



Integrated collector storage solar water heater under partial vacuum

Souliotis, M., Papaefthimiou, S., Caouris, Y., Zacharopoulos, A., Quinlan, P., & Smyth, M. (2017). Integrated collector storage solar water heater under partial vacuum. *Energy*, 139, 991-1002.
<https://doi.org/10.1016/j.energy.2017.08.074>

[Link to publication record in Ulster University Research Portal](#)

Published in:
Energy

Publication Status:
Published (in print/issue): 15/11/2017

DOI:
[10.1016/j.energy.2017.08.074](https://doi.org/10.1016/j.energy.2017.08.074)

Document Version
Author Accepted version

General rights

Copyright for the publications made accessible via Ulster University's Research Portal is retained by the author(s) and / or other copyright owners and it is a condition of accessing these publications that users recognise and abide by the legal requirements associated with these rights.

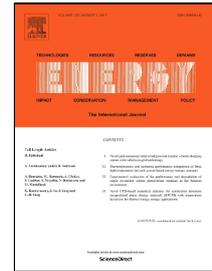
Take down policy

The Research Portal is Ulster University's institutional repository that provides access to Ulster's research outputs. Every effort has been made to ensure that content in the Research Portal does not infringe any person's rights, or applicable UK laws. If you discover content in the Research Portal that you believe breaches copyright or violates any law, please contact pure-support@ulster.ac.uk.

Accepted Manuscript

Integrated collector storage solar water heater under partial vacuum

Manolis Souliotis, Spiros Papaefthimiou, Yiannis G. Caouris, Aggelos Zacharopoulos, Patrick Quinlan, Mervyn Smyth



PII: S0360-5442(17)31450-0
DOI: 10.1016/j.energy.2017.08.074
Reference: EGY 11443
To appear in: *Energy*
Received Date: 03 July 2017
Revised Date: 15 August 2017
Accepted Date: 16 August 2017

Please cite this article as: Manolis Souliotis, Spiros Papaefthimiou, Yiannis G. Caouris, Aggelos Zacharopoulos, Patrick Quinlan, Mervyn Smyth, Integrated collector storage solar water heater under partial vacuum, *Energy* (2017), doi: 10.1016/j.energy.2017.08.074

This is a PDF file of an unedited manuscript that has been accepted for publication. As a service to our customers we are providing this early version of the manuscript. The manuscript will undergo copyediting, typesetting, and review of the resulting proof before it is published in its final form. Please note that during the production process errors may be discovered which could affect the content, and all legal disclaimers that apply to the journal pertain.

Integrated collector storage solar water heater under partial vacuum

*Manolis Souliotis^{1,2,3}, Spiros Papaefthimiou⁴, Yiannis G. Caouris⁵,

Aggelos Zacharopoulos⁶, Patrick Quinlan⁶, Mervyn Smyth⁶

¹*Department of Mechanical Engineering, University of Western Macedonia, Greece*

²*Department of Environmental Engineering, University of Western Macedonia, Greece*

³*Mechanical Engineering Department, Technological Educational Institute of Western Greece, Greece*

⁴*School of Production Engineering and Management, Technical University of Crete, Chania, Greece*

⁵*Department of Mechanical Engineering & Aeronautics, University of Patras, Greece*

⁶*Centre for Sustainable Technologies, Ulster University, Jordanstown, N. Ireland*

Abstract

The work focuses on the experimental study of the heat diode mechanism in an Integrated Collector Storage Solar Water Heater (ICSSWH) for domestic applications. The solar device combines a horizontal cylindrical vessel with an asymmetric reflector trough (Compound Parabolic Concentrator – CPC). The cylindrical storage tank comprises two concentric cylindrical vessels: the outer absorbing vessel and the inner storage vessel. The annulus between the cylindrical vessels is partially depressurized and contains a small amount of water serving as Phase Change Material (PCM), which changes phase (mainly at low temperatures) thus producing vapor and creating a thermal diode transfer mechanism from the outer to the inner surfaces of the vessels. Several experimental results, including uncertainty analysis, are demonstrated through diagrams depicting temperature variations, mean daily efficiency and thermal losses coefficient. Additionally results from the variation of the temperature and the total pressure inside annulus are also presented. The results clearly show that the vapor's pressure plays the most important role regarding the thermal performance of the device.

Keywords: Integrated Collector Storage Solar Water Heaters (ICSSWH); Heat Diode Mechanism; Thermal Behavior; Phase Change Materials (PCM); Compound Parabolic Concentrators (CPC)

*Corresponding author: Tel/Fax: +30 24610 56623

E-mail address: msouliotis@uowm.gr (Manolis Souliotis)

1 1. Introduction

2 Integrated Collector Storage Solar Water Heaters (ICSSWH) belong to the category of low cost
3 solar devices that mainly cover needs for domestic applications. The development of these systems has
4 been recently discussed in detail in the works of Singh et al [1] and Souliotis et al [2]. However,
5 ICSSWH suffer by ambient heat losses, at nighttime, and extended research activities have been focused
6 on the potential improvement of the thermal behavior in ICS systems by reducing the tank's thermal
7 losses. Smyth et al. [3] studied the thermal properties of different types of integrated collector storage
8 solar water heaters during day and night operation, while Henderson et al. studied the thermal
9 performance of flat plate ICS for various inclinations [4]. The thermal performance of ICS systems can
10 be improved using Compound Parabolic Concentrator (CPC) reflectors. In this category of ICS systems,
11 the solar radiation is reflected and collected on the absorbing surface, while the water tank(s) can be
12 directly connected to the water mains [5-7]. Tripanagnostopoulos et al. [8] and Souliotis et al. [9] studied
13 ICS systems combined with reflectors of symmetrical and asymmetrical CPC geometry while
14 significant articles also include computational and experimental studies [10-15].

15 Phase Change Materials (PCM) can improve the thermal behavior of solar thermal systems,
16 especially in the case of incorporated storage tanks mainly by reducing thermal losses. Hailiot et al. [16]
17 studied a system in which the storage tank was covered with a composite PCM, while Eames and
18 Griffiths [17] presented a numerical study of the thermal behavior of an ICS unit using PCM at 65°C.
19 Griffiths et al. [18] studied experimentally the heat retention of ICS systems using PCM slurries, and
20 similarly, Al-Hinti et al. [19] performed an experimental study of a conventional water heating system
21 using Paraffin wax as PCM. In that work, the system advantage, regarding the hot water availability at
22 higher temperature and over and extended time period, was clearly demonstrated. Recently, Chaabane
23 et al. [20] presented a numerical study of the thermal performance of an ICSSWH with a cylindrical
24 water storage tank (with and without PCM), properly mounted in a CPC reflector. The combination of
25 partial evacuation and PCM in an ICS heater was originally presented in the work of De Beijer [21]. In
26 that work, the evaporator consisted in the collector-absorbing surface and the condenser consisted in

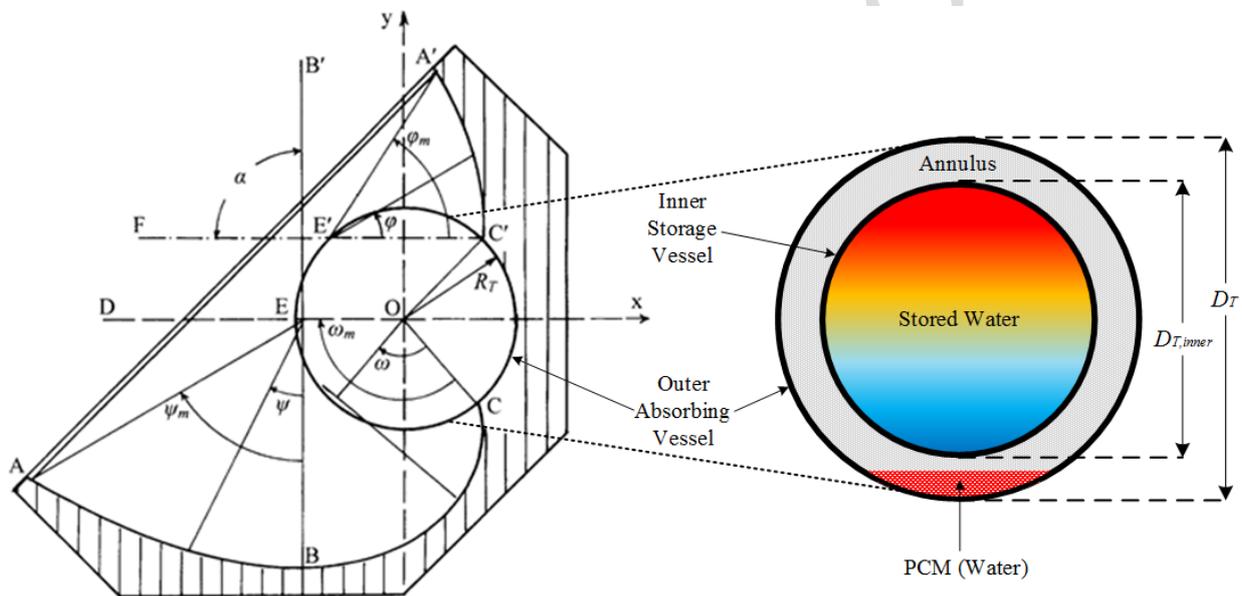
1 the surface of the inner storage vessel. Pugsley et al. [22] have developed a building integrated hybrid
2 solar ICS water heater working on the same principle achieving instantaneous thermal and electrical
3 efficiencies of 60% and 12%, respectively. Rhee et al. [23] experimentally measured the temperature
4 stratification in a solar hot water storage tank using a double chimney device that acts as a thermal
5 diode. In the work of Souliotis et al. [9] the effective water temperature rise during daily operation and
6 heat retention during nighttime operation, for a device working with respective heat diode mechanisms,
7 was discussed.

8 In the present work, an ICSSWH comprising two concentric cylindrical vessels (forming an
9 annulus in which water was incorporating as PCM) and an asymmetric CPC reflector, is experimentally
10 studied. The device was designed focusing on the achievement of both highly time response temperature
11 rise of stored water during the day and low nighttime thermal losses. The scope of the work is the
12 investigation of the influence of the vacuum inside the annulus, both in the temperature rise of the stored
13 water during the daily operation and the heat retention during the non-collection period. A detailed
14 experimental study for two consecutive years was accomplished as the next step of previous work
15 published by the authors [9]. Variation diagrams of the stored water temperature and the total pressure
16 within the annulus during 24-hour operation of the device are presented and discussed in detail. Similar
17 diagrams combining the total pressure within the annulus and the vapor temperature, can provide
18 additional information regarding the heat transfer mechanisms. In addition, the variation of both mean
19 daily efficiency and the thermal night losses coefficient for the studied system have been experimentally
20 defined and presented. The paper ends up with comparative experimental results for the ICS device and
21 a commercial Flat Plate Thermosiphonic Unit (FPTU).

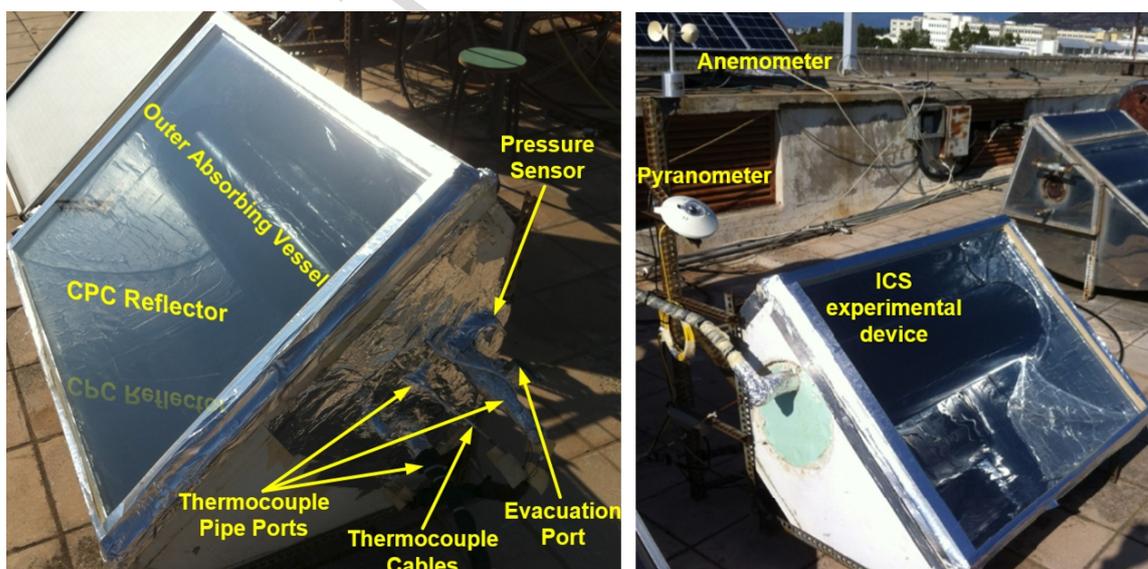
22 **2. Description and operation of the ICS experimental model**

23 The experimental unit design was based on an ICS vessel comprising two concentric cylindrical
24 tanks, placed at horizontally orientation inside a truncated CPC reflector trough. The specific reflector
25 scheme was selected in order to maximize the concentration of solar radiation onto the upper surface of
26 the cylindrical tank, and at the same time for the reduction of the heat losses, by convection. The

1 specified vessel–reflector configuration proposed by Tripanagnostopoulos and Souliotis [8], as
 2 illustrated in Fig.1. The tank consists of two concentric cylindrical vessels: the outer absorbing vessel
 3 and the inner storage vessel. The annulus between these vessels was partially depressurized and
 4 contained a small amount of water (4 kg at 15 °C), acting as PCM which changes phase. The produced
 5 vapor creates a thermal diode transfer mechanism, from the outer absorbing surface to the inner storage
 6 vessel’s surface. Fig. 2 presents the fabricated ICS model at the test field, indicating also some of the
 7 used components for the experimental study. The design – construction details and the geometrical
 8 analysis for the specific configuration have been presented in a previous work by the authors [9].



