

Design, experimental characterization and numerical analysis of a compact nearly spherical solar dryer

Alessandra Nigro^{a,*}, Sebastiano Tomassetti^a, Tariku Negash Demissie^a,
Giovanni Di Nicola^{a,b}

^a Department of Industrial Engineering and Mathematical Sciences, Marche Polytechnic University, Via Brecce Bianche 12, 60131 Ancona, Italy

^b Construction Technologies Institute, National Research Council (ITC-CNR), 35127 Padova, Italy

HIGHLIGHTS

- A compact nearly spherical solar dryer operating without sun tracking was experimentally and numerically investigated.
- Indoor solar-simulator tests provided controlled and reproducible conditions, while indoor and outdoor experiments confirmed temperature levels suitable for fruit drying.
- The nearly spherical geometry promoted a more uniform distribution of absorbed solar energy within the drying chamber.
- Indoor temperature measurements enabled validation of a CFD model which was then used as a design tool.
- High-conductivity absorbent materials improved temperature uniformity while reducing buoyancy-driven ventilation, revealing a passive-design trade-off.

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ABSTRACT

A compact solar dryer with a nearly spherical geometry is investigated through a combined experimental and numerical approach to assess its thermal behaviour under controlled and real operating conditions. An extensive experimental campaign is carried out, including outdoor tests under natural solar irradiation and indoor experiments performed using a solar simulator, which allows reproducible testing conditions for the experimental characterization of the dryer and subsequent numerical validation. No-load thermal tests are conducted to characterize the temperature field inside the drying chamber, verifying the suitability of the achieved temperature levels for apple drying. Complementary drying tests with apple slices are then performed, confirming the actual drying capability of the prototype under both indoor and outdoor conditions. Based on the no-load experimental temperature measurements, a three-dimensional CFD model accounting for buoyancy-driven airflow, conjugate heat transfer, and radiative effects is developed and validated, with average deviations comparable to the measurement uncertainty. The validated CFD model is subsequently employed as a design tool to investigate the influence of the absorbent surface material. Replacing painted steel with aluminium improves vertical temperature uniformity due to the higher thermal conductivity, but reduces the intensity of buoyancy-driven ventilation. Overall, the combined experimental and numerical analysis provides quantitative insight into the thermal performance of the proposed dryer and supports informed material-based design choices for naturally ventilated solar drying systems.

1. Introduction

Food drying is one of the most widespread and well-known preservation techniques, increasing food shelf life and reducing post-harvest losses and spoilage. It can be especially useful for reducing food waste in the household sector, which accounted for 60% of global food waste

in 2022 [1]. Currently, the most widely used drying systems rely on conventional energy sources, primarily fossil fuels, which have a negative environmental impact due to greenhouse gas emissions [2]. A sustainable and easily accessible alternative for the drying process has been identified in solar energy, which is expected to play an increasingly central role in the future [3]. The growing interest in solar drying systems

* Corresponding author.

Email address: a.nigro@staff.univpm.it (A. Nigro).

reflects the broader importance of solar-powered technologies in areas such as electricity generation [4,5], fuel production [6,7], heating [8], cooking [9], and environmental sustainability [10,11].

Although open sun drying is one of the oldest traditional methods for food preservation, it has several limitations that can compromise the quality of the dried product, such as uncontrolled operating conditions and food contamination [3,12]. Consequently, as evidenced by several studies conducted over the years [2,3,13], a large number of solar dryers characterized by different sizes, shapes, and design features have been developed to ensure high performance and a suitable drying time.

Solar dryers are generally classified on the basis of the sun radiation exposure to the products to be dried, i.e., direct, indirect, and mixed mode, and air circulation in the system, i.e., active and passive [2]. Over the years, various solar dryer designs have been developed to shorten drying time, enhance drying efficiency, regulate drying air temperature, and ensure continuous drying [14]. In this regard, solar dryers have been integrated with various technologies, such as photovoltaic/thermal panels [15], phase change materials [16], and biomass [17], to develop hybrid systems with enhanced drying performance.

Various simple, compact, and low-cost solar dryers have been specifically designed for households and small-scale farmers [18,19]. However, some of them have drawbacks, such as the need for sun tracking, significant spatial requirements, and high cost [20]. For this reason, increasing attention has been devoted to simple and low-cost designs that do not require sun tracking. Some examples are described below.

Fournier and Guinebault [21] developed a mathematical model of a passive African solar dryer, known as the “shell” dryer, for determining the optimal design of the device to allow its best use. The system is a very simple domestic dryer, comprising an upper cone with an opening at the top, a cylinder with tray holders mounted inside, and a bottom cone drilled with several holes. All components are black to maximize the capture of sunlight. Although the model was validated against measurements from the literature, a systematic analysis of dryer performance based on experimental tests has not been reported.

The “shell” dryer has recently been used by Bignan-Kagomna et al. [22] to dry onions and tomatoes. The results of a drying test with onions and one with tomatoes, both carried out on three consecutive days, have been reported, obtaining an efficiency of the dryer equal to 29.88%.

Joshi et al. [23] developed and experimentally studied a very simple direct conical shape solar dryer, which can exploit solar radiation from all directions. It also comprises LEDs to maintain the temperature during the night and provide heat for the drying process. No-load thermal tests were carried out to evaluate the temperatures within the drying chamber, yielding values of about 40 °C. In addition, the results of drying tests with potato slices, rice, and banana slices were presented, showing that the maximum moisture reduction occurred in the potato slices.

Recently, Kokate et al. [20] designed a hemispherical solar dryer to overcome drawbacks such as tracking and high spatial requirements. The hemispherical prototype has a diameter of 1.5 m and a height of 0.75 m and comprises two concentric hemispherical domes made of polycarbonate and aluminium. The air gap between the domes creates a greenhouse effect, enhancing the temperature of the hemispherical solar dryer. The prototype was experimentally compared with traditional methods for drying grapes to raisins. It provided better results than the other methods, achieving an efficiency of 5.67% and drying the load in only 15 days.

Despite the wide variety of solar dryer configurations developed over the years, their performance is still strongly affected by complex coupled phenomena, such as buoyancy-driven airflow, radiative heat transfer, and non-uniform temperature distribution inside the drying chamber. These aspects are often difficult to fully characterize through experimental measurements alone, especially in compact passive systems, where access for instrumentation is limited and the flow is driven by small temperature gradients.

