

Experimental evaluation of a 3-D dynamic solar-thermal collector model under time-varying environmental conditions

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Abstract:

Reliable dynamic models are required for the correct prediction of the performance of solar-thermal collectors under variable solar-irradiance conditions. In this paper we present a 3-dimensional (3-D) dynamic thermal model applied to three different collector geometries: a flat plate collector (FPC), an evacuated tube collector (ETC), and also a hybrid photovoltaic-thermal (PVT) collector. Results from the model are evaluated against real data from a series of dynamic and steady-state experiments performed in Limassol, Cyprus and London, UK. The 3-D model equations are summarised and the test apparatuses and procedures are described. In the transient response tests, the model is found to under-predict the time constant for the ETC and PVT collectors by 35-55%, while for the simpler FPC the time constant is under-predicted by 20-35%. The collector model is also implemented into a wider domestic hot-water system model that includes a hot-water storage tank, in order to assess performance predictions over a diurnal operating period on an intermittently cloudy day. The results are compared to a single-node quasi-steady state model that uses the collector steady-state efficiency coefficients and a single-node dynamic model that uses a lumped collector thermal capacity (determined using experimental and calculation-based methods in the European Standard for solar collector testing). The 3-D model is shown to provide promising results that are within the range predicted by the two single-node dynamic models. For the PVT collector simulated under intermittent conditions, the predicted net daily energy gain to the store is found to be within 2% of experimentally obtained results. By comparison, a quasi-steady state model based on the collector's steady-state efficiency curve is found to over-predict the thermal energy gain to the store by 8% over the same operating period.

Keywords:

Solar thermal, Dynamic model, Heat transfer, Domestic hot water, Hybrid PVT, System performance.

1. Introduction

Non-concentrating and low-concentration solar thermal systems have the potential to provide thermal energy for water and space heating as well as for low-temperature thermodynamic heat-engine cycles such the organic Rankine cycle (ORC), making them a low-cost and versatile technology for the harnessing of renewable energy [1,2]. The dynamic behaviour of solar thermal collectors has an important bearing on their annual performance that varies across different geographical settings. Year-round intermittency of direct solar irradiance due to cloud interruption is a dominant feature of northerly European climates such as the UK. Southern European locations such as Cyprus on the other hand experience clear-sky conditions for the majority of the year, and solar hot water systems are widely installed on most homes [3]. Steady-state testing and characterisation of solar collectors according to the standardised method in EN 12975-2 [4] is well established in Europe. As a result, steady-state performance data is published as standard by manufacturers and testing institutions and is freely available for comparison via various online resources such as the Institute for Solar Energy database [5]. The European Standard also includes experimental and calculation-based procedures for characterising the dynamic performance of solar collectors. The calculation-based procedure is typically preferred in practice due to its greater simplicity although its accuracy has been a matter of critical discussion in subsequent reviews [6]. The experimental procedure on the other hand requires suitable conditions when performed outdoors and can therefore be costly and time-consuming [7]. In

an earlier paper, Guarracino et al. [8] presented a 3-D dynamic numerical model of a hybrid PV-thermal (PVT) collector and used it to predict the annual thermal and electrical outputs for operation in a UK setting. The thermal component of the model can be adapted to a wide range of solar collector designs and geometries, making it promising as a flexible design and optimisation tool. Furthermore, the ability simulate dynamic behaviour under variable climate conditions presents an alternative/companion approach to the aforementioned dynamic characterisation methods with their inherent limitations and difficulties. In this paper we present three variants of the 3-D dynamic solar collector model adapted to different geometries based on real collectors, namely a flat plate thermal collector (FPC); a photovoltaic-hybrid collector (PVT); and an evacuated tube collector (ETC). For each model variant the collector is simulated under steady-state and dynamic solar irradiance conditions and key performance parameters are compared with those predicted by employing the experimental and calculation methods described in EN12975-2.

2. Methodology

In this section the 3-D numerical model is described for the three different solar collector geometries, followed by a description of the experimental procedures used to evaluate the model under steady-state and dynamic conditions. Finally a simple hot water system model is presented as a tool for use with the collector model to assess the daily thermal energy yield in a dynamic climate simulation.

2.1. 3-D dynamic thermal model

The numerical model solves the energy balance at each component of the collector (glass, thermal absorber, pipe and fluid, discretized as in Fig. 1). The following assumptions are used in the model: (i) thermophysical properties are constant; (ii) the collector is well insulated (i.e. the thermal losses at the collector edges and manifold are neglected); (iii) the temperature between two pipes for the flat plate PVT and thermal collectors is symmetrical; (iv) the circulating fluid flow rate is evenly distributed between the individual tubes of the ETC and between the riser pipes of the FPC and PVT collectors. The model takes into account the thermal capacity of the collector components and the energy balance is solved numerically at each finite element. The main component energy balance equations are summarised below (for the complete set of detailed equations refer to Guarracino et al. [8]):

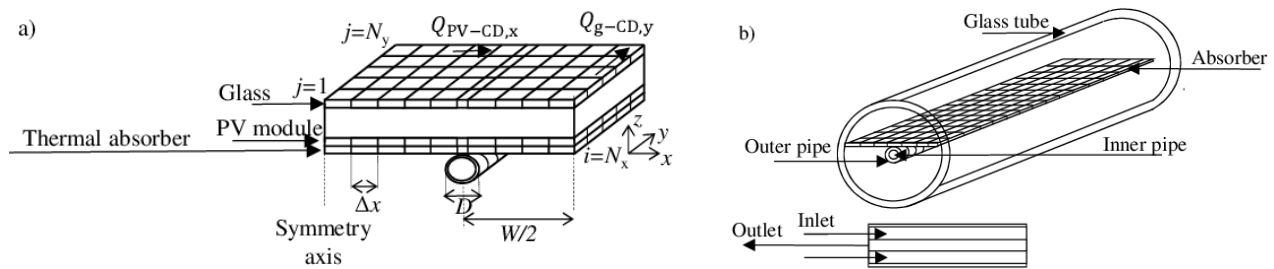


Fig. 1. a) Schematic of the PVT collector; each collector component is discretized into nodes along the flow direction y , and in the x direction. b) Schematic of the ETC. Each component of the ETC (glass tube, pipes and copper absorber) is discretized into nodes along the flow direction y , and in the x direction

Equation (1) is the energy balance for the glass cover which considers conduction between adjacent nodes in the x and y planes, radiative losses to the sky, convective losses to ambient air, convective and radiative heat transfer between the glass and the thermal absorber (or the PV module in the case of the PVT collector), and the fraction of incident irradiance absorbed by the glass cover. Equation (2) is the energy balance for the thermal absorber which considers radiative and convective heat losses to the glass, conduction to the pipe/riser tube (described with the thermal resistance R_p), conduction through the rear insulation and convection to the environment (described with the combined thermal resistance R_{loss}), and the fraction of irradiance which is transmitted and then absorbed by the thermal absorber. Equation (3) is the energy balance for the pipe which considers conduction along the flow

direction y , convection to the fluid and conduction from the thermal absorber. Equation (4) is the energy balance at the fluid node, which describes the convective heat transfer between the pipe wall and the fluid, and the enthalpy rise of the fluid between each node in the flow direction. The heat transfer coefficient correlations are taken from Incropera et al. [9] and Buchberg et al. [10], and are given in Eqs (5-8). Free convection in Eq. (5) is modelled as that for the top of an inclined plate.