9
10 **Fig. 1.** Cross section of the model and the concentric cylindrical vessels.



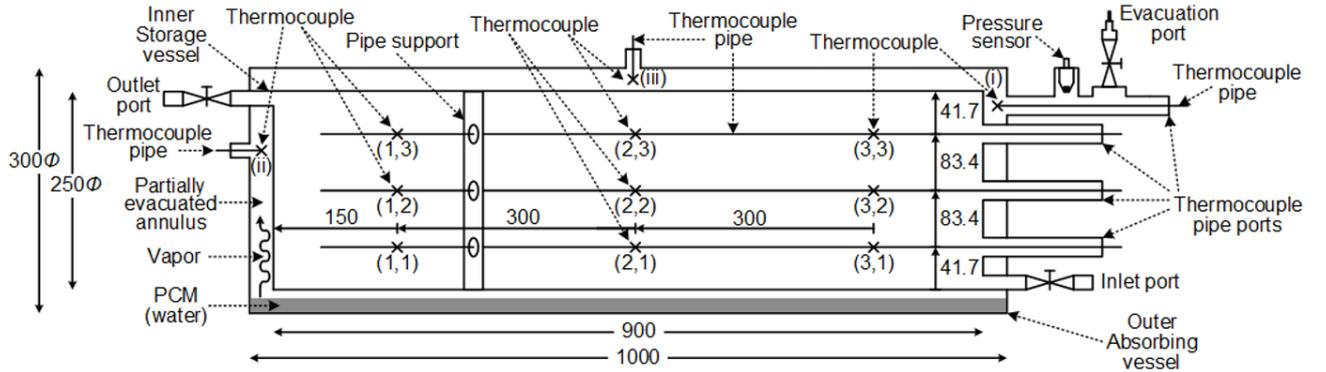
11
12 **Fig. 2.** The ICS experimental device.

1 3. Experimental set up and method of study

2 The ICS system was experimentally studied outdoors for the calculation of the mean daily
 3 efficiency, η_D) and coefficient of thermal losses, U_S . A CMP6 (Kipp & Zonen) pyranometer, with
 4 expected daily uncertainty $< 5 \%$ and an A100LK (Windspeed Limited / Vector Instruments)
 5 anemometer, with accuracy $\pm 0.1 \text{ m}\cdot\text{s}^{-1}$, were employed to measure the solar radiation G (Wm^{-2}) and
 6 the wind speed v_w (ms^{-1}), respectively. For the measurement of the water temperature inside the inner
 7 vessel and the vapor temperature in the annulus between the two concentric cylindrical tanks, T type of
 8 class 1 copper-constantan (Cu-CuNi) thermocouples, with $\pm 0.5 \text{ }^\circ\text{C}$ accuracy, were applied. The total
 9 pressure within the annulus was measured by a PMP 4311 (General Electric Measurement & Control)
 10 pressure transducer sensor, with $\pm 2.325 \%$ accuracy (Figs. 2 and 3). The experimental measurements
 11 were recorded by using a CR1000 data logger (Campbell Scientific Inc.). In the Appendix a detailed
 12 uncertainty analysis is presented, indicating so the accuracies of the involved sensors and logger as the
 13 uncertainties of all used magnitudes.

14 The recording of the stored water temperatures throughout a 24-hour experiment is helpful in
 15 order to verify the temperatures stratification day and night. The stratification is important for the
 16 estimation of the practical benefits of the whole device. During the day, it is related to the non-
 17 uniformity of the absorbed solar radiation [15], as well as the evaporation process of the PCM [9]. The
 18 setup that facilitates the effective measuring of relevant water temperatures is achieved by employing a
 19 grid of thermocouples suitably positioned inside the inner storage vessel, as indicated in Fig. 3
 20 (positions: plane 1 (1,1), (1,2), (1,3), plane 2: (2,1), (2,2), (2,3) and plane 3: (3,1), (3,2), (3,3), for the
 21 measurement of the corresponding temperatures of the stored water: $T_{1,1}$, $T_{1,2}$, $T_{1,3}$, $T_{2,1}$, $T_{2,2}$, $T_{2,3}$, $T_{3,1}$,
 22 $T_{3,2}$, and $T_{3,3}$). Within the annulus, three thermocouples were placed in positions (i), (ii) and (iii) (for
 23 the measurement of the corresponding vapor temperatures: $T_{v,i}$, $T_{v,ii}$ and $T_{v,iii}$). As indicated in Figs. 2
 24 and 3, the pressure sensor PMP 4311 (pressure's measurement) was installed close to the evacuation
 25 port and through the nearest thermocouple pipe port. The recorded parameters: the stored water

- 1 temperature: $T_{1,1}$, $T_{2,1}$, $T_{3,1}$ (Bottom), $T_{1,2}$, $T_{2,2}$, $T_{3,2}$ (Middle) and $T_{1,3}$, $T_{2,3}$, $T_{3,3}$ (Up) (all in °C), the
 2 temperature of the vapor within the annulus: $T_{v,i}$, $T_{v,ii}$, $T_{v,iii}$, the total pressure within annulus P_{total}
 3 (mbar), the ambient temperature T_α (°C), the solar irradiance G (Wm^{-2}) and the wind speed v_w (ms^{-1}).



4 **Fig. 3.** Cross section of the: inner storage vessel – outer absorbing vessel – numbering / positioning
 5 of the thermocouples within the inner storage vessel and the annulus – positioning of the
 6 pressure sensor, inlet, outlet, evacuation and thermocouple pipe ports.
 7

8 The mean temperature of the stored water $T_{w,m}$ (°C) at any time during the operation of the ICS
 9 experimental model, was determined using the equation:

$$10 \quad T_{w,m} = \frac{1}{9} \cdot \left[\sum_{j=1}^3 \left(\sum_{i=1}^3 T_{i,j} \right) \right] \quad (1)$$

11 It varies with seasons and the prevailing weather conditions. The mean water temperatures at the Bottom
 12 $T_{1,m}$ (°C), Middle $T_{2,m}$ (°C) and Up $T_{3,m}$ (°C) of the inner vessel were calculated by the equation:

$$13 \quad T_{j,m} = \frac{1}{3} \cdot \sum_{i=1}^3 T_{i,j}, \text{ for } j = 1, 2 \text{ and } 3 \quad (2)$$

14 In addition, the mean temperature of the vapor within the annulus $T_{v,m}$ (°C), at any time during
 15 the operation of the ICS experimental model, was determined using the equation:

$$16 \quad T_{v,m} = \frac{1}{3} \cdot \sum_{j=i}^{iii} T_{v,j} = \frac{1}{3} \cdot (T_{v,i} + T_{v,ii} + T_{v,iii}) \quad (3)$$

17 The experimental methodology for the calculation of the mean daily efficiency η_D and thermal
 18 losses coefficient during the night-time operation of the system was based on a well-known method [24]
 19 which has been extensively implemented in previous published works [5-10], [15], [25], [27, 28].

20 4. Experimental results and discussion

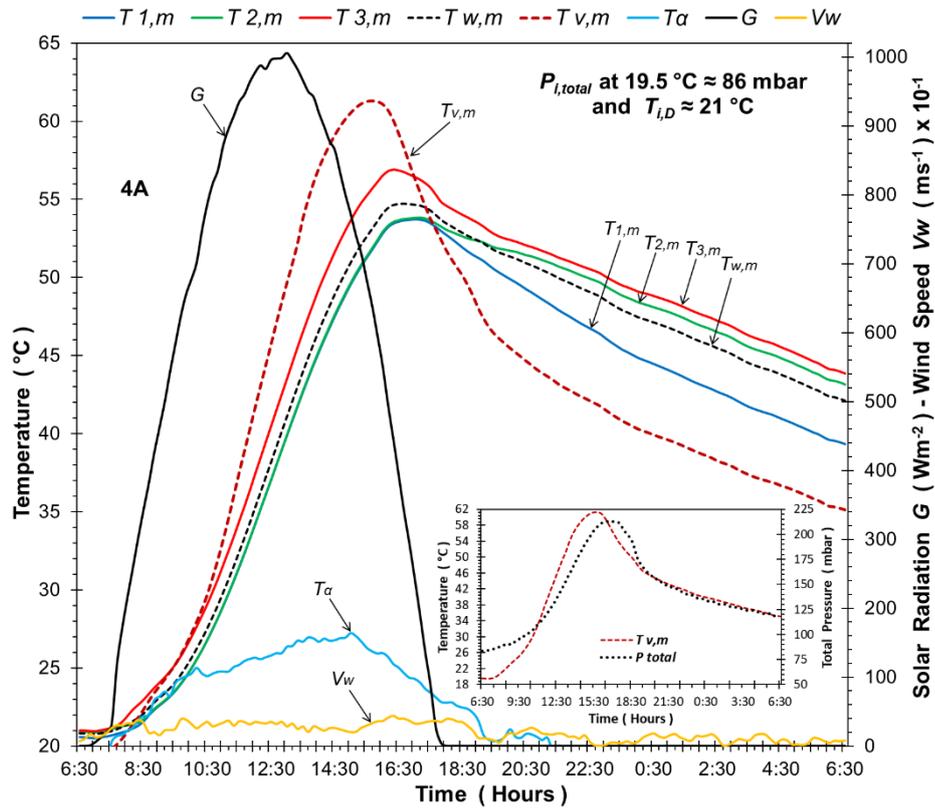
1 The experimental study focused on the investigation of the thermal behavior of the device under
2 partial vacuum and at different initial values of the total pressure within the annulus. The main purpose
3 of the experimental procedure was the determination of the time response for the temperature increase
4 of the stored water during the daily operation and the retention of heat during the nighttime operation.
5 The partial vacuum within the annulus affects both the above-mentioned operations [9], [21, 23]. Thus,
6 the experimental results include the 24-hour variation diagrams of all recorded parameters
7 (temperatures, total pressure, solar irradiance, wind speed), in addition to the calculation and the
8 variation diagrams of the mean daily efficiency and the night thermal losses coefficient versus various
9 operational parameters. All the above parameters are measured under partial vacuum for different initial
10 total pressure values in the annulus, while at the same time the corresponding initial temperatures of the
11 water vapor in the annulus and the stored water are recorded as well. For the experimental procedure
12 eight (8) initial values of the total pressure within the annulus (at the lowest temperature possible values
13 of the water-vapor in the annulus) have been chosen: 86 ± 2 mbar (at 19.5 ± 1 °C), 140 ± 3 mbar (at
14 26.0 ± 1 °C), 245 ± 6 mbar (at 24.0 ± 1 °C), 340 ± 8 mbar (at 25.0 ± 1 °C), 490 ± 11 mbar (at 23.0 ± 1
15 °C), 670 ± 16 mbar (at 22.0 ± 1 °C), 790 ± 18 mbar (at 20.0 ± 1 °C) and 990 ± 23 mbar (at 24 ± 1 °C).
16 In the above values, the uncertainties for the measurement of the temperature (± 1 °C) and the pressure
17 (± 2.325 %) have been considered according to the uncertainty analysis in the Appendix.

18 *4.1 24-hour operation of the ICS experimental model*

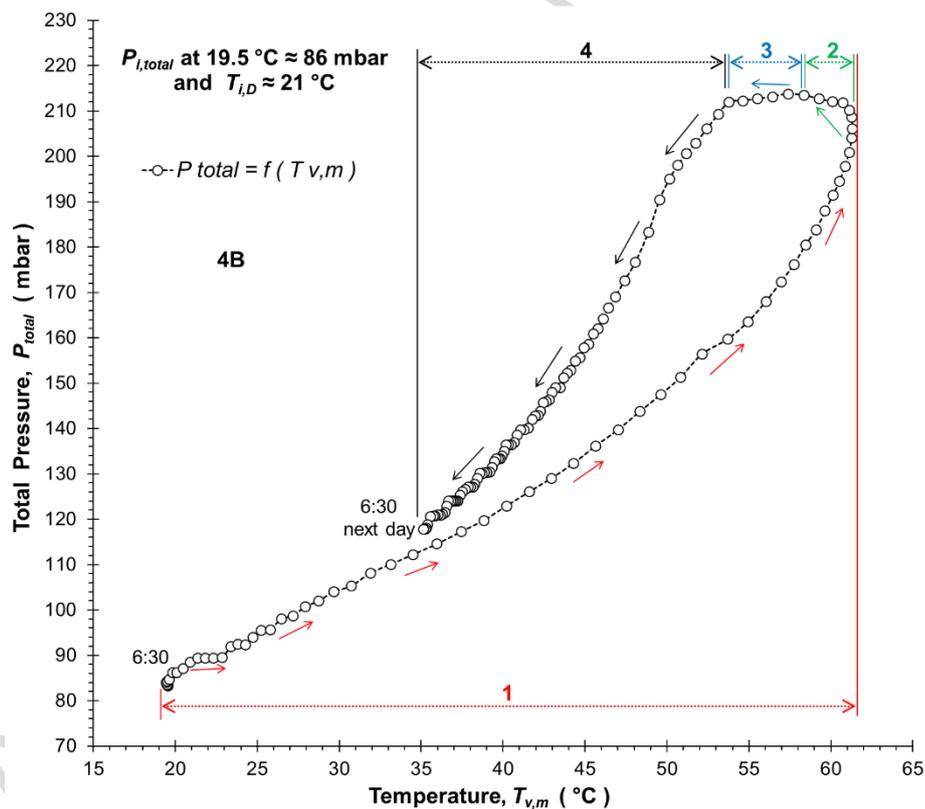
19 The practical benefits from the operation of the tested ICS system are reflected in the temperature
20 variation of the stored water, which is affected by three factors: **i)** the ratio of the total stored water
21 volume per aperture area, **ii)** the distribution of the absorbed solar radiation on the outer vessel [9] and
22 **iii)** the evaporation-condensation process within the annulus between the two concentric cylindrical
23 vessels. The ratio V_T/A_a and the distribution of the absorbed solar radiation on the outer vessel are related
24 to the adopted geometry of the reflectors and its realization in practice [15]. On the other hand, the total
25 pressure and the temperature of the vapor, dominantly specify the evaporation-condensation process in
26 the annulus, as the pressure and temperature are not independent during phase change processes.