In this framework, computational fluid dynamics (CFD) has become an effective tool for complementing experimental investigations of solar

dryers, enabling detailed analysis of airflow patterns and thermal fields through the numerical solution of the governing fluid flow and heat transfer equations. By providing access to transport phenomena that are difficult to measure experimentally, CFD enhances the understanding of drying mechanisms and supports energy-efficient design optimization. Consequently, CFD has been increasingly adopted for the performance assessment and improvement of solar drying systems. Representative recent CFD investigations on solar dryers include detailed thermo-fluid dynamic analyses of airflow and temperature distribution in indirect drying systems [24], validated numerical-experimental modelling of solar dryers incorporating thermal storage components [25], and optimization-oriented simulations aimed at improving dryer performance [26,27]. These studies demonstrate the growing role of CFD as a physics-based framework for analysing transport phenomena and guiding the design and optimization of solar drying systems, a development also reflected in recent bibliometric analyses identifying CFD-based modelling among the most rapidly expanding research directions in this field [28,29].

This work addresses the experimental and numerical characterization of a compact solar dryer with a nearly spherical geometry, specifically conceived for household scale food drying applications. The proposed device consists of two black nearly hemispherical shells, allowing effective absorption of solar radiation from multiple directions without the need for sun tracking, while air circulation inside the drying chamber is driven by buoyancy-induced natural convection. One of the main features of the proposed dryer that distinguishes it from those in the literature is its compactness, as well as its ease of use and transport. In fact, it has been designed as a low-cost, lightweight, and easily transportable system, assembled from widely available components. Unlike what is usually done in the literature for simple and low-cost designs, the thermal behaviour and drying capability of the prototype are systematically investigated through an extensive experimental campaign, including no-load thermal tests and drying tests conducted under both controlled indoor conditions and outdoor natural solar irradiation. This complete experimental analysis enables the evaluation of its drying capability under different conditions and the investigation of its performance reproducibility. To complement the experimental analysis, a three-dimensional CFD model is developed and validated against temperature measurements obtained during the indoor tests. It is worth noting that the CFD model has been validated using indoor experimental tests conducted under controlled, repeatable conditions, ensuring high accuracy in the model development. The validated numerical framework is then employed to investigate the influence of the absorbent surface material on the thermal and fluid-dynamic behaviour of the dryer, providing insight into the trade-offs between temperature uniformity and natural ventilation and supporting informed material-based design choices.

The remainder of the paper is organized as follows. **Section 2** describes the design and assembly of the solar dryer prototype. **Section 3** presents the experimental setup and procedures adopted for both indoor and outdoor testing campaigns. The experimental results are discussed in **Section 4**. **Section 5** is devoted to the numerical methodology and CFD analysis, including model validation and the investigation of alternative design configurations. Finally, conclusions and perspectives for future work are summarized in the last section.

2. Design and assembly of the solar dryer prototype

The solar dryer prototype is shown in **Fig. 1**. The nearly spherical drying chamber is obtained by joining two commercially available low-cost stainless steel shells, each characterized by a maximum horizontal diameter of 360 mm, a height of 160 mm, and a thickness of 0.7 mm. The external surfaces of the shells are painted black to enhance solar radiation absorption.

The shells are held in position by bolts and nuts (labeled as C1 in **Fig. 1(b)**) at three evenly distributed points along the circumferential

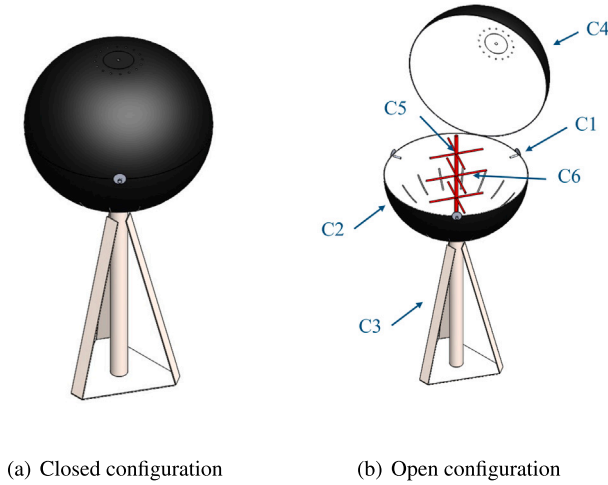


Fig. 1. Schematic view of the nearly spherical solar dryer.

joining edge of the two shells. The lower shell (C2) is equipped with 16 longitudinal apertures at the bottom to allow air to flow into the dryer by natural convection. Each aperture is 65 mm long and 1 mm wide, with a 3.5 mm hole at the lower end to enable the thermocouple wire to pass through. The lower end of the apertures originates at a curvilinear distance of 50 mm from the lowest point of the shell surface. A circular hole with a radius of 10 mm was drilled at the lowest point of the same shell to secure a vertical stand (C3). The vertical stand (C3), 400 mm in height, was used to support the dryer during outdoor tests. A bolt-nut assembly provided a rigid mechanical connection between the stand and the lower shell. As shown in Fig. 1(b), the upper shell (C4) can be removed to allow loading of the product into the drying chamber. In this shell, a central hole with a radius of 4 mm was drilled, with its center located at the highest point of the shell surface. Additionally, 16 holes with a radius of 2 mm were drilled along a circumference the centers of which lie at a curvilinear distance of 60 mm from the highest point of the shell, measured along the shell surface.

To hold the thermocouples inside the dryer and to support a circular mesh tray used during drying tests, a vertical stainless steel rod (in red and labeled as C5 in Fig. 1(b)), 310 mm in height and 10 mm in diameter, was positioned along the central axis and secured through the hole in the lower shell. Twelve stainless steel rods (in red and labeled as C6 in Fig. 1(b)), each with a diameter of 5 mm, were mounted on the central rod. Specifically, the twelve rods were arranged on three levels, with four rods per level, spaced at equal vertical intervals of 80 mm from the bottom. The length of each rod was selected to be equal to half of the

local radius of the drying chamber cross-section at the corresponding vertical level.