$$M_g c_g \frac{dT}{dt} = \dot{Q}_{g-CD(x,y)} - A_{xy} \varepsilon_g \sigma (T_g^4 - T_{sky}^4) - A_{xy} h_{top} (T_g - T_a) + A_{xy} \frac{\sigma(T_a^4 - T_g^4)}{\frac{1}{\varepsilon_g} + \frac{1}{\varepsilon_A} - 1} + \quad (1)$$

$$A_{xy} h_{gap} (T_A - T_g) + \alpha_g G ;$$

$$M_A c_A \frac{dT}{dt} = \dot{Q}_{A-CD(x,y)} - \frac{A_{xy} \sigma (T_A^4 - T_g^4)}{\frac{1}{\varepsilon_g} + \frac{1}{\varepsilon_A} - 1} - A_{xy} h_{gap} (T_A - T_g) - \frac{A_{xy} (T_A - T_p)}{R_p} - \frac{A_{xy} (T_A - T_a)}{R_{loss}} + \overline{\tau} \alpha G ; \quad (2)$$

$$M_p c_p \frac{dT}{dt} = \dot{Q}_{p-CD(y)} + A_{xy} k_b / \delta_b (T_A - T_p) - \pi D h_f dy (T_p - T_f) ; \quad (3)$$

$$M_f c_f \frac{dT}{dt} = \pi D h_f dy (T_p - T_f) + \dot{m} c_f (T_i - T_o) ; \quad (4)$$

$$h_{top} = \sqrt[3]{h_w^3 + h_{free}^3}, \quad \text{where } h_w = 2v_w + 3.8 ; \quad (5)$$

$$h_{gap} = \frac{k_{air}}{\delta_{gap}} \left[1 + 1.44 \left(1 - \frac{1708}{Ra \cos \gamma} \right)^* \left(1 - \frac{1708 (\sin 1.8 \gamma)^{1.6}}{Ra \cos \gamma} \right) + \left(\left(\frac{Ra \cos \gamma}{5830} \right)^{\frac{1}{3}} - 1 \right)^* \right] ; \quad (6)$$

$$h_f = 4.36 \frac{k_f}{D_H}, \quad \text{for } Re < 2300 ; \quad (7)$$

$$h_f = 0.23 Re^{0.8} Pr^{0.4}, \quad \text{for } Re > 2300 . \quad (8)$$

For the special case of the PVT collector, a PV module is laminated on top of the thermal absorber which will absorb the incident solar radiation, converting part to electricity and the rest to heat. The fraction of irradiance converted to electricity is $1 - \beta(T_{PV} - T_0)$, where T_{PV} is the PV temperature, β the temperature coefficient (0.04 K^{-1} for c-Si cells), and η_0 the electrical efficiency at a reference temperature T_0 . For the special case of the ETC collector, each outer glass tube has been divided into two parts with the top part radiating to the sky while the bottom part radiates to the ground. The energy balance is then applied to the two sides of the glass tube. The cold fluid enters through the outer annulus of the coaxial absorber while the hot fluid exits from the inner tube as shown in Fig. 1 (b). The manifold pipe is also of a coaxial configuration, allowing for an even pressure drop and flow distribution across each tube.

2.2. Experimental methods

2.2.1 Apparatus

This section provides descriptions of the two solar collector testing configurations used to obtain experimental data for evaluation of the 3-D numerical model. The first (Fig. 3 (a)) is located at the Technical University of Limassol, Cyprus, while the second is located at Imperial College London, on the roof of a building belonging to the Department of Chemical Engineering (Fig. 3 (b-c)). In the experiments, the Limassol apparatus is used to test the FPC and PVT collector, while the London apparatus is used to test the ETC. The details of the three collectors are listed in Table 1.

2.2.1.1. Limassol testing apparatus

The apparatus consists of collector, pump and storage tank in a closed-loop configuration. In this configuration, water from the tank is directly heated in the collector and returned to the tank. Alternatively, if constant inlet temperature to the collector is required as in the transient performance test, the system can be configured in “open-loop” to directly take water from the utility mains bypassing the storage tank and discharging the heated water to drain. Inlet and outlet

temperatures to the collector are measured using K-type thermocouples. Fluid flow-rate is measured using a turbine flow meter and can be adjusted by means of a needle valve. Solar irradiance is measured using an Eppley PSP pyranometer, installed in the plane of the collector.



Fig. 3. Test setup: a) Solar collector testing apparatus with storage tank in Limassol used for testing of FPC and PVT collector (shown); b) ETC collector; and c) control apparatus including pump, electric heater, flow meter and temperature controller used in the London testing apparatus

2.2.1.2. London testing apparatus

This testing apparatus is based on the configuration described in Kreith and Kreider [11]. The system features a closed-loop configuration with a circulation pump, but does not include a storage tank. Inlet temperature to the collector is controlled by means of an electrical heater for heat addition and a fan-assisted heat exchanger for heat rejection. The circulating fluid is a mixture of water and propylene glycol (to avoid freezing during low ambient temperatures in winter). Inlet and outlet fluid temperatures are measured using T-type thermocouples. Fluid flow-rate is measured using a turbine flow meter and can be adjusted by means of a pump speed controller or bypass valve. Solar irradiance is measured in the plane of the collector using a pyranometer by Davis Instruments.

Table 1: Solar collector properties, taken from manufacturers data [12–14]

	FPC	PVT	ETC
Manufacturer/model	Elcora 1.5	Solimpeks Powertherm	Thermomax DF 100
Aperture area, m ²	1.44	1.42	2.15
Mass (dry), kg	30.2	34.4	54.8
Fluid capacity, L	3.1	1.2	3.8
Number of risers/tubes	7	14	20
Riser/tube diameter, mm	15	8	8 (inner), 12 (outer)
η_0 ,	0.67	0.486	0.773
a_1 , m ² K W ⁻¹	4.859	4.028	1.43
a_2 , m ² K ² W ⁻¹	0.012	0.067	0.0059

2.2.2 Steady-state testing

For the steady state test, data points are obtained over a range of operating temperatures that are representative of the range of the collector. Requirements regarding incidence angle, collector flow-rate, minimum solar irradiance intensity and allowable variations in the steady-state measured parameters are followed according the specifications in EN 12975-2. Instantaneous solar collector efficiency is determined as follows:

$$\eta_c = \dot{m}c_f(T_o - T_i)/GA = \eta_0 - a_1T_m^* - a_2GT_m^* . \quad (9)$$

In order to determine the standard solar collector efficiency coefficients η_0 , a_1 , a_2 according to EN12975-2 the instantaneous collector efficiency is plotted against the reduced mean temperature $T_m^* = (\bar{T}_c - T_a)/G$, where $\bar{T}_c$ is the linear average of the measured inlet and outlet temperatures, and a second-order least-squares fit is applied to the data.