1 Figures 4A to 8A demonstrate the 24-hour variation profile diagrams of all the involved
2 parameters, i.e. the temperature of the stored water, the temperature of the vapor, the pressure in the
3 annulus, the ambient temperature, the wind speed and the solar radiation intensity. In each figure there
4 is an inset diagram, depicting the time variation of the pressure and the temperature of the vapor, thus
5 providing further necessary information. The initial temperature of the stored water is the lowest
6 possible and hereupon it has been chosen in order to ascertain the maximum stored water temperature
7 rise, during the daily operation. The inset diagrams designate the heat process, indicating the time delay
8 between the maximum values of temperature and pressure of the vapor within the annulus. Figures 4B
9 to 8B depict the variation of the total pressure versus the temperature of the vapor within the annulus.
10 These diagrams present the pressure's time-dependency on temperature (called P-T-t paths) during the
11 dynamically unsteady vaporization-condensation process.

12 The results show that the observed water temperature stratification is limited. During the day, the
13 non-uniformity of the absorbed solar radiation on the outer vessel seems to have minimal effect on the
14 thermal stratification. During nighttime, the (de)stratification dominantly depends on the thermal
15 convection mechanism within the inner vessel. On the contrary, the temperature rise of the stored water
16 is significant, indicating that the water evaporation in the annulus is crucial. In particular, the lower
17 value of the initial (at 6:30 am) annulus pressure drives to lower temperature difference between the
18 lower (initial at 6:30 am) and maximum temperature of the stored water, during the daily operation of
19 the device (almost at 17:30 pm). For initial pressure of 86 mbar, the temperature difference is near 34
20 °C while for initial pressure of 990 mbar the corresponding temperature difference is over 42 °C. For
21 intermediate values of the initial pressure, the temperature differences vary almost linearly between the
22 above values. The preservation of the heat during the night, strongly depends on the vapor pressure, as
23 seen by the experimental results. Low pressure leads to low convective heat transfer coefficient in
24 annulus, and subsequently to low temperature differences (17:30 pm until 6:30 am next morning).

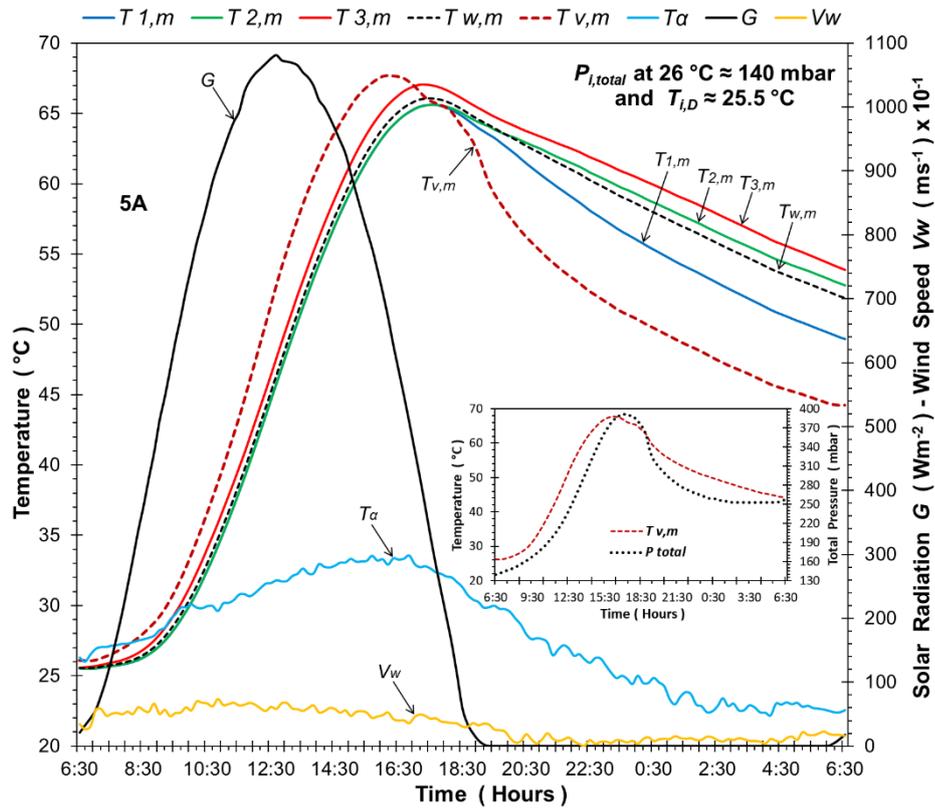


1

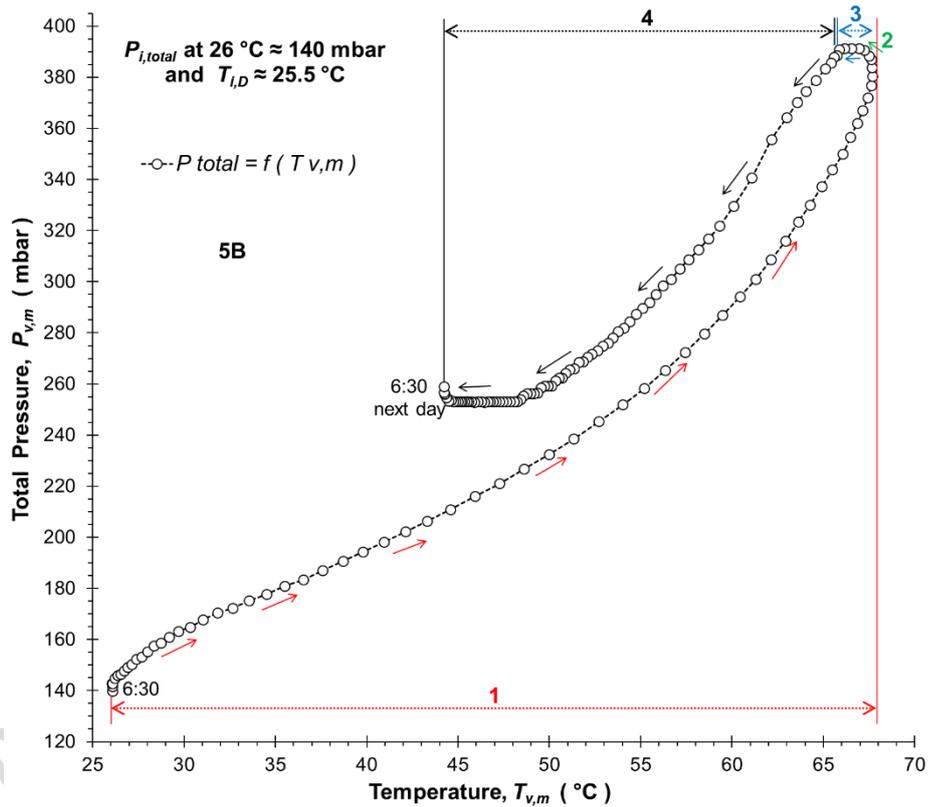


2

3 **Fig. 4.** 4A. Variation in: the mean stored water temperatures within inner storage vessel, the temperature
 4 of the vapor and the total pressure within annulus, the ambient temperature, the solar radiation
 5 intensity and the wind speed velocity, during the 24-hour test period without draw-off (initial
 6 condition, at 06:30: total pressure 86 ± 2 mbar at vapor temperature 19.5 ± 1 °C). 4B. Variation
 7 in the total pressure versus the temperature of the vapor within annulus.

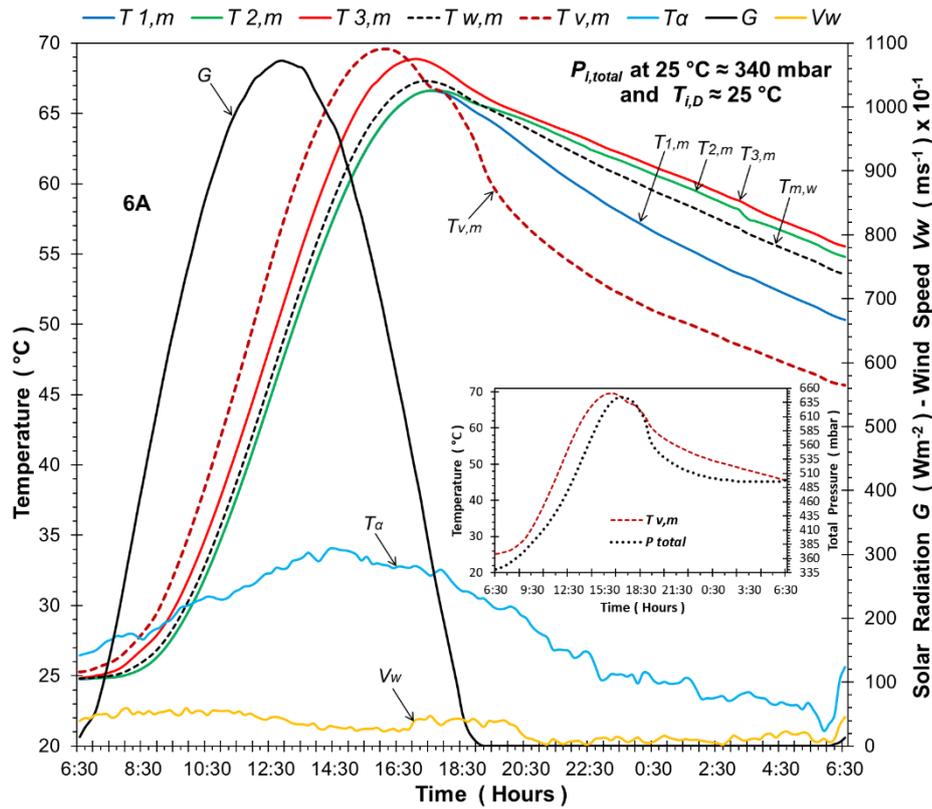


1

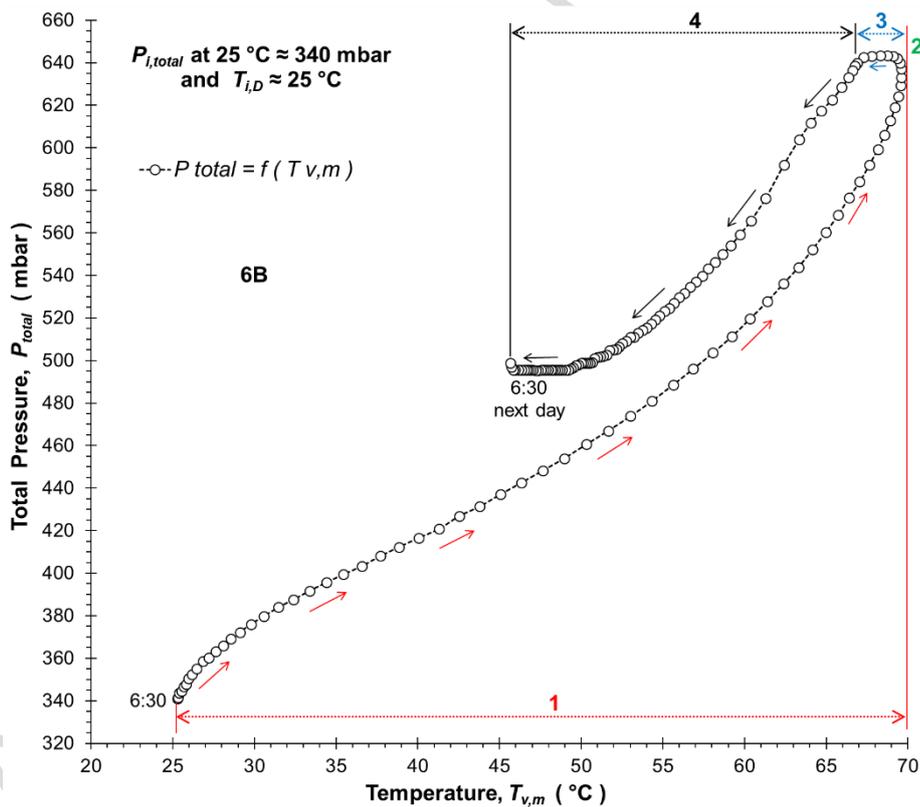


2

3 **Fig. 5.** 5A. Variation in: the mean stored water temperatures within inner storage vessel, the temperature
 4 of the vapor and the total pressure within annulus, the ambient temperature, the solar radiation
 5 intensity and the wind speed velocity, during the 24-hour test period without draw-off (initial
 6 condition, at 06:30: total pressure 140 ± 3 mbar at vapor temperature 26.0 ± 1 °C). 5B. Variation
 7 in the total pressure versus the temperature of the vapor within annulus.