This configuration allows precise positioning of 15 thermocouples at symmetrically distributed locations within the drying chamber: 12 are radially arranged on the horizontal rods, while the remaining three are positioned along the vertical rod at the corresponding three levels. Consistent with the rod arrangement, three horizontal measurement planes are defined along the chamber height (bottom, middle, and top), each including five thermocouples symmetrically distributed around the central axis. The measurement points and their spatial arrangement are illustrated in Fig. 2. The corresponding point labels are used consistently throughout the paper for the presentation and discussion of both experimental and numerical results. The radial distance of the sensors varies according to the local chamber radius, resulting in distances of 90 mm on the middle plane and 78.5 mm on the top and bottom planes, while the vertical spacing between consecutive levels is 80 mm. This layout enables evaluation of both radial temperature uniformity and vertical thermal gradients inside the chamber.

During drying tests, a circular mesh tray with a diameter of 300 mm was installed at the midpoint of the vertical rod to support the product inside the drying chamber.

3. Experimental setup and procedure

Experimental tests were conducted both in a controlled indoor environment and in outdoor conditions under natural solar irradiation to evaluate the performance of the proposed solar dryer under different operating conditions.

Indoor experiments were carried out in the Dipartimento di Ingegneria Industriale e Scienze Matematiche (DIISM) at Università Politecnica delle Marche (UNIVPM) using a large-scale solar simulator [30]. The system comprises 20 metal halide lamps, each having a power of 360 W, positioned at the focal point of parabolic reflectors to deliver a luminous flux with uniform distribution over the target surface. The lamps are installed on an aluminium-frame structure and arranged in a 4×5 matrix configuration for creating a large target surface with the same luminous characteristics. The system has a vertical guide to change the lamp–target distance for controlling the intensity of the simulated radiation. This simulator was experimentally validated according to the IEC 60,904–9 standard. It achieved Class C performance in terms of spatial uniformity over a surface of $200 \times 90 \text{ cm}^2$, while the temporal stability measurements satisfied the requirements for Class A.

As shown in Fig. 3, the prototype was placed in the focal zone of the simulator without the vertical stand, in order to ensure consistent irradiance in the zenith direction. The irradiance was maintained at a constant 900 W m^{-2} to simulate standardized solar conditions. Indoor testing allows the analysis of the prototype under controlled conditions, in which irradiance is held constant and the effects of

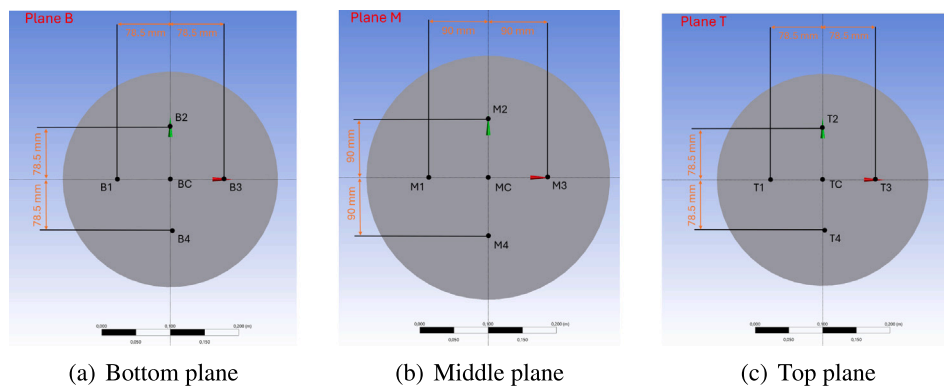


Fig. 2. Schematic view of the planes and points adopted for the experimental analysis.

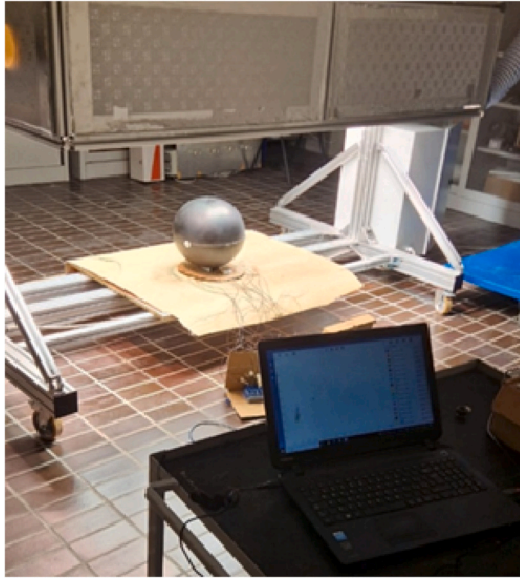


Fig. 3. Experimental configuration used in the indoor tests with a solar simulator.

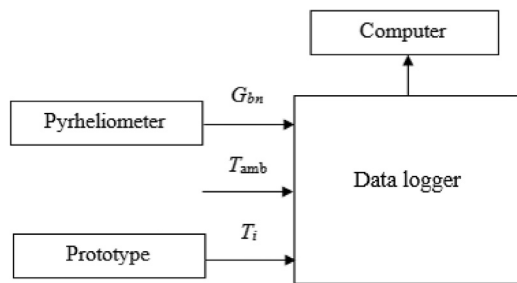


Fig. 4. Experimental setup used in the outdoor tests.

external environmental factors, such as wind and clouds, are eliminated. Therefore, this approach ensured that the prototype's thermal performance was evaluated under repeatable boundary conditions. However, during these tests, only a zenithal solar configuration was simulated, and the direction of the incident radiation could not be varied. To evaluate the thermal distribution inside the drying chamber, 15 T-type thermocouples with an uncertainty of $\pm 1^\circ\text{C}$ were placed in the prototype as described in Section 2. An additional thermocouple was placed externally, near the longitudinal apertures of the lower shell, to record the inlet air temperature. The temperature values were acquired in real time using two Pico Technology TC-08 thermocouple data loggers, which were connected to a laptop computer running PicoLog software.