2.2.3 Transient performance testing

EN 12975-2 [4] contains two alternative methodologies for obtaining the parameters that characterise the transient performance of solar collectors. These parameters are the time constant τ and the effective thermal capacity C . An experimental method is described for the time constant, while the effective thermal capacity may be obtained either by an experimental method or a calculation procedure. Both approaches will be compared in this study.

The time constant is obtained by exposing the collector to a step-change in solar irradiance and recording the change in collector temperature with time between the initial state (State 1) and final state (State 2). The final state is identified when the outlet temperature of the collector is no longer observed to change significantly with time ($< 0.05 \text{ K min}^{-1}$, as required by EN 12975-2), at which point the collector is approximated to be in steady state. The step-change in irradiance is achieved by initially shielding the collector from solar irradiance using a reflective cover. A steady fluid inlet temperature is required for the duration of the tests, approximately equal to the ambient temperature. The time constant τ is the time taken for the transition from the initial state to the instant at which the temperature rise through the collector is as follows:

$$(T_o - T_a)_\tau = (1 - e^{-1})[(T_o - T_a)_2 - (T_o - T_a)_1]. \quad (10)$$

The effective thermal capacity parameter meanwhile provides a simplified approach to transient modelling of a solar collector by a modification of the steady state efficiency equation to include an additional time-derivative term. In this approach, the useful thermal power extracted by the solar collector at a particular instant in time is approximated by:

$$\dot{m}c_{p,f}(T_o - T_i) = \eta_0 A(K_{\theta,b}G_b + K_{\theta,d}G_d) - a_1 A(\bar{T}_c - T_a) - a_2 A(\bar{T}_c - T_a)^2 - C \frac{d\bar{T}_c}{dt}. \quad (11)$$

A simple approximation method is presented in EN 12975-2 for estimating the effective thermal capacity that requires only the mass of the individual material components of the collector to be measured. The effective thermal capacity of each component is calculated as the product of its mass, specific heat capacity, and a weighting factor p that describes the degree of thermal interaction it has with the circulating fluid. Thus the metal absorber is designated a high weighting factor while the insulation and external glazing layer are designated low weighting factors. The total effective thermal capacity of the collector is found by summing the values for each n element:

$$C_{wt} = \sum p_n M_n c_n. \quad (12)$$

In the standard, recommended weighting factors are provided only for a simple example of a glazed FPC which includes the absorber, insulation, glazing and heat transfer fluid, but not the frame or fixing brackets which are assumed to be thermally isolated from the heat transfer fluid. Component weightings applying to other collector designs (for example the manifold tube and casing for an ETC) are not provided in the standard. This weighted-component calculation method is widely used for determining the effective thermal capacity and is the approach used in the official test reports for all three of the collectors compared in this study. However, an alternative experimental method is also described in EN12975-2 that uses the same step-response procedure as for the time constant test. By integrating and rearranging Eq. (11):

$$C_{exp} = \frac{A\eta_0 \int_{t_1}^{t_2} G dt - \dot{m}c_{p,f} \int_{t_1}^{t_2} \Delta T_f dt - AU \left[\int_{t_1}^{t_2} (T_i - T_a) dt + \frac{1}{2} \int_{t_1}^{t_2} \Delta T_f dt \right]}{\bar{T}_{c,2} - \bar{T}_{c,1}}. \quad (13)$$

Results from both methods are discussed and compared to 3-D dynamic model results in Section 3.3.

2.3. Hot-water system model

The 3-D dynamic numerical model will be compared to a quasi-steady model based on the steady state efficiency curve equation in Eq. (9) and a 1-D dynamic model based on the collector efficiency equation with the added time-derivative term in Eq. (11). For the comparison, the FPC, PVT and ETC collector models are implemented into a simple hot water system model, consisting of a pump, collector and storage tank (see Fig. 4) based on the closed-loop arrangement of the Limassol testing apparatus. This allows the models to be compared in terms of their predictions of the total thermal energy transferred to the tank over a diurnal period. Two climate days will be compared: a clear day with an approximately parabolic solar irradiance profile and a cloudy day with short periods of intermittent cloud cover causing a high degree of variability in irradiance intensity. Climate data used in the simulation was collected from the Limassol testing apparatus at a 1-second time resolution. For the PVT system model, the simulation results will be compared to experimental performance data for the real PVT collector operating in closed loop configuration with the storage tank on the days in question, allowing further evaluation of the collector model integrated into the wider system model.

The default system model uses a fully-mixed (lumped) model for the storage tank. This will be compared to an alternative tank model in which thermal stratification in the tank is modelled using a 1-D finite difference approach. For the latter model an energy balance is performed at each N node of the tank. The thermal energy balance at each node is:

$$M_{t,N}c_p \frac{dT_{t,N}}{dt} = \dot{m}c_f(T_{t,N+1} + T_{t,N-1} - 2T_{t,N}) - UA_{t,N}(T_{t,N} - T_a) + \left(\frac{k + \Delta k}{\Delta H}\right)A_c(T_{t,N+1} - 2T_{t,N} + T_{t,N-1}). \quad (14)$$

Equation (14) is written for a discretization where each N node has the same height ΔH . The thermal conductivity Δk is the thermal conductivity that accounts for the mixing at each node interface and for thermal conductivity at the tank wall [15-16]. For a steel tank with a diameter of 0.30 m, Δk is 2.3 W/m K. For the top node, $T_{t,N+1}$ is the collector outlet temperature, the fluid leaves the tank at the temperature $T_{t,1}$, which is also collector the collector inlet temperature.

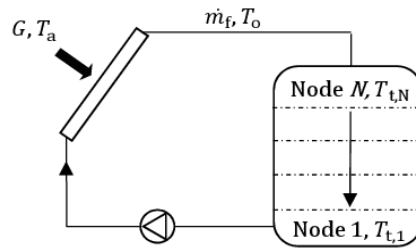


Fig. 4. Schematic of the solar-thermal hot water system model with stratified hot water storage tank, representing the closed-loop variant of the Limassol testing apparatus

The system model parameters are chosen to simulate those of the experiment. The UA value of the tank is 1.9 W/K, the tank volume is 100 L and the fixed flow rate is set to 1.68 L/min. The total energy transferred to the hot water cylinder over the daily operational period is calculated as:

$$Q_{t,net} = M_t c_p (T_{t,final} - T_{t,initial}). \quad (15)$$

3. Results and discussion

3.1. Steady-state testing

The plots in Figs. 5 and 6 show a comparison between the numerical model values and the individual test points for the PVT and ETC collectors from the official test reports and from a set of

comparison tests performed using the apparatuses in Limassol and London respectively. Local testing was not performed for the FPC collector as part of this study; however the numerical model is compared to the test report values for this collector in Fig. 7.

