. It was demonstrated that the proposed model can be applied to any kind of solar simulator testing (large or compact), correcting for the artificial conditions introduced by the simulator.

The numerical model accurately estimates the indoor and the outdoor thermal efficiencies of the two PVT collectors within 0.9% accuracy when compared against the indoor measurements. For outdoor measurement, the difference between the individual measured points and corresponding simulated data is higher, primarily due to unsteady outdoor conditions. However, these dynamic effects cancel out while estimating the cumulative thermal yield. Therefore, the proposed model can be used for accurate estimation of annual thermal yield within an accuracy of 2.9%. This numerical model can be further used to derive the thermal efficiency curve for the PVT collector, which can be used for system simulations and for annual yield calculations with less computational effort.

In conclusion, the numerical model and the indoor testing can be used together to derive accurate outdoor thermal performance characteristics for the unglazed PVT collectors. The method proposed in this paper is able to determine the thermal performance curve of unglazed PVT collectors without the outdoor tests which can save time and effort that goes in the outdoor testing.

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