Outdoor tests were conducted from April to June 2025 on the rooftop of the DIISM at UNIVPM (43.5871° N, 13.5149° E). Fig. 4 shows the experimental setup used for the outdoor experimental campaign. In addition to the 16 thermocouples for measuring the temperatures inside the prototype and the inlet air temperature (T_i in Fig. 4), the ambient temperature (T_{amb}) and the direct normal solar irradiance (G_{bn}) were also measured. A T-type thermocouple was used for T_{amb} , while a normal incidence pyrheliometer (NIP) with a linearity of $\pm 0.5\%$ in the $0\text{--}1400\text{ W m}^{-2}$ range was used for G_{bn} . The measured values were also recorded using Pico Technology TC-08 thermocouple data loggers.

The prototype, mounted on the vertical stand, was positioned on the floor in an unshaded area and not moved during experiments. As shown in Fig. 5, some tests were performed by placing the dryer on a flat L-shaped reflector to enhance the amount of solar radiation incident on

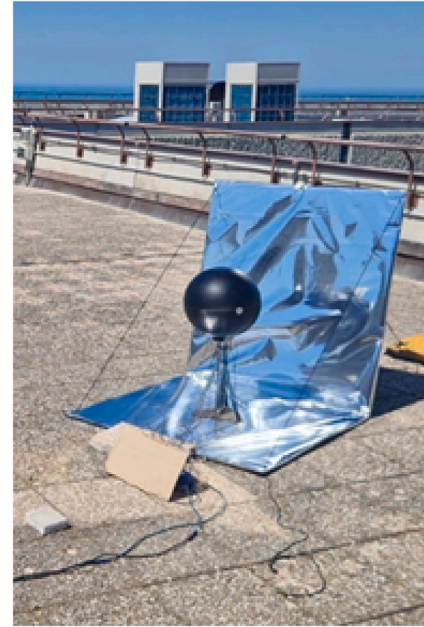


Fig. 5. L-shaped reflector used in some outdoor experimental tests.

its surface. The reflector was made of cardboard covered with a commercially available Mylar film and has a $104 \times 107\text{ cm}^2$ base and a $104 \times 119\text{ cm}^2$ vertical back. During the tests, the reflector was oriented to face south, enhancing the redirection of sunlight towards the dryer absorbent surface during daylight hours. The purpose of using the reflector was to study its contribution to improving the solar dryer thermal performance under outdoor conditions.

The experimental tests were designed to assess temperature distribution, repeatability, and the system capacity to dry the loaded product under controlled and natural solar input. Indoor and outdoor tests without load were conducted to evaluate whether the temperature distribution within the drying chamber is suitable for drying under the studied conditions. In addition, indoor and outdoor drying tests were carried out by loading the prototype with apple slices as the drying sample. Apples were selected as the test product due to their widespread production in Italy and their frequent use in fruit drying investigations reported in the literature [3,12,31]. In addition, the temperatures achieved by the proposed solar dryer are consistent with typical operating conditions for fruit and vegetable drying. The apple slices were uniformly distributed on the circular mesh tray in the chamber and weighed before and after each test using a digital balance, allowing calculation of moisture loss, drying rate, and drying efficiency. The average drying rate (DR_{av}) was calculated for each test as follows [32]:

$$DR_{av} = \frac{m_w}{m_i \Delta t} \quad (1)$$

where $m_w = m_i - m_f$ is the mass loss during the drying test, m_i is the sample mass before drying, m_f is the sample mass after drying, and Δt is the drying time. The overall drying efficiency (η_d) of the tested prototype was determined for complete drying experiments. The following expression of η_d for a solar dryer based on natural convection was used [33]:

$$\eta_d = \frac{m_w L}{G_{bn} A_a \Delta t} \quad (2)$$

where L is the latent heat of evaporation of water at the average temperature inside the prototype and A_a is the aperture area of the prototype, i.e., the projection of the area delimited by the prototype's outer edges

onto a plane perpendicular to the sun's rays. The latent heat of evaporation of water at 40 °C was used in the calculations ($L = 2040 \text{ kJ kg}^{-1}$). In the tests without the reflector, A_a was determined as the orthogonal projection of an oblate spheroid onto a plane perpendicular to the sun's rays, by using the following equation:

$$A_a = \pi a^2 \sqrt{\sin^2 \beta + \frac{c^2}{a^2} \cos^2 \beta} \quad (3)$$

where $a = 0.18 \text{ m}$ is the equatorial semi-axis, $c = 0.16 \text{ m}$ is the polar semi-axis, and β is the solar elevation angle. In the indoor test, A_a is equal to 0.102 m^2 since the irradiance is in the zenith direction ($\beta = 90^\circ$). Instead, the actual aperture area of the prototype varies with the sun's elevation during outdoor tests. Moreover, in the test with the reflector, the aperture area also accounts for the additional variable irradiance provided by the reflecting surfaces. However, to simplify the analysis, the effective aperture area ($A_{a,\text{eff}}$) was calculated by considering β at solar noon for each test day and assuming a complete projection of the prototype onto the reflecting surfaces. Under this assumption, the effective aperture area was determined as:

$$A_{a,\text{eff}} = A_a (1 + 2R) \quad (4)$$

where R is the reflection coefficient of the reflecting surfaces. To account for potential losses due to surface roughness, folds, imperfect specular reflection, and non-optical-grade metallization, a conservative value of $R = 0.7$ was assumed for the reflective Mylar film. For the drying tests carried out in May ($\beta = 65^\circ$) and June ($\beta = 69^\circ$) with the reflector, the following values of $A_{a,\text{eff}}$ were determined and used in the calculations: 0.240 m^2 and 0.241 m^2 .

The evaluation of these parameters, combined with the thermal data, enabled the assessment of the dryer effectiveness in moisture removal. This allowed evaluation of its potential suitability for food preservation under both simulated and natural solar conditions. Finally, to ensure the reproducibility of the experimental behaviour of the proposed solar dryer, drying tests were repeated on consecutive days using the same prototype configuration, data acquisition system, sample preparation procedure, and product loading arrangement.

4. Experimental results

In this section, the results of the indoor and outdoor tests are reported and commented.

4.1. Indoor tests

An indoor thermal test lasting approximately an hour and a half was conducted without a load by maintaining a constant irradiance of 900 W m^{-2} . Fig. 6 shows the behaviour of the 16 temperatures recorded during the test, while their average and maximum values are reported in Table 1. In particular, both the average temperatures over the entire test duration ($T_{\text{av,tot}}$) and those obtained under steady-state conditions during the last 10 min of the test ($T_{\text{av,ss}}$) are reported, the latter being used as reference values for the CFD model validation. It is evident that the prototype heated the inlet air to temperatures suitable for drying (above 40 °C and below 70 °C) under the test conditions. Moreover, it showed an average vertical temperature gradient of about 8 °C, suggesting the possibility of natural convection.