In the case of the ETC collector, the test report conditions were limited to the typical range encountered in water heating applications, with the inlet temperatures in the range 21–80 °C. In fact, evacuated tube collectors are capable of operating with reasonable efficiency at higher temperatures than this, which is of interest for alternative applications such as thermodynamic heat engines or absorption refrigeration machines. Therefore, in the present study, additional steady-state efficiency data is obtained for inlet temperatures up to 112 °C. The experimental data shows good agreement with the manufacturer's efficiency curve across the full range of test conditions.

The 3-D numerical model was used to simulate the performance of each collector under steady state conditions. The environmental and operational input conditions were chosen where possible to replicate the steady state test conditions reported in the official test report for each collector [12–14]. The numerical model shows a reasonable agreement with the experimental efficiency values in the manufacturers test reports for each of the three collectors. The relative deviation in the predicted thermal efficiency across the full range of values (reported to a 95% confidence interval) is 8% for the FPC, 16% for the PVT collector and 12% for the ETC. At higher values of T_m^* the numerical model is found to deviate further from the experimental data, under-predicting the reduction in collector thermal efficiency with increasing temperature.

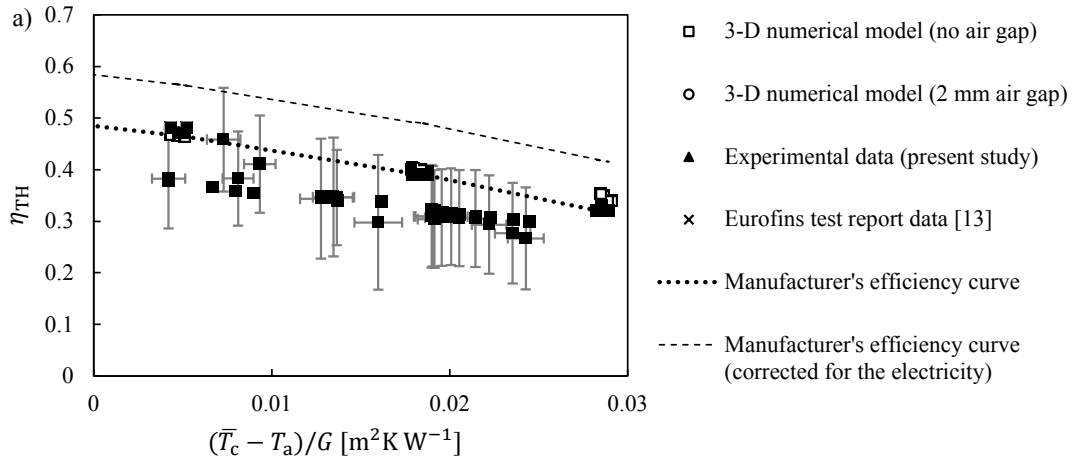


Fig. 5. Comparison of steady-state thermal efficiency data generated by the 3-D numerical model with experimental data from testing in Limassol, and from the official test report

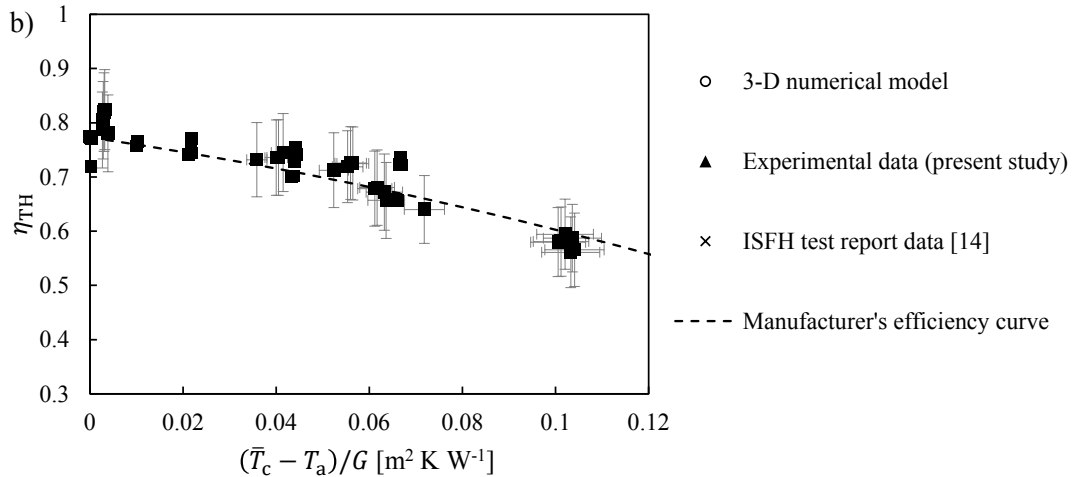


Fig. 6. Comparison of steady-state thermal efficiency data generated by the 3-D numerical model with experimental data from testing in London, and from the official test report

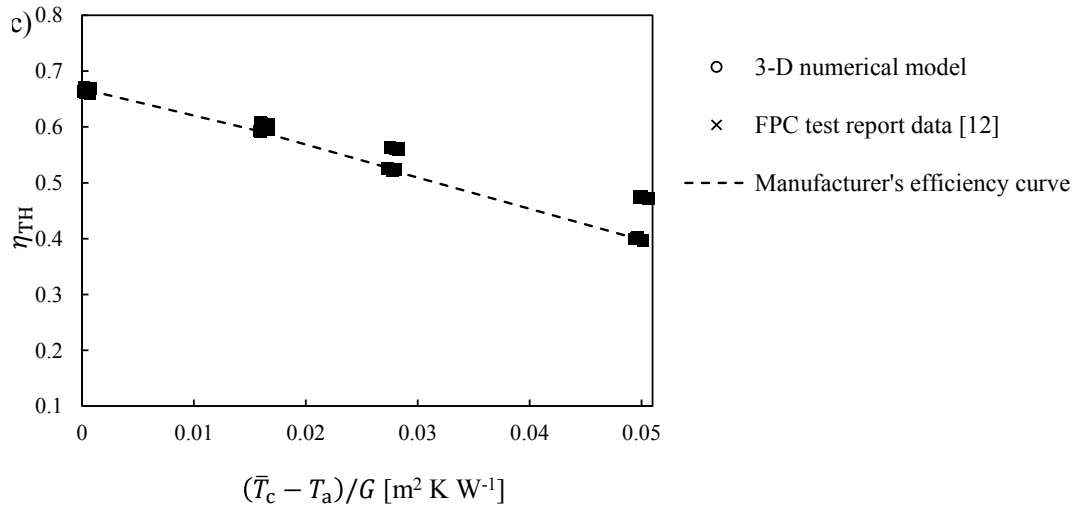


Fig. 7. Comparison of steady-state thermal efficiency data generated by the 3-D numerical model with experimental data from the official test report

For the PVT collector, some disagreement between the experimental data and the manufacturer's test report data was expected because electrical power was not being extracted from the PV module during the Limassol tests (while the manufacturer's data represents a continuous electrical output operation at the maximum power-point). This should result in a higher thermal efficiency for the Limassol tests than that predicted by the manufacturer's thermal efficiency curve. However, the opposite was found to occur, with the Limassol experiment results showing a significantly lower thermal efficiency than expected. Upon inspection of the collector it was found that there was a poor thermal contact between the PV module and the copper sheets that form the thermal absorber plate, due to insufficient application of thermal grease between the layers. This situation can be emulated by modifying the model to include a small air gap between the PV module and the absorber plate. By setting the thickness of the air gap to 2 mm the agreement between the numerical model and the Limassol test data is found to significantly improve, as shown in Fig. 5.