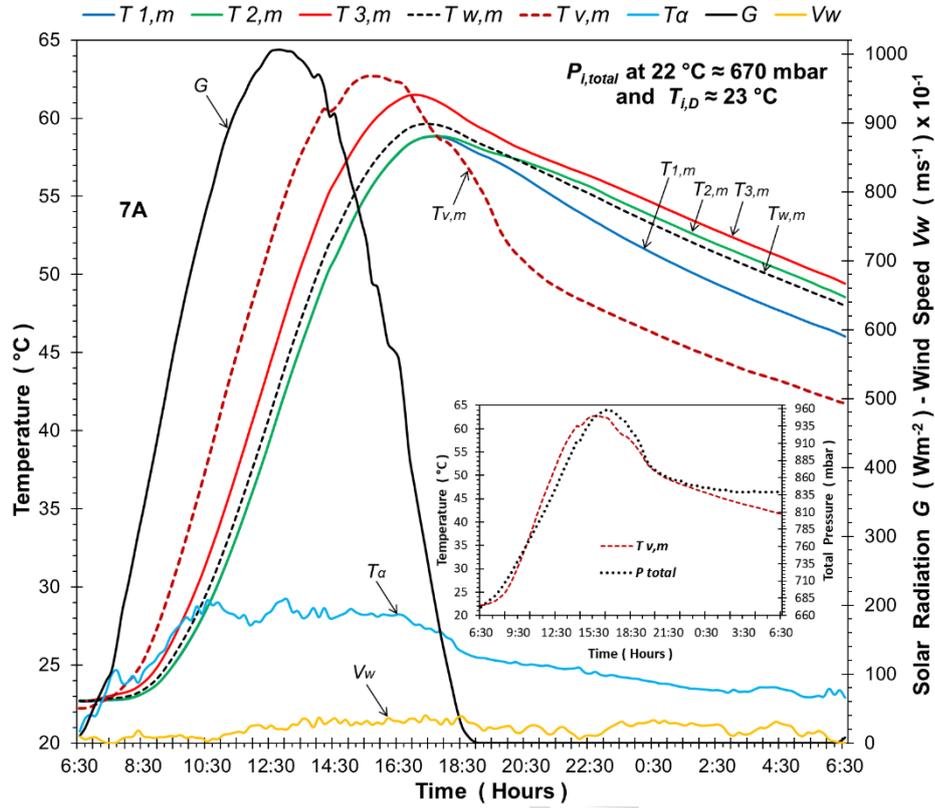


1

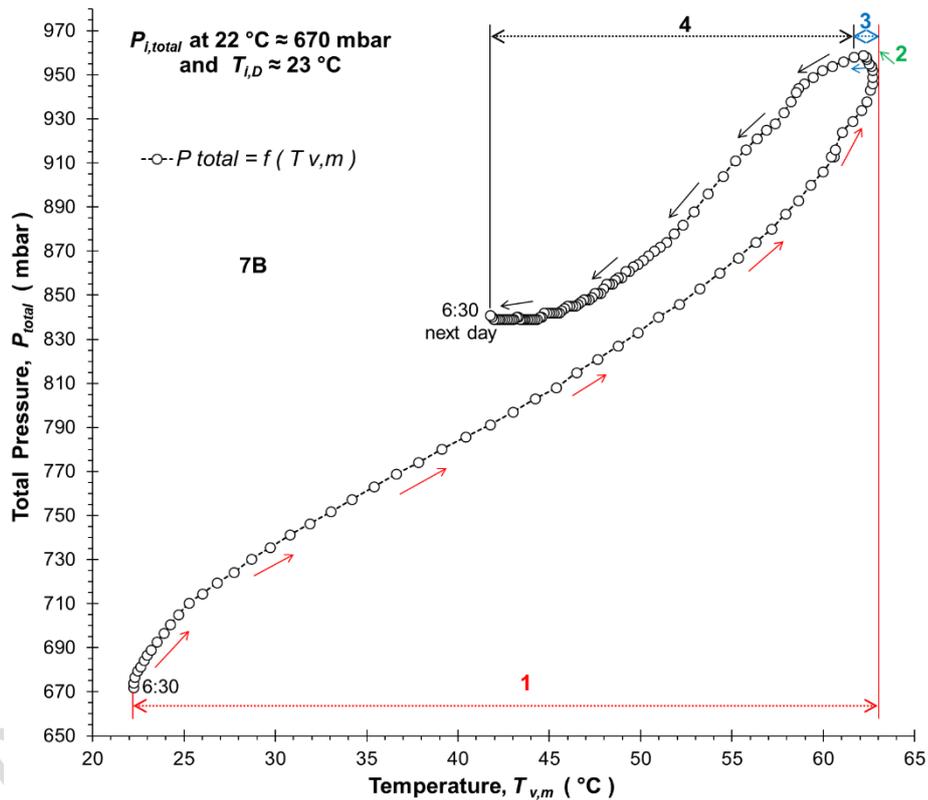


2

3 **Fig. 6.** 6A. Variation in: the mean stored water temperatures within inner storage vessel, the temperature
 4 of the vapor and the total pressure within annulus, the ambient temperature, the solar radiation
 5 intensity and the wind speed velocity, during the 24-hour test period without draw-off (initial
 6 condition, at 06:30: total pressure 340 ± 8 mbar at vapor temperature 25.0 ± 1 °C). 6B. Variation
 7 in the total pressure versus the temperature of the vapor within annulus.

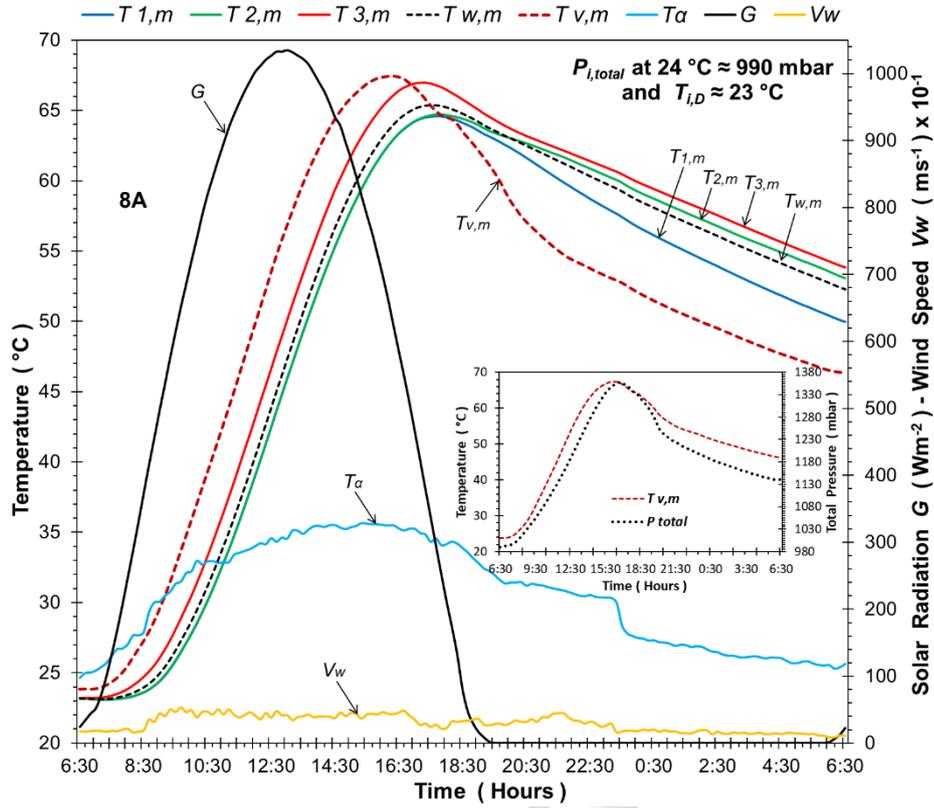


1

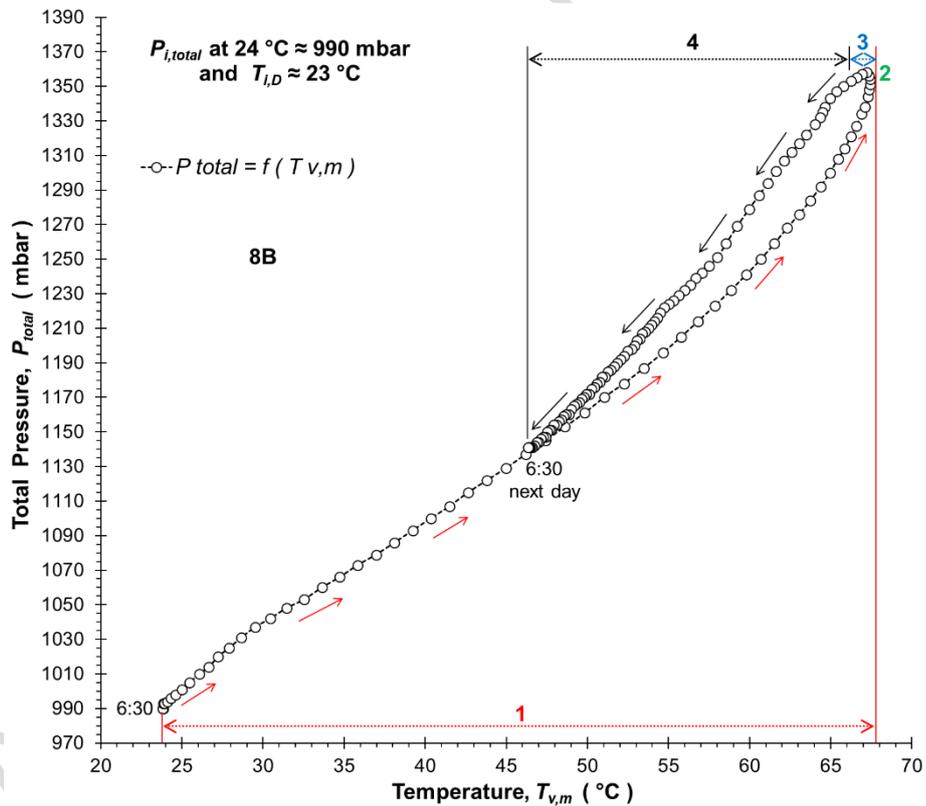


2

3 **Fig. 7.** 7A. Variation in: the mean stored water temperatures within inner storage vessel, the temperature
 4 of the vapor and the total pressure within annulus, the ambient temperature, the solar radiation
 5 intensity and the wind speed velocity, during the 24-hour test period without draw-off (initial
 6 condition, at 06:30: total pressure 670 ± 16 mbar at vapor temperature 22.0 ± 1 °C). 7B. Variation
 7 in the total pressure versus the temperature of the vapor within annulus.



1



2

3 **Fig. 8.** 8A. Variation in: the mean stored water temperatures within inner storage vessel, the temperature
 4 of the vapor and the total pressure within annulus, the ambient temperature, the solar radiation
 5 intensity and the wind speed velocity, during the 24-hour test period without draw-off (initial
 6 condition, at 06:30: total pressure 990 ± 23 mbar at vapor temperature 24.0 ± 1 °C). 8B. Variation
 7 in the total pressure versus the temperature of the vapor within annulus.

1 Regarding the time response of the temperature rise, the results indicate that the initial pressure
2 within the annulus is not crucial but it depends mainly on the incoming solar radiation. In particular, the
3 time interval between lower and higher values of the stored water temperature, in all initial conditions,
4 is almost the same (~10 hours, at 17:30 pm). There is a constant time delay between the maximum
5 values of the solar radiation intensity and the stored water temperature, almost 5 hours (12:30 pm up to
6 17:30 pm). Additionally, there is an almost 80 minutes constant time delay between the maximum
7 values of the vapor temperature and the stored water temperature, which depends on the initial pressure.