Three indoor drying tests were performed on consecutive days by using the solar simulator to dry apples. In the first test, 96 g of apples, cut into 2 mm thick slices, were placed on the circular mesh tray loaded into the prototype. The mass of the loaded product was measured at the start and end of each test. An airtight container was used to store the dried product at the end of each test to prevent moisture absorption from the environment. The test duration, the initial and final masses of product, the mass percentage reduction, and the average drying rate for the three drying tests are reported in Table 2. By contrast, Table 3 presents the average and maximum temperatures recorded through the

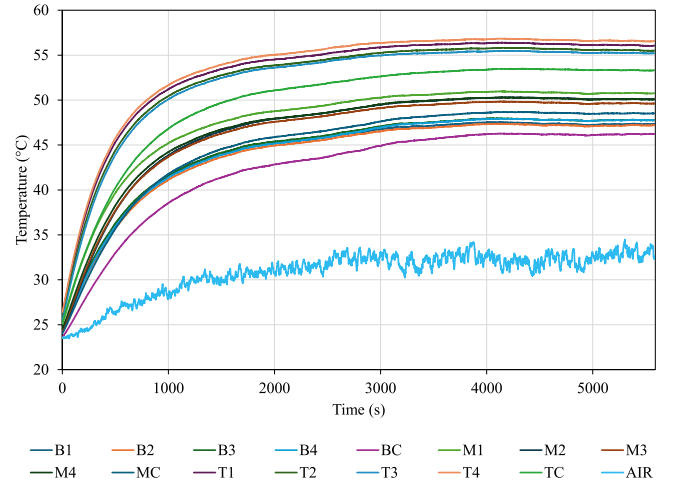


Fig. 6. Behaviour of temperatures recorded during the indoor no-load thermal test. The temperature sensor locations are shown in Fig. 2. AIR corresponds to the inlet air temperature.

Table 1

Average temperatures over the entire test ($T_{\text{av,tot}}$) and under steady-state conditions ($T_{\text{av,ss}}$), together with maximum temperatures (T_{max}), recorded during the indoor no-load thermal test. The temperature sensor locations are shown in Fig. 2. AIR corresponds to the inlet air temperature.

Position	$T_{\text{av,tot}}$ (°C)	$T_{\text{av,ss}}$ (°C)	T_{max} (°C)
B1	44.0	47.2	47.4
B2	44.5	47.8	48
B3	44.4	47.8	47.9
B4	44.2	47.3	47.6
BC	42.3	46.2	46.3
M1	46.7	50.1	50.3
M2	46.4	49.6	49.8
M3	46.8	50.3	50.3
M4	47.7	50.8	51.0
MC	44.9	48.5	48.7
T1	52.5	55.6	55.8
T2	52.2	55.3	55.5
T3	53.6	56.6	56.8
T4	53.2	56.1	56.4
TC	49.7	53.3	53.5
AIR	30.8	32.3	34.5

Table 2

Test duration and performance parameters of the indoor apple drying tests.

	Duration (h:min)	Initial mass (g)	Final mass (g)	Mass reduction (%)	DR_{av} ($\text{g g}^{-1} \text{ h}^{-1}$)
Test 1	3:00	96.0	69.3	27.8	0.09
Test 2	3:07	69.0	38.1	44.8	0.14
Test 3	2:54	38.0	17.7	53.4	0.18

16 thermocouples during the indoor drying tests. From the tables, it can be noted that the prototype maintained an average temperature of about 40 °C at the middle plane during all tests. The mass percentage reduction and the average drying rate increased in successive tests, which can be partly attributed to the lower initial mass of the samples. Overall, the prototype reduced the mass of the apple slices by about 81.6% in about 9 h while maintaining an average temperature above 40 °C with $\eta_d = 6.3\%$. These results demonstrate the good drying performance of the studied solar dryer under the indoor test conditions.

4.2. Outdoor tests

Five outdoor thermal experiments were conducted between April and June 2025, with and without the L-shaped reflector. Table 4 reports the

Table 3

Average ($T_{av,tot}$) and maximum (T_{max}) temperatures recorded during the indoor apple drying tests. The temperature sensor locations are shown in Fig. 2. AIR corresponds to the inlet air temperature.

Position	Test 1		Test 2		Test 3	
	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)
B1	39.1	42.1	39.5	43.2	41.5	46.0
B2	38.9	42.0	39.6	43.3	41.7	46.3
B3	38.6	41.6	39.3	42.9	41.8	46.4
B4	38.3	41.2	39.2	42.6	42.3	47.0
BC	38.0	41.1	38.7	42.3	41.5	46.0
M1	37.4	40.2	38.8	42.7	42.2	48.2
M2	38.0	40.6	38.7	42.3	41.6	47.0
M3	37.8	40.7	38.4	42.0	41.1	46.1
M4	37.8	40.5	39.2	42.1	42.4	47.4
MC	38.9	41.8	39.9	43.5	42.6	47.3
T1	50.2	53.1	51.4	54.8	52.4	56.4
T2	49.0	51.8	49.8	53.2	51.1	55.3
T3	49.3	52.1	50.1	53.6	51.6	55.8
T4	50.7	53.7	51.6	55.1	53.1	57.2
TC	45.7	48.8	46.5	50.2	48.4	52.9
AIR	32.8	36.4	32.6	37.3	34.0	36.6

average and maximum temperatures measured in the solar dryer during these tests without load, together with the average values of T_{amb} and G_{bn} , and the test start and end times. It can be noted that the tests performed in April were carried out around noon under similar environmental conditions. However, the prototype reached temperatures above 40 °C, suitable for drying, only when the reflector was used. This outcome is evident in Fig. 7, which shows the temperature behaviour recorded during the tests carried out on 4 April 2025. This could be because the radiation directly absorbed by the dryer is not sufficient to reach suitable drying temperatures due to the low ambient temperature. Instead, Table 4 shows that the prototype with the reflector reached suitable temperatures in June, despite the relatively low G_{bn} .