3.2. Time-constant and effective thermal-capacity

The experimental conditions for the step-response tests are presented in Table 2. The same values are used as inputs to the numerical model, and the experimental and numerically obtained values of τ for each collector are compared in Table 3. Experimental values of C are also presented in the table, and compared to those provided in the official manufacturer's test reports, which in all cases are obtained using the non-experimental weighting-factor method discussed in Section 2.1.2.

Table 2. Experimental conditions for the transient response tests; the deviation from the mean measured value of each parameter over the period of the test is given to a 95% confidence interval

	Collector	Location	$\dot{V}$, l/min	G , W/m ²	T_a , °C	T_i , °C
Test #1	FPC	Limassol	1.7 ± 0.1	950 ± 20	20.5 ± 0.6	21.9 ± 0.2
Test #2	FPC	Limassol	1.7 ± 0.1	950 ± 30	20.2 ± 0.6	20.9 ± 0.2
Test #3	PVT	Limassol	1.7 ± 0.1	930 ± 20	22.6 ± 0.6	25.0 ± 0.2
Test #4	PVT	Limassol	1.4 ± 0.1	910 ± 20	22.1 ± 0.4	26.2 ± 0.4
Test #5	PVT	Limassol	1.7 ± 0.1	930 ± 40	18.3 ± 0.4	20 ± 0.4
Test #6	ETC	London	2.2 ± 0.1	840 ± 10	16.3 ± 0.8	15.1 ± 0.6
Test #7	ETC	London	2.3 ± 0.1	900 ± 20	17.6 ± 0.8	16.6 ± 1.0
Test #8	ETC	London	2.5 ± 0.1	950 ± 90	20.0 ± 0.7	19.8 ± 1.2
Test #9	ETC	London	2.5 ± 0.1	910 ± 50	19.6 ± 1.0	19.1 ± 0.6

Table 3. Transient performance tests results; the time constant is compared to the value obtained using the 3-D dynamic numerical model, while the effective thermal capacity is compared to the manufacturer's value obtained using the weighted-component method

	Collector	Experimental results		Numerical model *		Manufacturer value
		τ , s	C_{exp} , kJ K ⁻¹	τ , s	C_{wt} , kJ K ⁻¹	
Test #1	FPC	115	13.9	75		14.9
Test #2	FPC	98	15.8	79		
Test #3	PVT	545	292.9	140 (272)		
Test #4	PVT	605	195.6	191 (332)		20.3
Test #5	PVT	516	165.7	179 (269)		
Test #6	ETC	283	77.8	117		
Test #7	ETC	292	65.7	107		20.1
Test #8	ETC	189	68.7	96		
Test #9	ETC	204	82.2	82		

* Value reported in brackets is the PVT model modified to include the 2 mm air gap between PV module and absorber

In all cases the values for τ determined using the numerical model are smaller than the experimental values. The temperature response of the collectors is plotted against time in Figs. 8 and 9. For the PVT collector, results are shown for the model with and without the air gap correction described in Section 3.1. For the FPC, the numerical model under-predicts the response time of the collector by between 20-40%. The FPC effective heat capacity value calculated from the experimental data also agrees very closely with the manufacturer's value, which is obtained using the weighted component method. For the PVT and ETC collectors the difference between the numerical and experimental results is more significant. For the PVT collector, the average value of τ obtained from the experimental results is 1-2 times larger than that obtained from the numerical model corrected for the air gap between the PV module and thermal absorber, while for the EPC it is 2-3 times larger. The EPC is more complex in design than the FPC, with a larger number of material components in thermal interaction. For the PVT and the ETC collectors, the values of C calculated using the experimental data are also significantly larger than the values presented in the manufacturer's test report found by summing the weighted component thermal capacities.

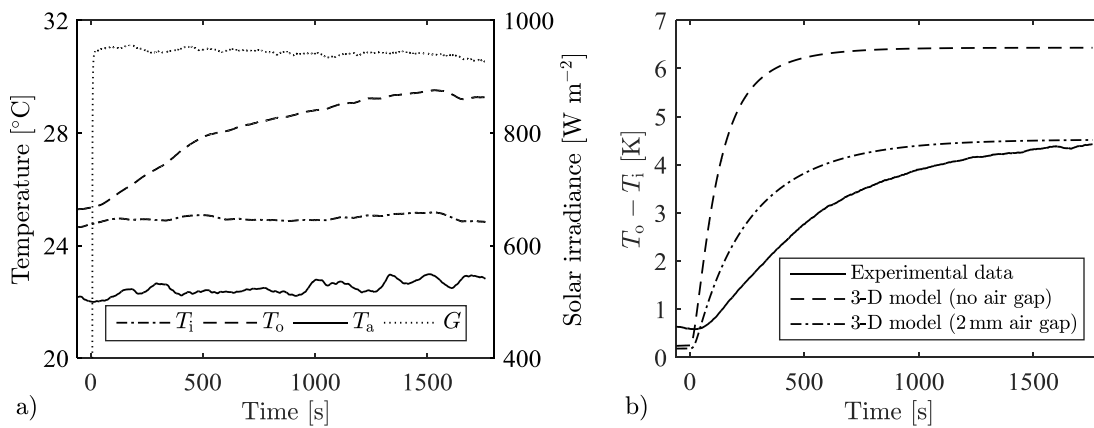


Fig. 8. Step response of the PVT collector, Test #3: a) collector inlet and outlet temperatures, ambient temperature and solar irradiance (see Table 2 for test conditions); b) comparison of the numerical model with experimental results, showing collector temperature rise ($T_o - T_i$). Numerical results are presented for the model with and without a 2 mm air gap between the PV layer and copper absorber