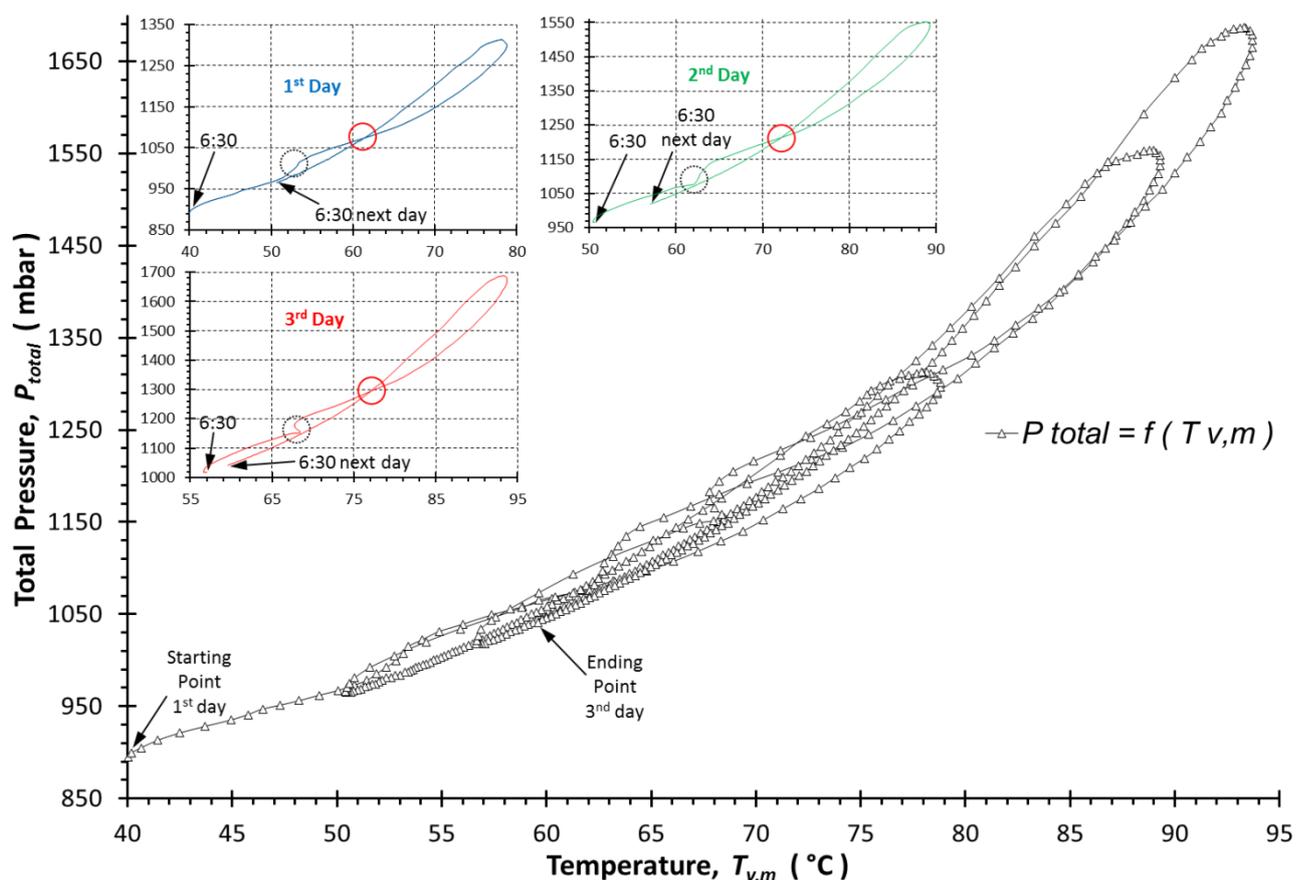
8 From the inset diagrams of Figs 4A to 8A, where the time variation of the vapor temperature and
9 pressure are shown, it is evident that there is also a time delay between their maximum values. The
10 results indicate that the time delay decreases as the initial pressure increases (70 minutes in 86 mbar and
11 20 minutes in 990 mbar). For the initial pressure between the above values, the time delays vary almost
12 linearly between 70 and 20 minutes. There are also differentiations regarding the rise of the total
13 pressure and the temperature of the vapor during the daily operation of the device. As the initial pressure
14 increases, the temperature rises are immediate. This fact determines the actual role of the initial pressure
15 within the annulus during the heating process.

16 The pressure-temperature variation diagrams of figures 4B to 8B show the time dependency on
17 the heat diode mechanism within the annulus. These diagrams are supplemental to the insets of figures
18 4A to 8A, and indicate the heating and cooling processes for different initial values of the pressure
19 within the annulus. According to these results, the 24-hour operation of the system is separated into four
20 consecutive processes which are indicated in each of the figures with four numbers: 1, 2, 3 and 4.
21 Process 1 begins at 6:30 in the morning and expires when the temperature of vapor reaches its highest
22 value. During the process 1, both pressure and temperature increase almost linearly up to the time when
23 the solar irradiance is stabilized at its maximum value. Process 2 begins at the end of process 1 and ends
24 when the pressure receives its maximum value, when process 3 begins and ends where the pressure
25 begins to decrease. During this process, the pressure is almost constant and only the temperature of the
26 vapor decreases. Finally, process 4 begins at the end of process 3 and ends at 6:30 in the next morning.

1 During the last process, both pressure and temperature decrease.

2 According to the extracted results, processes 2 and 3 almost disappear as the initial pressure within
3 the annulus increases. In particular, process 2 vanishes at initial pressure of 140 mbar, while process 3
4 almost dies out at initial pressure of 670 mbar. At the pressure of 990 mbar, the whole procedure was
5 practically restricted only to processes 1 and 4. It is worth mentioning the cases when the initial pressure
6 is at 140, 340 and 670 mbar, in which during the process 4 (near 6:30 in the next morning) the pressure
7 is almost constant. At this time range, the results indicate that the system has reached a steady state
8 condition.

9 The P-T-t paths, for the initial water temperatures above the water mains temperature, seem to be
10 more complicated, as presented in Figure 9, which shows the variation of the pressure versus the vapor
11 temperature for the operation of the device for three continuous days without any storage water
12 temperature reset (draw off). The initial temperature is 40 ± 1 °C and the pressure is almost 870 ± 20
13 mbar. The figure also includes three inset diagrams where the variation of the parameters for each
14 consequent day of are depicted. The results show that the heating and cooling procedures are practically
15 developed by the processes 1, 2 and 4, as discussed in the previous paragraph. However, there are two
16 additional subjects for discussion. The first refers to the process of heating in which discontinuity
17 regions during process 1 are observed. These discontinuity regions are indicated by black dashed circles
18 in inset diagrams. The discontinuity region is bigger at the third consequent day, compared with the
19 corresponding discontinuities at the first and the second day, when the values of the pressure and the
20 temperature are high. The second subject deals with the intersection points during process 1 and 4 in
21 which they appear for everyone of the 3 days operation of the device and are marked by red solid circles
22 in the inset diagrams. These points are defined by the initial conditions of vapor temperature and
23 pressure and they are located at a constant distance from the maximum temperature during the 24-hour
24 process (almost 16.0 ± 1 °C).

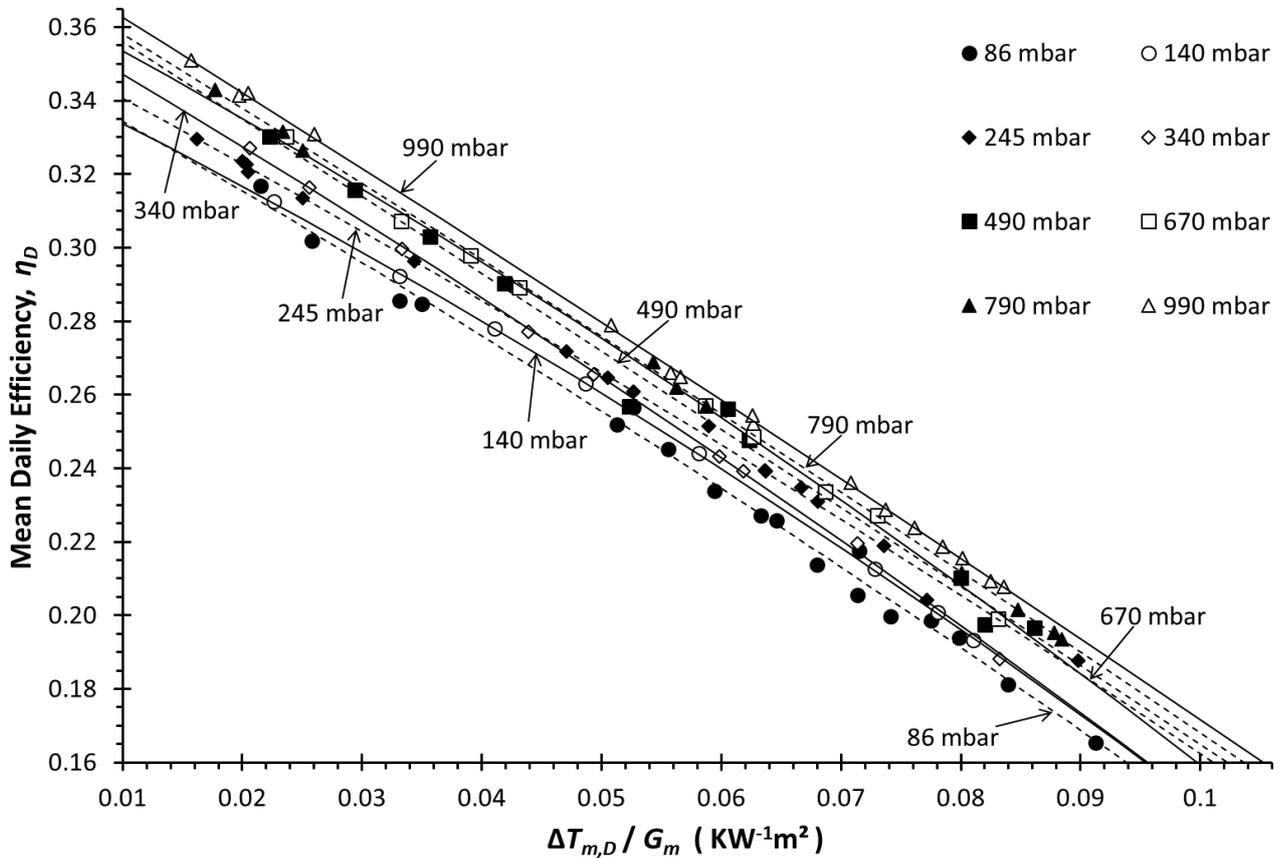


1
2 **Fig. 9.** Variation in the total pressure versus the temperature of the vapor within annulus during 3
3 continuous days of ICS operation without draw-off (initial conditions at 6:30 in the first day:
4 total pressure 870 ± 20 mbar at vapor temperature 40 ± 1 °C). For the clarification of the
5 thermal performance, inset diagrams depict the variation in the total pressure versus the
6 temperature of the vapor within annulus for each of the three days.

7 From the above experimental results, it can be deduced that the thermal behavior of the device
8 depends vitally on the initial pressure of the vapor inside the annulus. Concerning the initial conditions
9 at which the device presents the most optimal behavior, it can be concluded that the pressure of the
10 vapor in the range between 670 ± 16 and 790 ± 18 mbar constitutes the more suitable choice. In that
11 range of the initial pressure, the temperature increase, during the daily operation of the device, varies
12 between 37 and 39 °C and at the same time, during nighttime operation, the temperature decrease varies
13 between 11 to 13 °C, which are generally accepted.

1 4.2 Variation of the mean daily efficiency and thermal loss coefficient

2 Both the mean daily efficiency and thermal losses coefficient were determined on the basis of
 3 consecutive 24-hour operations in which the initial values of the stored water temperatures and partial
 4 annulus vacuum were different. Therefore, eight different configurations were studied and thus the total
 5 number of the 24-hours experiments arises to almost 100. These experiments were accomplished during
 6 the summer period for two consecutive years, and the experimental parameters of the weather conditions
 7 (i.e mean solar radiation values, ambient temperature and wind speed) were chosen to be comparable
 8 and favorable.



9
 10 **Fig. 10.** Variation of the mean daily efficiency η_D of the ICS system, at several initial pressure of
 11 partial vacuum, versus $\Delta T_{m,D} / G_m$.

12 In Fig. 10 the variation in the mean daily efficiency (η_D) for the ICS system versus the parameter

13 $\Delta T_{m,D} / G_m$ ($\Delta T_{m,D} / G_m = [(T_{i,D} + T_{f,D}) / 2 - T_{a,D}] / G_m$ in $\text{K} \cdot \text{W}^{-1} \cdot \text{m}^2$), are shown for all mentioned cases.

14 The uncertainty for the calculation of the mean daily efficiency η_D has been presented in the Appendix.

1 For the daily operation of the device, $T_{i,D}$ ($^{\circ}\text{C}$) is the initial temperature of the stored water 6 hours
2 before solar noon (at $\sim 6:30$ in the morning), $T_{f,D}$ ($^{\circ}\text{C}$) is the final temperature of the stored water 6
3 hours after solar noon (at $\sim 18:30$ in the afternoon) and $T_{a,D}$ ($^{\circ}\text{C}$) is the mean ambient temperature during
4 the daily operation, according ISO 9459-2 [24]. For $\Delta T_{m,D} / G_m$ values ranging between 0.0150 ± 0.0002
5 and $0.045 \pm 0.002 \text{ K}\cdot\text{W}^{-1}\cdot\text{m}^2$, the ICS system behaves better as the initial pressure increases, while the
6 best performance is observed for initial pressure of 990 mbar. This is in agreement with the results
7 presented in the previous section, in which the initial temperature of the stored water varies at low
8 values. In the range between 0.045 ± 0.002 and $0.075 \pm 0.004 \text{ K}\cdot\text{W}^{-1}\cdot\text{m}^2$ the thermal performance of the
9 device indicates that for initial pressure of 990 mbar the device still behaves better than the rest
10 configurations, but also for 490, 670 and 790 mbar the mean daily efficiencies of the device are quite
11 similar. It is important to be noticed that this operational range can be considered as typical for the ICS
12 system, as the mean stored water temperature during the daily operation varies between 35 and 60 $^{\circ}\text{C}$.
13 For the operational range between 0.075 ± 0.004 and $0.090 \pm 0.005 \text{ K}\cdot\text{W}^{-1}\cdot\text{m}^2$ the values of the mean
14 daily efficiencies, at the initial pressure of 990 mbar, are higher than the corresponding values in other
15 cases. However, the mean daily efficiencies at the initial pressures of 245, 490, 670 and 790 mbar are
16 almost equal, approaching the corresponding value for the initial pressure of 990 mbar. In Table 1
17 equations of the mean daily efficiency η_D and their regression coefficient R , for all studied cases of the
18 tested device, against $\Delta T_{m,D} / G_m$, are also depicted.