Outdoor drying tests were performed in May and June 2025 on two consecutive days by using the reflector. Masses of 93 g and 137.6 g of apples, cut into 2 mm thick slices, were loaded into the prototype in the first test of May and June, respectively. Also for the outdoor tests, the mass of the loaded product was measured at the start and end of each test, and the dried product was stored in an airtight container at the end of each test. The test duration, the initial and final product masses,

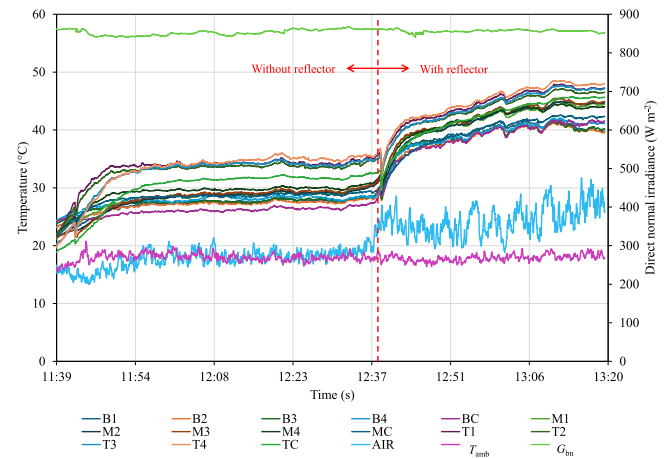


Fig. 7. Behaviour of temperatures recorded during the outdoor no-load thermal tests both without and with reflector carried out on 4 April 2025. The temperature sensor locations are shown in Fig. 2. AIR corresponds to the inlet air temperature.

the mass percentage reduction, and the average drying rate for the four drying tests are reported in Table 5. The average and maximum temperatures measured during the outdoor drying tests are shown in Table 6. It can be seen that the values of temperature, mass percentage reduction, and average drying rate reported in these tables are of the same order of magnitude as those obtained in the indoor test, indicating that the prototype can achieve comparable drying performance under indoor and outdoor conditions. In this case, the prototype reduced the mass of the apple slices by about 83.2% in about 9 h with $\eta_d = 2.6\%$ during the May tests and by about 86.4% in about 10 h with $\eta_d = 4.2\%$ during the June tests. It is worth noting that the lower efficiency values obtained in the outdoor tests compared with the indoor tests could be partly attributed to a possible overestimation of $A_{a,eff}$, since approximate values of the solar elevation angle and reflection coefficient were used in its calculation.

In addition to confirming the capability of the solar dryer to dry apples under outdoor test conditions, provided that the reflector is used, these results indicate that the drying behaviour of the prototype

Table 4

Average ($T_{av,tot}$) and maximum (T_{max}) temperatures recorded during the outdoor no-load thermal test. The temperature sensor locations are shown in Fig. 2. AIR corresponds to the inlet air temperature.

Reflector	3 April 2025 11:22–14:33 $T_{amb,av} = 16.5$ °C $G_{bn,av} = 811.1$ W m ⁻²		4 April 2025 12:39–12:39 $T_{amb,av} = 18.0$ °C $G_{bn,av} = 853.4$ W m ⁻²		4 April 2025 12:40–13:20 $T_{amb,av} = 17.9$ °C $G_{bn,av} = 856.7$ W m ⁻²		11 April 2025 11:15–14:41 $T_{amb,av} = 22.8$ °C $G_{bn,av} = 854.1$ W m ⁻²		6 June 2025 12:55–16:08 $T_{amb,av} = 32.6$ °C $G_{bn,av} = 713.4$ W m ⁻²	
	no	yes	no	yes	yes	yes	yes	yes	yes	yes
Position	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)
B1	22.2	25.0	27.1	30.4	39.3	42.0	45.1	50.5	50.3	55.3
B2	22.3	25.0	26.9	29.9	38.5	41.3	42.8	48.8	–	–
B3	22.6	25.3	27.4	30.4	38.9	41.6	44.0	49.7	50.3	54.9
B4	22.5	25.1	27.8	30.2	39.2	42.1	44.3	50.5	50.0	54.8
BC	21.2	23.7	26.0	29.3	38.7	41.8	43.9	49.6	49.5	53.9
M1	24.6	27.6	28.1	32.1	41.6	45.0	–	–	–	–
M2	24.5	27.4	27.9	31.6	41.5	44.9	49.3	55.1	53.8	59.2
M3	24.4	27.3	28.1	32.2	42.1	45.5	49.8	56.0	54.1	59.9
M4	24.8	28.1	29.1	32.3	41.7	44.5	48.8	54.5	54.3	59.6
MC	23.6	26.9	28.1	30.3	39.6	42.6	46.7	52.1	52.6	57.2
T1	28.5	33.8	33.3	36.2	44.4	47.9	52.8	57.6	57.0	61.7
T2	28.4	34.0	32.6	35.6	43.6	47.2	52.0	57.0	56.8	61.4
T3	27.8	32.2	32.4	35.7	43.9	47.7	51.8	56.3	56.5	61.2
T4	27.8	31.7	32.9	36.9	44.9	48.5	52.5	57.0	56.3	60.9
TC	24.2	26.8	29.7	33.1	42.1	45.8	49.0	53.6	53.0	57.9
AIR	14.5	17.0	17.8	24.4	24.4	31.7	25.1	31.2	37.5	43.1

Table 5

Test duration and apple sample masses of the outdoor drying tests.

	Date	Duration (h:min)	Initial mass (g)	Final mass (g)	Mass reduction (%)	DR_{av} (g g ⁻¹ h ⁻¹)
Test 1	14 May 2025	4:00	93.0	39.3	57.7	0.14
Test 2	15 May 2025	5:20	39.0	15.6	60.0	0.11
Test 3	3 June 2025	5:00	137.6	62	54.9	0.11
Test 4	4 June 2025	5:20	61.6	18.7	69.7	0.13

is reproducible. However, outdoor operation remains affected by unavoidable variations in ambient temperature, solar irradiance, and other environmental conditions.