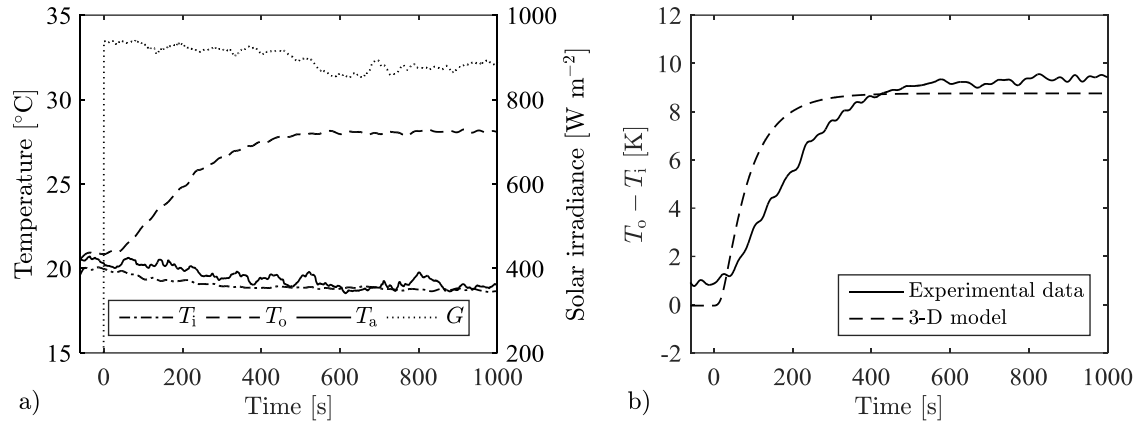


Fig. 9. Step response of the ETC collector, Test #4: a) collector inlet and outlet temperatures, ambient temperature and solar irradiance (see Table 2 for test conditions); b) comparison of the numerical model with the experimental results, showing collector temperature rise ($T_o - T_i$)

3.3. Factors influencing the effective thermal-capacity

It was shown in Section 3.2 that the time constant of the PVT and ETC collectors was significantly underestimated by the numerical model when compared with the experimental results. Furthermore, the effective thermal capacity values determined from the experimental data are less than those obtained from the weighted component calculation method in the official test reports for each collector. In the latter method, only the thermal capacity of certain components are included in the calculation, based on their estimated degree of thermal interaction with the heat transfer fluid. In Table 4, the full list of material components making up the ETC and test assembly are shown. The mass of each component was measured and the fluid capacity of the collector tubes and manifold piping were also measured. The recommended weighting factors for the components included in the effective heat capacity calculation according to EN 12975-2 are included where available.

Table 4. ETC component masses (determined by weighing), specific heat capacities and weighting factors (according to EN 12975-2)

Component	Material	M , kg	c_p , kJ kg ⁻¹ K ⁻¹	p , - [1]
Absorber plate/tubing	copper	12.7	0.4	1
External glazing	borosilicate glass	26.5	0.8	0.014
Insulation	foamed melamine resin	0.06	1.4	0.5
Heat transfer fluid	water/propylene glycol mixture	4.3	3.6	1
Manifold pipe	copper	2.1	0.4	-
Manifold casing	aluminium	6.0	0.9	-
A-frame/brackets	aluminium	18.6	0.9	-
Instrumentation sections	brass/copper	5.2	0.4	-

Components such as the frame and the manifold casing are assumed to be thermally isolated from the collector fluid, but thermal bridging due to poor installation practices can render this assumption invalid. The glazing is also assumed to have minimal thermal interaction; however a loss of vacuum in one or more tubes will result in convective heat transfer between the copper absorber and the glass. In order to investigate the validity of the assumption that thermal interaction between the fluid and the external/structural elements of the collector (in particular the frame, glazing and manifold casing) is negligible, a series of infra-red images were taken as the circulating fluid is raised to an elevated temperature and then maintained at steady state. Meanwhile, temperatures were measured with thermocouples located at various points on the frame, glazing and manifold casing. The image in Fig. 10 shows that the glazing and manifold casing are reasonably well thermally-isolated from the fluid. Some hotter regions are identified at the points of connection between the tubes and the manifold, and in the instrumentation sections connected at the inlet and outlet (comprising of brass

fittings including bleeding and isolation valves, and ports for temperature and pressure sensors). However thermal bridging to more significant structural elements was not found to occur, and in particular the frame was also found to have little or no thermal interaction with the fluid.

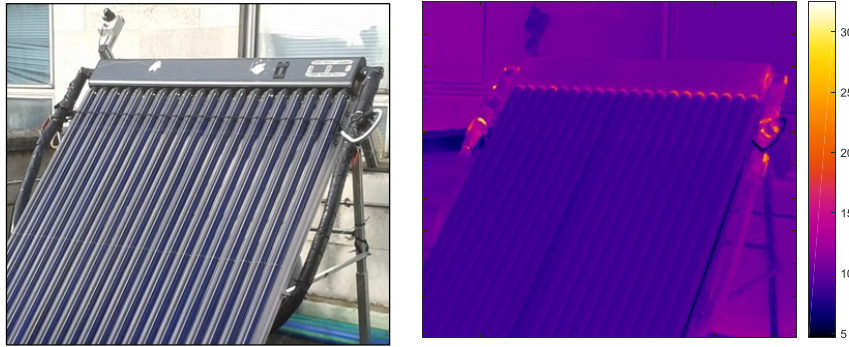


Fig. 10: Infrared image of the ETC in steady state operation with $T_i = 50\text{ }^{\circ}\text{C}$, and $G < 100\text{ W m}^{-2}$

3.4. Hot-water system performance

The results of the hot water system simulation with the 3-D dynamic numerical model for the FPC, PVT and ETC collectors are presented in the Table 5. The results are compared to a quasi-steady state (QS) collector model that uses the steady state efficiency in Eq. (9) to predict the rise in enthalpy of the fluid flowing through the collector at each time instant; and a single-node (lumped) dynamic collector model based on Eq. (11) using the effective thermal capacity values obtained from the experimental and weighted-component methods. Results are presented for a clear-sky day and an intermittently cloudy day. Results for the PVT system model are compared and evaluated using experimental data for the real PVT system. For the 1-node dynamic collector model, two comparison values of effective thermal capacity are used. The first is the manufacturer's value determined using the weighted-component thermal capacities (C_{wt}). The second is the average of the experimentally obtained values (C_{exp}) presented in Table 3. To evaluate the extent to which thermal stratification in the hot-water tank influences the daily system performance, simulations are also performed for a 2-node, stratified tank model and the results compared to those for the default fully-mixed (lumped) tank model. The difference in daily net energy accumulation between the two tank models is found to be $\sim 1\%$ for the system configuration shown in Section 2.3.

For the PVT system, the 3-D dynamic model shows reasonably good agreement with the experimental data. The net thermal energy increase to the tank is predicted to within 2% of the experimental value for the cloudy day and the final tank temperature is predicted to within $1\text{ }^{\circ}\text{C}$, as shown in Fig. 11a. Figures 11b and 11c show that the peaks in instantaneous energy in response to sudden increases in solar irradiance are mostly over-predicted by the 3-D dynamic model, while the delay in the response time to these peaks is consistently under-predicted by 1-2 min in comparison to the experimental results. This supports the findings in Section 3.2 that suggest the thermal inertia of the collector is underestimated by the model. The most pronounced period of discrepancy between the model and the experimental data occurs at the very beginning of the day when solar irradiance is low and the angle of incidence is large. The temperature rise through the collector is over-predicted by 1-2 K during this period.