1 **Table 1**

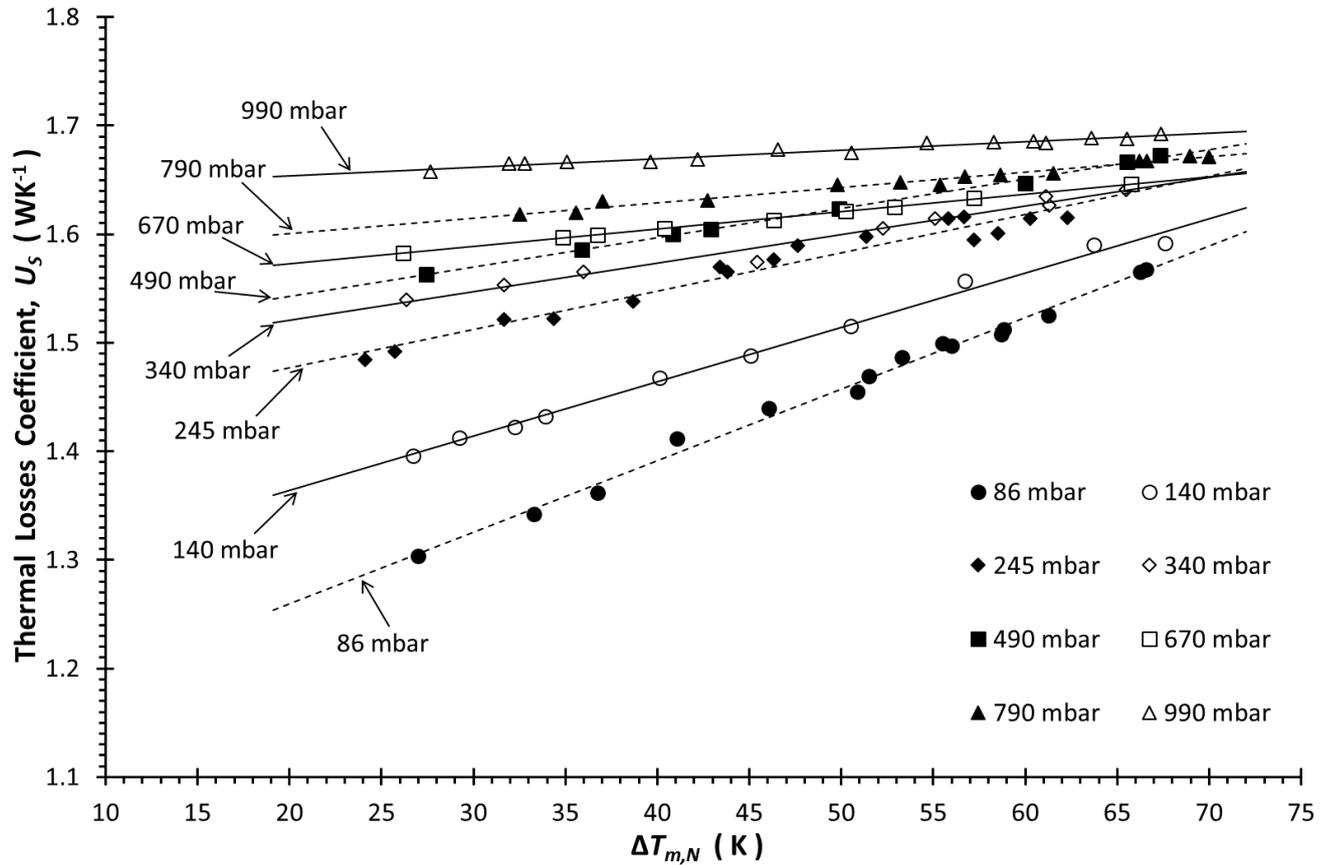
2 Correlations for mean daily efficiency η_D , including the Regression Coefficient R , for initial values of
 3 total partial vacuum pressure (at specific temperatures), versus $\Delta T_{m,D} / G_m$.

Initial total pressure of partial vacuum (at temperature)	Mean daily efficiency, η_D	Regression Coefficient, R
86 ± 2 mbar (at $19.5 \pm 1^\circ\text{C}$)	$0.353 - 1.823 \cdot (\Delta T_{m,D} / G_m) - 2.476 \cdot (\Delta T_{m,D} / G_m)^2$	0.9964
140 ± 3 mbar (at $26.0 \pm 1^\circ\text{C}$)	$0.350 - 1.574 \cdot (\Delta T_{m,D} / G_m) - 4.356 \cdot (\Delta T_{m,D} / G_m)^2$	0.9999
245 ± 6 mbar (at $24.0 \pm 1^\circ\text{C}$)	$0.358 - 1.716 \cdot (\Delta T_{m,D} / G_m) - 2.389 \cdot (\Delta T_{m,D} / G_m)^2$	0.9988
340 ± 8 mbar (at $25.0 \pm 1^\circ\text{C}$)	$0.367 - 1.894 \cdot (\Delta T_{m,D} / G_m) - 2.780 \cdot (\Delta T_{m,D} / G_m)^2$	0.9997
490 ± 11 mbar (at $23.0 \pm 1^\circ\text{C}$)	$0.377 - 2.091 \cdot (\Delta T_{m,D} / G_m) - 0.300 \cdot (\Delta T_{m,D} / G_m)^2$	0.9954
670 ± 16 mbar (at $22.0 \pm 1^\circ\text{C}$)	$0.371 - 1.720 \cdot (\Delta T_{m,D} / G_m) - 3.981 \cdot (\Delta T_{m,D} / G_m)^2$	0.9993
790 ± 18 mbar (at $20.0 \pm 1^\circ\text{C}$)	$0.378 - 1.986 \cdot (\Delta T_{m,D} / G_m) - 1.143 \cdot (\Delta T_{m,D} / G_m)^2$	0.9999
990 ± 23 mbar (at $24.0 \pm 1^\circ\text{C}$)	$0.383 - 2.019 \cdot (\Delta T_{m,D} / G_m) - 0.939 \cdot (\Delta T_{m,D} / G_m)^2$	0.9999

4 Figure 11 presents the variation of the thermal loss coefficient U_S ($\text{W} \cdot \text{K}^{-1}$) during nighttime
 5 operation of the ICS system, versus the parameter $\Delta T_{m,N}$ ($\Delta T_{m,N} = T_{i,N} - T_{a,N}$) (K). During nighttime
 6 operation, $T_{i,N}$ ($^\circ\text{C}$) is the initial temperature of the stored water 6 hours after solar noon (at $\sim 18:30$ in
 7 the afternoon) and $T_{a,N}$ ($^\circ\text{C}$) is the mean ambient temperature during the nighttime operation. The
 8 uncertainty for the calculation of the thermal loss coefficient U_S has been presented in the Appendix.

9 The thermal loss coefficient ranged from 1.30 ± 0.02 to 1.70 ± 0.02 $\text{W} \cdot \text{K}^{-1}$ for all configurations.
 10 The ICS experimental system at 990 mbar exhibits the highest thermal loss coefficient values ($\sim 1.67 \pm$
 11 0.02 $\text{W} \cdot \text{K}^{-1}$), in the total range of $\Delta T_{m,N}$, compared to the corresponding values in all other cases. In
 12 the range of $\Delta T_{m,N}$ between 25 and 45 $^\circ\text{C}$ for initial pressure of 86 mbar, the device presents the lowest
 13 thermal loss coefficient values (1.30 ± 0.02 to 1.54 ± 0.02 $\text{W} \cdot \text{K}^{-1}$). For intermediate values of the initial
 14 pressure and for the same operational range, the corresponding thermal losses vary between a minimum
 15 (at 86 mbar) and a maximum (at 990 mbar). In the range between 45 and 70 $^\circ\text{C}$, the difference in the

1 variation of the thermal losses coefficient for all cases are almost the same as in the range between 25
 2 and 45 °C, except for the case at the initial pressure of 670 mbar, at which the thermal losses coefficient
 3 is almost constant (varies between 1.60 ± 0.02 to $1.62 \pm 0.02 \text{ W} \cdot \text{K}^{-1}$). It is worth mentioning the variation
 4 of the thermal losses coefficient for the cases 86 and 140 mbar at which the increase rate of the thermal
 5 losses is significantly higher than the corresponding rate for the other configurations. According to the
 6 above results, the selection of initial pressure of 670 mbar seems to be the most suitable choice.



7
 8 **Fig. 11.** Variation of the thermal loss coefficient U_s for the ICS system, at several initial pressure of
 9 partial vacuum, versus $\Delta T_{m,N}$.

10 It is noteworthy that the present results give strong evidence that the thermal loss coefficients for
 11 the ICS model under consideration, for all cases, exhibit lower values as compared to existing literature
 12 [1, 2]. This can be considered as a very significant improvement for this type of solar devices. Table 2
 13 contains the fitting equations of U_s for all studied cases, including their regression coefficient R ,
 14 against $\Delta T_{m,N}$.

1 **Table 2**

2 Correlations for thermal loss coefficient U_S , including the Regression Coefficient R , for initial values
 3 of total partial vacuum pressure (at specific temperatures), versus $\Delta T_{m,N}$.

Initial total pressure of partial vacuum (at temperature)	Thermal loss coefficient, U_S (WK^{-1})	Regression Coefficient, R
86 ± 2 mbar (at $19.5 \pm 1^\circ C$)	$1.128 + 0.0066 \cdot \Delta T_{m,N}$	0.9966
140 ± 3 mbar (at $26.0 \pm 1^\circ C$)	$1.264 + 0.0050 \cdot \Delta T_{m,N}$	0.9971
245 ± 6 mbar (at $24.0 \pm 1^\circ C$)	$1.407 + 0.0035 \cdot \Delta T_{m,N}$	0.9775
340 ± 8 mbar (at $25.0 \pm 1^\circ C$)	$1.468 + 0.0026 \cdot \Delta T_{m,N}$	0.9897
490 ± 11 mbar (at $23.0 \pm 1^\circ C$)	$1.489 + 0.0027 \cdot \Delta T_{m,N}$	0.9990
670 ± 16 mbar (at $22.0 \pm 1^\circ C$)	$1.541 + 0.0016 \cdot \Delta T_{m,N}$	0.9994
790 ± 18 mbar (at $20.0 \pm 1^\circ C$)	$1.573 + 0.0014 \cdot \Delta T_{m,N}$	0.9879
990 ± 23 mbar (at $24.0 \pm 1^\circ C$)	$1.638 + 0.0008 \cdot \Delta T_{m,N}$	0.9805