These results also highlight the important role of the reflector in enhancing the thermal performance of the solar dryer. In particular, the reflector becomes especially beneficial under moderate solar irradiance and/or low ambient temperature, extending the operating effectiveness of the dryer and improving its applicability thanks to the additional solar radiation contribution. Furthermore, from a practical perspective, the proposed L-shaped reflector represents a simple, low-cost, lightweight, and easily transportable solution. The reflector was manufactured using cardboard covered with a Mylar reflective film, thus ensuring safe handling and practical implementation without the need for rigid metallic reflecting structures. Moreover, during the outdoor tests, the reflector was kept fixed and only oriented southward, without any active solar tracking system, thus demonstrating the ease of use of the proposed configuration.

5. Numerical analysis

This section presents a detailed description of the numerical model adopted for the CFD analysis of the solar dryer. The numerical model is developed with reference to the indoor no-load thermal test described in Section 4.1, which provides the boundary conditions and temperature measurements used for validation. The main modelling assumptions and the numerical setup are first introduced, followed by a discussion of the results obtained, with particular emphasis on the predictive capability of the model as assessed through comparison with experimental data. Once validated against the experimental configuration employing a painted steel absorbent surface, the CFD model is subsequently used as a design tool to investigate alternative configurations. In particular, the validated model is employed to analyze the effect of replacing the steel absorbent

surface with aluminium, in order to assess its impact on the thermal and fluid-dynamic behaviour of the dryer.

The CFD analysis is therefore used both for validation purposes and to gain insights into the key physical mechanisms governing the dryer performance under different design choices.

5.1. CFD modelling

The simulations are performed using the commercial CFD software ANSYS Fluent, which employs the finite volume method (FVM) to discretize the governing equations of fluid flow and heat transfer. In their general form, these equations correspond to the Navier–Stokes equations for a viscous compressible fluid and express the conservation of mass, momentum, and energy. Using Einstein summation notation, they can be written as:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j) = 0,$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho g_i, \quad i, j = 1, \dots, d, \quad (5)$$

$$\frac{\partial}{\partial t}(\rho E) + \frac{\partial}{\partial x_j} [u_j(\rho E + p)] = -\frac{\partial q_j}{\partial x_j} + \frac{\partial}{\partial x_j}(u_i \tau_{ij}),$$

where ρ denotes the air density, p the pressure, E the total energy per unit mass, u_i the i th velocity component, and g_i the i th component of the gravitational acceleration vector in a system with d spatial dimensions. In Eq. (5), τ_{ij} and q_j represent the viscous stress tensor and the heat flux vector, respectively, and are defined as:

$$\tau_{ij} = \mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \frac{\partial u_k}{\partial x_k} \delta_{ij} \right), \quad (6)$$

$$q_j = -\frac{\mu}{Pr} \frac{\partial}{\partial x_j} \left(E + \frac{p}{\rho} - \frac{1}{2} u_k u_k \right), \quad k = 1, \dots, d,$$

where μ is the dynamic viscosity, Pr is the Prandtl number, and δ_{ij} denotes the Kronecker delta. Finally, assuming an ideal gas, the equation of state is given by:

$$p = (\gamma - 1) \rho \left(E - \frac{1}{2} u_i u_i \right), \quad (7)$$

where $\gamma = c_p/c_v$ is the ratio of specific heats.

In the present study, the flow is assumed to be steady-state and incompressible. The steady-state assumption is justified by the fact that the

Table 6

Average ($T_{av,tot}$) and maximum (T_{max}) temperatures recorded during the outdoor apple drying tests. The temperature sensor locations are shown in Fig. 2. AIR corresponds to the inlet air temperature.

	14 May 2025 11:50–15:50 $T_{amb,av} = 24.1$ °C $G_{bn,av} = 874.7$ W m ⁻²		15 May 2025 10:40–16:00 $T_{amb,av} = 27.3$ °C $G_{bn,av} = 875.0$ W m ⁻²		3 June 2025 10:30–15:30 $T_{amb,av} = 32.5$ °C $G_{bn,av} = 776.7$ W m ⁻²		4 June 2025 9:50–15:10 $T_{amb,av} = 31.5$ °C $G_{bn,av} = 722.4$ W m ⁻²	
Reflector	yes		yes		yes		yes	
Position	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)	$T_{av,tot}$ (°C)	T_{max} (°C)
B1	35.3	37.9	39.8	43.7	45.0	49.9	55.9	60.8
B3	36.0	38.7	39.6	44.0	46.2	53.2	55.7	57.4
B4	37.9	41.2	41.1	46.2	46.6	53.1	54.9	56.3
BC	39.1	41.9	42.9	49.8	49.2	55.2	54.4	56.1
M2	38.9	41.2	45.0	49.8	45.4	51.2	56.4	58.8
M3	35.8	38.5	42.8	46.7	49.6	56.3	58.4	60.3
M4	33.4	35.9	41.1	45.0	45.0	51.3	57.0	58.9
MC	37.8	40.5	41.5	46.0	47.5	53.7	56.1	57.7
T1	44.5	49.0	46.4	50.8	52.6	60.1	59.4	61.6
T2	45.0	49.8	46.6	50.5	52.3	58.7	59.5	61.8
T3	44.0	49.6	47.2	51.9	51.7	60.4	60.1	62.6
T4	44.7	49.8	47.6	52.1	52.8	60.5	60.3	62.6
TC	39.8	42.2	44.3	49.3	49.7	54.8	57.8	59.6
AIR	23.0	31.9	25.4	33.0	41.5	48.9	35.0	36.0

analysis focuses on quasi-stationary operating conditions reached after the initial thermal transient. Moreover, the CFD model was validated against indoor experiments performed under controlled irradiation conditions using the solar simulator, which provided nearly steady thermal operating conditions during the validation tests. The incompressible assumption is justified by the relatively small variations in air density occurring during the drying process, as the temperature differences, pressure levels, and flow velocities typically encountered in solar dryers are not sufficiently high to require a fully compressible formulation. Adopting the incompressible flow assumption significantly simplifies the Navier–Stokes equations, reducing computational complexity without compromising the accuracy of the numerical results. Despite the incompressible assumption, buoyancy effects induced by temperature gradients play a key role in driving natural convection inside the drying chamber. For this reason, the Boussinesq approximation is used to account for density variations, which remain relatively limited under the operating conditions considered in this study, with an estimated relative variation of only a few percent. This approach has been widely employed in CFD studies of buoyancy-driven natural convection flows [34,35] and solar drying systems [25,31]. Under this approximation, density variations are neglected in all governing equations except in the buoyancy term of the momentum equations where they are assumed to vary linearly with temperature. The air density is therefore expressed as:

$$\rho = \rho_{\text{ref}} [1 - \beta (T - T_{\text{ref}})], \quad (8)$$

where β is the thermal expansion coefficient, T is the temperature, ρ_{ref} and T_{ref} denote the reference air density and temperature, respectively, defined in this work as representative volume-averaged values of the air within the drying chamber. Note that, under the Boussinesq approximation, the density variation defined above is introduced exclusively in the buoyancy term of the momentum equations, while density is treated as constant and equal to ρ_{ref} in the mass and energy conservation equations, ensuring consistency with the incompressible formulation.