The single-node collector model simulated with the experimentally obtained effective thermal capacity C_{exp} is shown to under-predict the net thermal energy transfer to the tank by 4% over the cloudy-day simulation period, however the instantaneous peaks in absorbed thermal energy predicted by this model coincide relatively well with the experimental data. The single-node dynamic model using C_{wt} (obtained via the weighted-component thermal capacity method) is found to result in a 6% overestimate in net energy gain compared to the cloudy-day experimental results. This model also shows an appreciable response to the very fast spikes in irradiance (such as that occurring in Fig. 11c at 12:33), which are entirely attenuated by the collector in the experimental results. The quasi-steady model

responds instantaneously to variations in irradiance, as shown in Fig. 11, and results in the largest over-prediction (8% on the cloudy day) in net thermal energy gain to the tank of all the models.

Table 5. Solar hot water system simulation results for cloudy day and clear-sky day showing the net energy gain to the tank over the simulation period; experimental results are shown for the PVT system obtained under the same climate conditions as used in the simulations

	Cloudy day $Q_{t,net}$, MJ			Clear-sky day $Q_{t,net}$, MJ		
	FPC	PVT	ETC	FPC	PVT	ETC
3-D dynamic model	6.72	3.20	15.54	11.92	5.95	26.80
Quasi-steady model	6.20	3.52	13.59	10.85	6.94	23.62
1-node dynamic model (C_{wt})	6.14	3.47	13.34	10.52	6.84	23.16
1-node dynamic model (C_{exp})	6.14	3.12	12.76	10.52	6.16	22.11
Experimental results	-	3.26	-	-	4.81	-

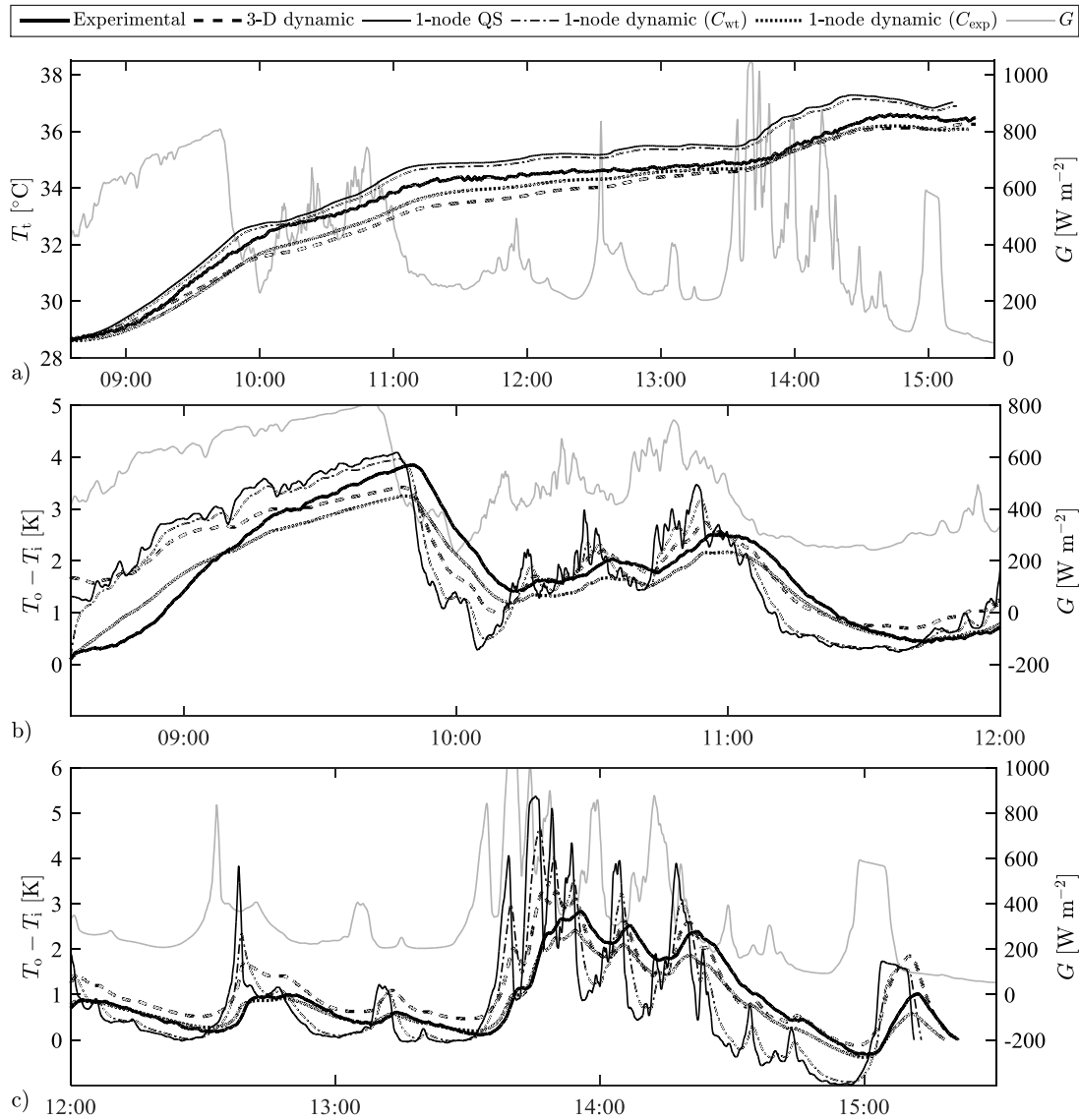


Fig. 11. Diurnal simulation plots for the 3-D dynamic model of the PVT collector, compared to experimental data for the same irradiance profile: a) tank temperature; b) collector temperature rise (morning), c) collector temperature rise (afternoon); also shown are simulation results for the 1-node quasi-steady (QS) collector model and the 1-node dynamic collector model using the lumped collector thermal capacities C_{wt} and C_{exp}

4. Conclusions

A 3-D dynamic solar-thermal collector model was adapted to three different geometries based on real collector designs: a flat plate collector (FPC), an evacuated tube collector (ETC), and also a hybrid PV-Thermal (PVT) collector. Steady-state and transient step-response simulations were performed using the model and these were compared with experimental data obtained from outdoor testing in Limassol, Cyprus and London, UK. The parameters chosen for the comparison were the steady-state efficiency, and the time-constant and effective thermal-capacity of the collector.

The steady-state efficiency predicted by the 3-D numerical model showed good agreement with the experimental data for each of the three collector configurations. In the transient response test the numerical model showed reasonable agreement with experimental results for the simple FPC, but under-predicted the time constant for the ETC and PVT collectors. The latter two collectors are considerably more complicated than the FPC, with features that involve greater modelling complexity such as the optical efficiency and thermal loss mechanisms in the manifold for the ETC, and the PV-thermal absorber interface for the PVT collector. For the effective thermal-capacity, the value obtained using the numerical model was compared to values obtained using the weighted-component calculation method and the transient-response experimental method described in EN 12975-2. The former method produced lower values compared to the predictions of the numerical model, whereas the latter method produced higher values than the same model.