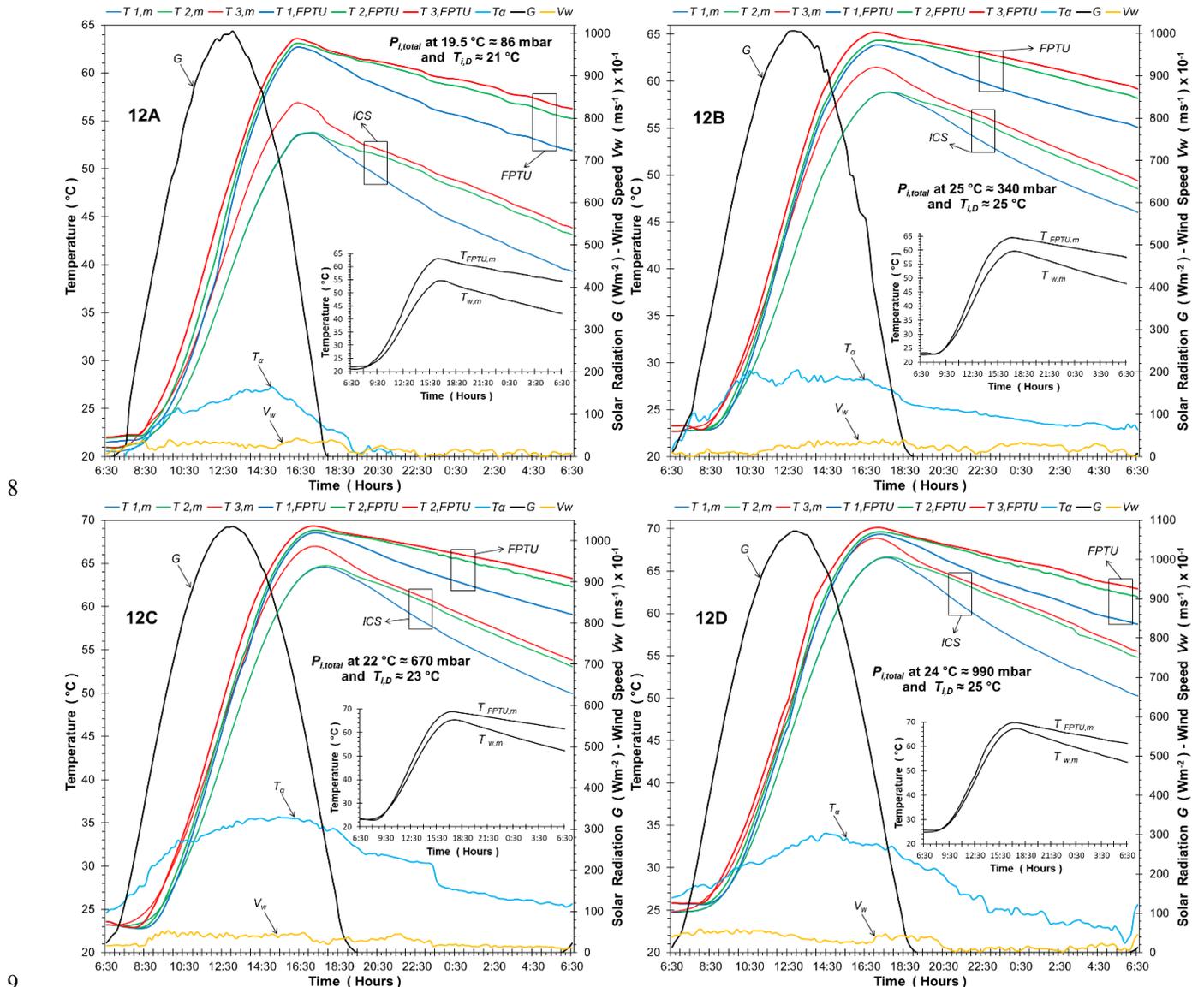
4 As mentioned above, the thermal behavior of the system defined substantially by the values of
 5 mean daily efficiency, in the typical operational range between 0.075 ± 0.004 and 0.090 ± 0.005 $K \cdot W^{-1} \cdot m^2$
 6 and the values of thermal loss coefficient in the typical operational range between 45 and 70 $^\circ C$.
 7 Conclusively, taking into account the overall performance, according to the 24-hour operation and the
 8 variations of the mean daily efficiency and the night thermal losses coefficient, the ICS model with
 9 initial pressure of 670 ± 16 mbar behaves at an optimal level. However, further improvement with regard
 10 to thermal performance could be achieved by optimizing or accelerating the thermal diode transfer
 11 mechanism.

1 4.3 Comparative experimental results between the ICS model and FPTU system

2 Since FPTU and ICS solar water heaters belong to the category of solar devices that cover needs
 3 of domestic applications, a recent comparative study has been accomplished [29]. In the present work,
 4 for the determination of the practical benefits of the studied ICS model, an additional experimental
 5 procedure has been realized. In particular, its thermal behavior of was compared to that of a commercial
 6 FPTU. The flat heat receiver was covered with a selective coating (with $\alpha_r = 0.92$ and $\varepsilon_r = 0.15$),
 7 whilst the stored water volume and the aperture area were $V_T = 96.00 \pm 0.01$ l and $A_\alpha = 1.77 \pm 0.01$
 8 m^2 , respectively. Thus the ratio of stored water volume to the aperture area is $V_T / A_\alpha = 54.360 \pm 0.014$
 9 lm^{-2} . It should be noted that that this ratio is almost the same as in the ICS experimental model (53.420
 10 $\pm 0.014 \text{ lm}^{-2}$) [9], thus validating the comparison of the temperature/time behavior amongst the two
 11 systems. The uncertainty for the measurement of the ratio V_T / A_α has been calculated in the Appendix.
 12 The temperature stratification was measured by three (3) thermocouples ($T_{1 \text{ FPTU}}$, $T_{2 \text{ FPTU}}$ and $T_{3 \text{ FPTU}}$)
 13 which were suitably placed in lower, middle and upper part of the tank.

14 In Figure 12 four comparative 24-hour variation diagrams in the stored water temperatures for the
 15 ICS and the FPTU systems without draw-off are presented. The experimental process started for both
 16 systems with initial mean water temperatures around $23 \text{ }^\circ\text{C}$ while the ICS was set up at different initial
 17 pressure inside the annulus each time. For each diagram there is an inset indicating the variation of the
 18 mean temperature of the stored water for both systems. In Fig. 12A the initial pressure inside the annulus
 19 was 86 mbar, while in Figs 12B, 12C and 12D the corresponding initial pressures were 340, 670 and
 20 990 mbar. As indicated in all figures, the water temperature of the FPTU system varies in the same
 21 range (between 23 and almost $70 \text{ }^\circ\text{C}$). On the contrary, the corresponding temperature variation of the
 22 ICS system depends on the initial pressure inside the annulus. In addition, the preservation of heat for
 23 the FPTU system is almost constant, but for the ICS system this depends also on the initial pressure
 24 inside the annulus. From the experimental results, it is obvious that during the daily operation, the ICS
 25 system with initial pressure of 990 mbar has a better performance compared to the remaining cases and
 26 it approaches the performance of the FPTU system. Conversely, as the initial pressure drops down, the

1 preservation of the heat increases. Inset diagrams in all cases show the same results. The ICS device (at
 2 initial pressures 670 and 990 mbar) operates slightly better than the FPTU in the early morning when
 3 the initial water temperature of both systems is around 23 °C. A direct comparison of the two systems
 4 indicates that the temperature stratification of ICS system, in all cases, is enhanced compared to the
 5 FPTU one. Taking into consideration so the temperature increase, during the daily operation of the ICS
 6 system, as the heat preservation, during the nighttime operation of it, the initial pressure around 670
 7 mbar represents the best choice among the rest cases.



8
 9 **Fig. 12.** Comparative 24 hour variation diagrams in the stored water temperatures of ICS and FPTU
 10 systems without draw-off at initial (06:30) mean water stored temperature, for both devices, in
 11 the range between 21 – 25 ± 1 °C. Variation diagrams in ambient temperature, solar radiation
 12 intensity and wind speed velocity. Inner diagrams show the 24-hour variation diagrams in the
 13 mean stored water temperatures of ICS and FPTU.
 14

1 **5. Conclusions**

2 A systematic and extensive experimental study of an ICS system, which comprises of two
3 horizontal concentric cylinders combined with a stationary truncated asymmetric CPC reflector trough,
4 has been performed in a time frame spanning more than two years. The novelty of the suggested working
5 principle is that, inside the annulus, a thermal diode mechanism was introduced with the aid of a PCM
6 material, while the area was partially depressurized. Our results indicate that despite of the non-
7 uniformity of the absorbed radiation, the temperature stratification of stored water is marginal. The
8 operation of the solar device is determined by the effectiveness of the thermal diode mechanism, with
9 the PCM vapor pressure being the crucial parameter. The results showed that the ICS system operates
10 at an optimal level when the initial pressure within annulus in the pressure located closed to 670 mbar.
11 The optimal thermal behavior of the device has to do mainly with the significant temperature increase
12 (around 37 °C) during its daily operation in combination with the low values of night thermal loss
13 coefficient, in which is varied in the range between 1.60 to 1.62 W·K⁻¹. These two findings can be
14 considered significant improvements for the ICS systems. Additionally, the comparison of the thermal
15 behavior between the ICS device and a commercial FPTU reveals an improved thermal performance of
16 the ICS system, offering an alternative solution for water heating. The present study might serve as a
17 model for future experiments with different PCM materials and variable design/geometry parameters.

18 **Acknowledgments**

19 The authors gratefully acknowledge COST Action TU1205 “Building Integrated Solar Thermal
20 Systems” for providing excellent research networking. Dr. Souliotis also would like to acknowledge the
21 Greek PPC (Public Power Corporation S.A. - Hellas) for the financial support.

1 **References**

- 2 [1] Singh R, Lazarus IJ, Souliotis M. Recent developments in integrated collector storage (ICS) solar
3 water heaters: A review. *Renewable & Sustainable Energy Reviews* 2016; 54: 270–298.
- 4 [2] Souliotis M, Singh R, Papaefthimiou S, Lazarus IJ, Andriosopoulos K.. Integrated Collector
5 Storage Solar Water Heaters: Survey and Recent developments. *Energy Systems* 2016; 7: 49–72.
- 6 [3] Smyth M, McGarrigle P, Eames PC, Norton B. Experimental Comparison of Alternative
7 Convection Suppression Arrangements for Concentrating Integral Collector Storage Solar Water
8 Heaters. *Solar Energy* 2005; 78: 223-233.
- 9 [4] Henderson D, Junaidi H, Muneer T, Grassie T, Currie J. Experimental and CFD investigation of
10 an ICSSWH at various inclinations. *Renewable and Sustainable Energy Reviews* 2007; 11: 1087-
11 1116.
- 12 [5] Tripanagnostopoulos Y, Souliotis M. ICS solar systems with two water tanks. *Renewable Energy*
13 2006; 31: 1698-1717.
- 14 [6] Tripanagnostopoulos Y, Souliotis M, Nousia Th. CPC Type Integrated Collector Storage Systems.
15 *Solar Energy* 2002; 72: 327-350.
- 16 [7] Souliotis M, Tripanagnostopoulos Y. Experimental Study of CPC type ICS solar systems. *Solar*
17 *Energy* 2004; 76: 389-408.
- 18 [8] Tripanagnostopoulos Y, Souliotis M. Integrated Collector Storage Solar Systems with
19 Asymmetric CPC Reflectors. *Renewable Energy* 2004; 29: 223-248.
- 20 [9] Souliotis M, Quinlan P, Tripanagnostopoulos Y, Smyth M, Zacharopoulos A, Ramirez M,
21 Yianoulis P. Heat retaining integrated collector storage system with asymmetric CPC reflector.
22 *Solar Energy* 2011; 85: 2474-2487.
- 23 [10] Souliotis M, Kalogirou S, Tripanagnostopoulos Y. Modelling of an ICS solar water heater using
24 Artificial Neural Networks and TRNSYS. *Renewable Energy* 2009; 34: 1333-1339.

- 1 [11] Helal O, Chaouachi S, Gabsi S. Design and thermal performance of an ICS solar water heater
2 based on three parabolic sections. *Renewable Energy* 2011; 85: 2421-2432.
- 3 [12] Gertzos K, Caouris Y. Experimental and computational study of the developed flow field in a flat
4 plate integrated collector storage (ICS) solar device with recirculation. *Experimental Thermal and*
5 *Fluid Science* 2007; 31: 1133-1145.
- 6 [13] Garnier C, Currie J, Muneer T. Integrated collector storage solar water heater: Temperature
7 stratification. *Applied Energy* 2009; 86: 1465-1469.
- 8 [14] Kumar R, Rosen M. Integrated collector - storage solar water heater with extended storage unit.
9 *Applied Thermal Engineering* 2011; 31: 348-354.
- 10 [15] Souliotis M, Tripanagnostopoulos Y. Study of the distribution of the absorbed solar radiation on
11 the performance of a CPC type ICS water heater. *Renewable Energy* 2008; 33: 846-858.
- 12 [16] Hailiot D, Py X, Goetz V, Benabdelkarim M. Storage composites for the optimisation of solar
13 water heating systems. *Chemical Engineering Research and Design* 2008; 86: 612-617.
- 14 [17] Eams PC, Griffiths PW. Thermal behaviour of integrated solar collector/storage unit with 65°C
15 phase change material. *Energy Conversion and Management* 2006; 47: 3611-3618.
- 16 [18] Griffiths PW, Huang MJ, Smyth M. Improving the heat retention of integrated collector/storage
17 solar water heaters using Phase Change Materials Slurries. *International Journal of Ambient*
18 *Energy* 2007; 28: 89-98.
- 19 [19] Al-Hinti I, Al-Ghandoor A, Maaly A, Abu Naqeera I, Al-Khateeb Z, Al-Sheikh O. Experimental
20 investigation on the use of water-phase change material storage in conventional solar water
21 heating systems. *Energy Conversion and Management* 2010; 51: 1735-1740.
- 22 [20] Chaabane M, Mhiri H, Bournot P. Thermal performance of an integrated collector storage solar
23 water heater (ICSSWH) with phase change materials (PCM). *Energy Conversion and*
24 *Management* 2014; 78: 897-903.