Finally, dynamic simulations over a day-long period of operation under clear-sky and cloudy/intermittent solar conditions were performed for each collector in a closed-loop hot-water configuration with a storage tank. For the cloudy-sky simulations with the PVT collector, the 3-D dynamic model under-predicts the net energy gain by 2% relative to the experimental data. For the clear-sky simulations the 3-D dynamic model over-predicts the net energy gain by 24%, which appears to be due to an underestimation of the thermal losses from the collector at higher temperatures. However for both simulations the results are within the range predicted by 1-node models using the lumped thermal capacities from the methods in EN 12975-2. The numerical model therefore shows strong potential as a versatile tool that can be applied to a range of collector geometries and can predict performance under dynamic conditions to within a margin of error that is comparable to the accepted alternative methods. Further work will focus on experimental validation of the FPC, ETC and PVT collector models under a wider range of operating conditions, and improvements to the modelling of the thermal losses from the collectors at higher temperatures.

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Nomenclature

Symbols

A	area, m^2
a_1, a_2	solar collector steady-state heat loss coefficients, $\text{W m}^{-2} \text{K}^{-1}$, $\text{W m}^{-2} \text{K}^{-2}$
C	effective heat capacity, J K^{-1}
c_p	specific heat capacity, $\text{J K}^{-1} \text{kg}^{-1}$
D	diameter, m
G	irradiance, W m^{-2}
h	heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$
K	solar collector incident angle modifier
M	mass, kg

$\dot{m}$	mass flow rate, kg s ⁻¹
N	number of riser pipes, node
R	thermal resistance, W m ⁻² K ⁻¹
Q	thermal energy, J
$\dot{Q}$	heat flux, W
P_{EL}	electrical power, W
p	weighting factor
T	temperature, K
T_m^*	reduced mean temperature, m ² K W ⁻¹
t	time, s
U	overall heat loss coefficient, W m ⁻² K ⁻¹
x, y, z	Cartesian axis directions

Greek symbols

α	absorption coefficient
β	temperature coefficient, K ⁻¹
γ	collector tilt angle, rad
η, η_0	efficiency, zero-loss efficiency
τ	time constant, s
$\overline{\tau\alpha}$	absorption coefficient of a glazed collector

Subscripts and superscripts

0	reference state
a	ambient air
A	absorber
b	beam irradiance
c	collector
CD	Conduction
d	diffuse irradiance
exp	experimental method
f	fluid
free	free convection
g	glass cover
H	Hydraulic diameter
i	inner, inlet
N	node
o	outer, outlet
p	pipe or tube
PV	solar cell
sky	sky
TH	thermal
t	tank
top	top surface of inclined plate
wt	weighted component method
x, y	x (fin) and y (flow) direction

References

- [1] Markides C.N., Low-concentration solar-power systems based on organic Rankine cycles for distributed-scale applications: overview and further developments. *Frontiers in Energy Research* 2015;3:47.
- [2] Freeman J., Hellgardt K., Markides C.N., An assessment of solar–thermal collector designs for small-scale combined heating and power applications in the United Kingdom. *Heat Transfer Engineering*. 2015;36(14-15):1332-47.
- [3] Kalogirou S.A., Solar Water Heaters in Cyprus: Manufacturing, Performance and Applications. In: *Proceedings of the 4th Congress on Energy Conservation in Buildings and Renewable Energy*, Tehran, Iran. Renewable Energy; 2005 Tehran, Iran.
- [4] European Standard EN 12975–2:2006 Thermal solar systems and components. Solar collectors. Test methods. 2006.
- [5] Institut für Solartechnik, SPF Online Collector Catalogue. Available at: <www.solarenergy.ch> [accessed 27.2.2016].
- [6] IEA Solar Heating and Cooling Programme Task 43: Solar rating and certification procedures. Subtask A: Roadmap of collector testing and certification issues. March 2012. Available at: <<http://archive.iea-shc.org/>> [accessed 27.2.2016].
- [7] Fischer S., Heidemann W., Müller-Steinhagen H., Perers B., Bergquist P., Hellström B., Collector test method under quasi-dynamic conditions according to the European Standard EN 12975-2. *Solar Energy*. 2004;76(1):117-23.
- [8] Guarracino I., Markides C.N., Mellor A., Ekins-Daukes N.J., Dynamic coupled thermal-and-electrical modelling of sheet-and-tube hybrid photovoltaic/thermal (PVT) collectors. *Applied Thermal Engineering*. 2016. [doi: 10.1016/j.applthermaleng.2016.02.056].
- [9] Bergman T.L., Incropera F.P., Lavine A.S., Fundamentals of heat and mass transfer. John Wiley & Sons; 2011.
- [10] H. Buchberg, I. Catton, D.K. Edwards, Natural convection in enclosed spaces – a review of application to solar energy collection, *Journal of Heat Transfer* 1976;98(2):182–188.
- [11] Kreith F, and Kreider J.F., Nonconcentrating solar thermal collectors. In: *Solar Energy Handbook*. McGraw-Hill, New York. 1981.
- [12] Κέντρο Εφαρμογών Ενέργειας, Έκθεση αποτελεσμάτων, Δομική θερμικής απόδοσης (thermal performance) ηλιακού συλλέκτη σύμφωνα με το πρότυπο EN 19975-2:2006 που έγινε από το Εργαστήριο του Κέντρου Εφαρμογών Ενέργειας για λογαριασμό της εταιρείας ΕΛΚΟΡΑ ΑΤΔ. Εργαστήριο Κέντρου Εφαρμογών Ενέργειας, Υπηρεσία Ενέργειας, Υπουργείο Εμπορίου, Βιομηχανίας και Τουρισμού, Λευκωσία, Κύπρος; 2010 Jun. Technical Report No.: Σ.13.04.10.03.
- [13] Eurofins Modulo Uno, Report of Performance Test According to EN12975-2 for Solimpeks Volther Powertherm, Torino, Italy; 2011 Nov. Technical Report No.: M1.11.NRG.0319/43724.
- [14] Institut für Solarenergieforschung GmbH, Report of Performance Test According to EN12975-2 for a Glazed Solar Collector (Thermomax DF 100 20). Test Centre for Solar Thermal Components and Systems, Emmerthal, Germany; 2006 Nov. Technical Report No.: 106-06/D.
- [15] Cruickshank C.A, Harrison S.J., Heat loss characteristics for a typical solar domestic hot water storage. *Energy and Buildings*. 2010;42:1703-1710.
- [16] TRNSYS, A Transient Simulation Program, Ver. 15, University of Wisconsin Solar Energy Laboratory, Madison, WI, 2000.