- 1 [21] De Beijer HA. Product development in solar water heating. Proceedings of the 5th World
2 Renewable Energy Congress 1998; Florence, Italy: 201-204.
- 3 [22] Pugsley A, Smyth M, Mondol J, Zacharopoulos A, Mattia L. Experimental characterisation of a
4 flat panel integrated collector-storage solar water heater featuring a photovoltaic absorber and a
5 planar liquid-vapour thermal diode. EuroSun 2016; Spain, 11–14 October 2016.
- 6 [23] Rhee J, Campbell A, Mariadass A, Morhous B. Temperature stratification from thermal diodes in
7 solar hot water storage tank. *Solar Energy* 2010; 84: 507–511.
- 8 [24] Solar heating — Domestic water heating systems — Part 2: Outdoor test methods for system
9 performance characterization and yearly performance prediction of solar-only systems. ISO 9459-
10 2:1995. www.iso.org.
- 11 [25] Caouris YG, Souliotis M. Examination of top and bottom inlet position in horizontal mantle heat
12 exchanger solar thermosiphonic circuits. *International Journal of Sustainable Energy* 2013; 32:
13 421-433.
- 14 [26] Aranovitch E, Gilliaert D, Gillett WB, Bates JE. Test methods for solar water heating systems. In
15 Proceedings of 5th meeting of European Solar Collector and Systems Group, Seville, Spain,
16 (1986); 149-194.
- 17 [27] Tripanagnostopoulos Y, Souliotis M. ICS solar systems with horizontal cylindrical storage tank
18 and reflector of CPC or involute geometry. *Renewable Energy* 2004; 29: 13–38.
- 19 [28] Tripanagnostopoulos Y, Souliotis M. ICS solar systems with horizontal (E–W) and vertical (N–
20 S) cylindrical water storage tank. *Renewable Energy* 2004; 29: 73–96.
- 21 [29] Arnaoutakis N., Souliotis M., Papaefthimiou S. Comparative experimental life cycle assessment
22 of two commercial solar thermal devices for domestic applications. *Renewable Energy* 2017; 111:
23 187-200.

1 **APPENDIX: Uncertainty analysis**

2 The characteristics of the instrumentation used, for the measurement of various magnitudes, are:

- 3 • *Temperature*: by T type class 1 thermocouples, with $\pm 0.5^\circ\text{C}$ accuracy.
- 4 • *Irradiation*: by a CMP6 (Kipp & Zonen) pyranometer, with expected daily uncertainty $< 5\%$
- 5 • *Wind speed*: by an A100LK (Windspeed Limited / Vector Instruments) anemometer, with accuracy
- 6 $\pm 0.1 \text{ m}\cdot\text{s}^{-1}$ (in the range between 1 and $10 \text{ m}\cdot\text{s}^{-1}$). The wind speed is not involved in the uncertainty
- 7 calculations, as it consists an indicative measure, for the comparativeness of results.
- 8 • *Total pressure within the annulus*: by a PMP 4311 (General Electric Measurement & Control)
- 9 pressure transducer sensor, with $\pm 2.325\%$ accuracy (see below).
- 10 • *Aperture area*: It is assumed an accuracy of $\pm 0.01 \text{ m}^2$.
- 11 • *Volume of the inner vessel*: It is assumed an accuracy of $\pm 0.01 \text{ l}$ or $\pm 0.00001 \text{ m}^3$.
- 12 • *Accuracy of Logger readings, operated between -25°C to $+50^\circ\text{C}$, as given by the manufacturer*:
- 13 • Maximum logger contribution to type T thermocouple errors (uncompensated): $\pm 0.9^\circ\text{C}$.
- 14 • Voltage reading resolutions for single and differential measurements:

15 Table A1: Resolutions for single and differential measurements of the CR1000 Logger

Range	Scale	Resolution for Single Measurement	Resolution for Differential Measurement
1	$\pm 2.5 \text{ mV}$	$0.67 \mu\text{V}$	$0.33 \mu\text{V}$
2	$\pm 7.5 \text{ mV}$	$2.0 \mu\text{V}$	$1.0 \mu\text{V}$
3	$\pm 25 \text{ mV}$	$6.7 \mu\text{V}$	$3.33 \mu\text{V}$
4	$\pm 250 \text{ mV}$	$66.7 \mu\text{V}$	$33.3 \mu\text{V}$
5	$\pm 2500 \text{ mV}$	$667 \mu\text{V}$	$333 \mu\text{V}$
6	$\pm 5000 \text{ mV}$	$1333 \mu\text{V}$	$667 \mu\text{V}$

- 16 • Voltage reading accuracies for single and differential measurements:

17 Table A2: Accuracies for single and differential measurements of the CR1000 Logger

18	Logger Temperature:	-25 to $+50^\circ\text{C}$
	Scale Uncertainty	$\pm 0.12 \%$
	Long Term Stability	$\pm 0.25 \%$
19	Differential Offset	$\pm 0.02 \%$
	Noise	$0.34 \mu\text{V RMS}$ (negligible)

1 • *Calculation of RMS uncertainties:*

2 **1. Temperature measurement uncertainty, $\sigma(T)$:**

$$3 \quad \sigma(T) = \sqrt{\sigma_{\text{thermocouple}}^2(T) + \sigma_{\text{Logger}}^2(T)} = \sqrt{0.5^2 + 0.9^2} = \pm 1.030 \text{ }^\circ\text{C} \approx \pm 1 \text{ }^\circ\text{C} \quad (\text{A1})$$

4 **2. Volume per aperture area measurement uncertainty, $\sigma(V_T/A_a)$:**

$$5 \quad \sigma(V_T / A_a) = \sqrt{\sigma^2(V_T) + \sigma^2(A_a)} = \sqrt{0.01^2 + 0.01^2} = \pm 0.014 \text{ lm}^{-2}$$

6 **3. Irradiation measurement percent uncertainty, $\sigma\%(Q_R)$:**

7 The pyranometer has sensitivity of $17.29 \text{ } \mu\text{V} \cdot \text{W}^{-1} \cdot \text{m}^2$, so for intensities between $0 - 145 \text{ W} \cdot \text{m}^{-2}$ the
 8 logger range 1 is used, for intensities between $146 - 434 \text{ W} \cdot \text{m}^{-2}$ the logger range 2 is used, while for
 9 greater intensities the logger range 3 finally used. It is noted that the measurements for the radiation
 10 intensities have been made differentially. The total solar radiation intercepted by the aperture surface
 11 A_a (m^2) of the system during the time interval Δt_D (i.e. 43,200 s) from the startup time $t_{i,D}$ ($\sim 6:30$ in
 12 the morning) till the end time $t_{f,D}$ ($\sim 18:30$ in the afternoon, as solar noon occurs at about 12:30 pm),
 13 is given by the parameter Q_R (J), which is determined by the integration of the solar radiation intensity
 14 $G(t)$ over the time interval Δt_D :

$$15 \quad Q_R = A_a \cdot \int_{t_{i,D}}^{t_{f,D}} G(t) \cdot dt \quad (\text{A2})$$

16 So the RMS percentage uncertainty of Irradiation measurement can be expressed as:

$$17 \quad \sigma\%(Q_R) = \sqrt{\sigma_{\text{daily uncertainty}}^2 + \left(\sigma_{\text{Logger Scale Uncertainty}} + \sigma_{\text{Logger Long Term}} \right)^2 + \sigma_{\text{aperture accuracy}}^2} \quad (\text{A3})$$

18 Using the values (Table A2) of 0.12 % for Logger Scale Uncertainty, 0.25 % for Logger Long Term
 19 Stability and supposing pyranometer daily uncertainty $\sigma_{\text{daily uncertainty}} = \pm 5\%$, then for mean irradiation
 20 intensities G , $\sigma\%(Q_R) = \pm 5.014 \%$ as calculated by (A3). As it is evident the Logger contribution to
 21 RMS error is negligible, which is dominated by the daily uncertainty of pyranometer error.

1 4. Pressure measurement percent uncertainty, $\sigma_{\%}(P)$:

2 The pressure transducer sensor PMP 4311 operates in the range between 0.2 to 4,500 Volts DC
3 according to the relation $P = 2.325 \cdot V - 465$, where V (in mV) is the voltage output of the sensor with
4 accuracy $\pm 1\%$. The percent RMS uncertainty on P is then calculated as:

$$5 \quad \sigma_{\%}(P) = \frac{\partial P}{\partial V} \cdot \sigma_{\%}(V) = 2.325 \cdot \sigma_{\%}(V) = \pm 2.325\% \quad (\text{A4})$$

6 5. Calculation of percent uncertainty in useful energy $\sigma_{\%}(Q_{w,inner})$:

7 The useful energy output is calculated as $Q_{w,inner} = M_{w,inner} \cdot \bar{C}_{p,w} \cdot (T_{f,D} - T_{i,D})$, where:

$$8 \quad M_{w,inner} = \rho \cdot V_{T,inner} \quad (\text{A5})$$

9 $V_{T,inner}$ Volume of the inner vessel m^3

10 ρ Water density at temperature T $\text{kg} \cdot \text{m}^{-3}$

11 $\bar{C}_{p,w}$ Mean (T_o, T_i) water specific heat capacity $\text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$

12 $T_{i,D}, T_{f,D}$ Initial and final water temperatures K

13 The percent RMS uncertainty on $Q_{w,inner}$ is then calculated as:

$$14 \quad \sigma_{\%}(Q_{w,inner}) = \sqrt{\sigma_{\%}^2(V_{T,inner}) + \sigma_{\%}^2(\rho) + \sigma_{\%}^2(\bar{C}_{p,w}) + \sigma_{\%}^2(T_{f,D} - T_{i,D})} \quad (\text{A6})$$

15 The density and the specific heat capacity of water are calculated, according to ISO-9459 and [25], with
16 the relations:

$$17 \quad \rho(\theta) = 1000,67 - 7,3845 \times 10^{-2} \theta - 3,547 \times 10^{-3} \theta^2 \quad \text{kg} \cdot \text{m}^{-3}, \quad (\text{A7})$$

$$18 \quad \bar{c}_p(\theta_1, \theta_2) = 4,20028 - 5,048 \times 10^{-4} (\theta_1 + \theta_2) + 4,097 \times 10^{-6} [(\theta_1 + \theta_2)^2 - \theta_1 \theta_2] \quad \text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}, \quad (\text{A8})$$

19 with $\theta, \theta_1, \theta_2$ in $^{\circ}\text{C}$. The percent uncertainty of $\rho, \bar{c}_{p,w}$ are expressed as:

$$20 \quad \sigma_{\%}(\rho) = \frac{\partial \rho}{\partial \theta} \sigma_{\%}(\theta), \quad \sigma_{\%}(\bar{c}_{p,w}) = \sqrt{\left(\frac{\partial \bar{c}_{p,w}}{\partial \theta_1} \sigma_{\%}(\theta_1) \right)^2 + \left(\frac{\partial \bar{c}_{p,w}}{\partial \theta_2} \sigma_{\%}(\theta_2) \right)^2} \quad (\text{A9})$$

21 Where:

$$1 \quad \frac{\partial \rho}{\partial \theta} = -7,3845 \times 10^{-2} - 7,094 \times 10^{-3} \theta \quad (\text{A10})$$

$$2 \quad \frac{\partial \bar{c}_{p,w}}{\partial \theta_1} = -5,048 \times 10^{-4} + 4,097 \times 10^{-6} (2\theta_1 + \theta_2) \quad (\text{A11})$$

$$3 \quad \frac{\partial \bar{c}_{p,w}}{\partial \theta_2} = -5,048 \times 10^{-4} + 4,097 \times 10^{-6} (2\theta_2 + \theta_1) \quad (\text{A12})$$

4 So, calculating the percent uncertainty of ρ , $\bar{c}_{p,w}$, computing the uncertainties $\sigma\%(T_{f,D} - T_{i,D})$ from the
 5 relation (A1) and taking into account the $\sigma\%(V_{T,inner})$, the percent uncertainty $\sigma\%(Q_{w,inner})$ is extracted
 6 by the relation (A7). It is found that $\sigma\%(Q_{w,inner})$ uncertainty is dominated by $\sigma\%(T_{f,D} - T_{i,D})$ and
 7 $\sigma\%(V_{T,inner})$, while the contribution of all other sources of uncertainties can be considered negligible.

8 **6. Calculation of percent uncertainty on system mean daily efficiency $\sigma\%(\eta_D)$.**

9 The system mean daily efficiency is calculated as $\eta_D = Q_{w,inner} / Q_R$. So a percent RMS uncertainty
 10 is expressed as:

$$11 \quad \sigma\%(\eta_D) = \sqrt{\sigma\%^2(Q_{w,inner}) + \sigma\%^2(Q_R)} \quad (\text{A13})$$

12 The $Q_{w,inner}$ and Q_R percent uncertainties are calculated by the relations (A6) and (A3), respectively.

13 **7. Calculation of percent uncertainty on system thermal losses $\sigma\%(U_S)$.**

14 The system thermal losses coefficient is calculated as:

$$15 \quad U_S = \left[(\rho \cdot \bar{c}_{p,w} \cdot V_{T,inner}) / \Delta t_N \right] \cdot \ln \left[(T_{i,N} - T_{a,N}) / (T_{f,N} - T_{a,N}) \right] \quad (\text{A14})$$

16 The percent RMS uncertainty on U_S , taking also into consideration the all above, is then calculated as:

$$17 \quad \sigma\%(U_S) = \sqrt{\sigma\%^2(V_{T,inner}) + \sigma\%^2(\rho) + \sigma\%^2(\bar{c}_{p,w}) + \sigma\%^2(T_{i,N} - T_{a,N}) + \sigma\%^2(T_{f,N} - T_{a,N}) + \sigma\%^2(\Delta t_N)} \quad (\text{A15})$$

18 For the extraction of the above uncertainty propagation analysis, the influence of uncertainty on
 19 coefficients in equations (A8) and (A9), uncertainty on Logger timing (sampling intervals) and inertia
 20 of sensors were considered negligible.

Highlights

- Heat diode mechanism in heating of an ICS solar water heater.
- Effect of the partial vacuum in the process of heating and retention of heat.
- Effective daily and night operation of the ICS system.
- Comparative experimental study between the ICS device and a FPTU.