Under the operating conditions considered in this study, the flow inside the solar dryer is driven by natural convection and is characterized by relatively low velocities and moderate temperature differences, as typically observed in passive solar drying systems. An order-of-magnitude estimate of the Rayleigh number, based on the characteristic length of the dryer and on the temperature differences observed during operation, yields values of the order of 10^7 – 10^8 , indicating a buoyancy-driven regime consistent with natural-convection-driven ventilation. At the same time, the estimated Rayleigh number remains within the range typically associated with predominantly laminar or weakly transitional natural convection in enclosed or semi-enclosed cavities.

Therefore, the incompressible Navier–Stokes equations coupled with the Boussinesq approximation and a steady laminar flow model provide an adequate description of the dominant transport mechanisms in the considered configuration. Finally, in the energy equation viscous heating was neglected, as its contribution is negligible for the low-velocity buoyancy-driven airflow typical of solar drying systems.

To account for solar radiation, the solar load model available in ANSYS Fluent is employed. This model allows the definition of the Sun position and the direct normal solar irradiation at the Earth surface based on user-specified inputs, including geographic location, date, time, and a sunshine factor. The sunshine factor is a dimensionless parameter ranging between 0 and 1, used to adjust the intensity of solar radiation, where a value of 1 represents full sunlight under clear-sky conditions, while lower values account for attenuation effects due to clouds, obstacles, or atmospheric pollution. In this work, the Sun is intentionally positioned at the zenith in order to reproduce the irradiation conditions generated by the solar simulator used in the experimental campaign. This configuration is obtained by setting the latitude and longitude to approximately 0° and 28° , respectively, and by selecting September 23 at 12:00 p.m. (GMT + 2), corresponding to the equinox, when the Sun

Table 7

Thermo-physical properties of air and steel adopted in the CFD simulations.

Material	Property	Value
Air	Density ρ_{ref}	1.057 kg m^{-3}
	Dynamic viscosity μ	$1.997 \times 10^{-5} \text{ kg m}^{-1} \text{ s}^{-1}$
	Thermal conductivity k	$2.72 \times 10^{-2} \text{ W m}^{-1} \text{ K}^{-1}$
	Thermal expansion coefficient β	$3.09 \times 10^{-3} \text{ K}^{-1}$
	Specific heat c_p	$1005 \text{ J kg}^{-1} \text{ K}^{-1}$
Steel shell	Density ρ	8030 kg m^{-3}
	Specific heat c_p	$502.48 \text{ J kg}^{-1} \text{ K}^{-1}$
	Thermal conductivity k	$16.27 \text{ W m}^{-1} \text{ K}^{-1}$

is located at the zenith at latitude 0° . This approach ensures consistency between the numerical model and the indoor experimental setup, where the incident radiation is applied in the zenith direction, thereby enabling a meaningful comparison between numerical predictions and experimental data for validation purposes. Under these conditions, the ambient air temperature can realistically reach values around 32.3°C , which is consistent with the temperature recorded in the experimental tests. Accordingly, the sunshine factor was set to 0.93 in order to match the direct normal solar irradiance of 900 W m^{-2} applied during the indoor experimental campaign.

Radiative heat transfer within the computational domain is modelled using the surface-to-surface (S2S) radiation model, based on the enclosure radiation theory and radiosity formulation extensively described by Howell et al. [36]. This model provides an effective compromise between accuracy and computational efficiency and is well suited for enclosures characterized by gray and diffuse surfaces, as in the present configuration. The thermo-physical properties of air are evaluated at a reference temperature of $T_{\text{ref}} = 50^\circ\text{C}$ and at the measured ambient pressure of 98,236 Pa. The reference temperature corresponds to a representative average value of the air temperature inside the solar dryer, computed as a volume-weighted average of the experimental measurements [37].

The absorbent surface of the prototype, made of painted steel, is modelled as a shell conduction zone using four layers, with a total thickness equal to that of the real prototype (0.7 mm). The thermo-physical properties adopted for air and steel are reported in Table 7. For the steel material, the properties were taken from the ANSYS Fluent materials database, since the exact alloy composition of the prototype was not specified. Nevertheless, the most commonly used stainless steels for food-related applications, such as AISI 304 and AISI 316, exhibit thermo-physical properties very similar to those reported in the table. In particular, AISI 304 stainless steel is widely employed in the food industry due to its good corrosion resistance and workability, and its properties closely match those adopted in the simulations.

Natural ventilation through the openings is modelled by prescribing pressure boundary conditions at both the inlet and outlet sections. A zero gauge pressure is imposed at the openings, while the operating pressure is set equal to the experimentally measured ambient pressure. The ambient air temperature measured during the experiments (32.3°C) is prescribed at the openings for inflow and possible backflow at outflow, ensuring consistency with buoyancy-driven flow conditions.

The spatial discretization and numerical solution schemes adopted in the simulations are summarized as follows:

- Pressure–velocity coupling is handled using the SIMPLE segregated algorithm;
- Pressure interpolation is performed using the PRESTO! scheme;
- The momentum and energy equations are discretized using second-order upwind scheme;
- Convergence is assessed by monitoring both the scaled residuals and the temperature evolution at the same monitoring points used in the experimental campaign;
- Iterations are continued until all residuals decrease by at least eight orders of magnitude and the monitored temperature values reach stable plateau conditions.

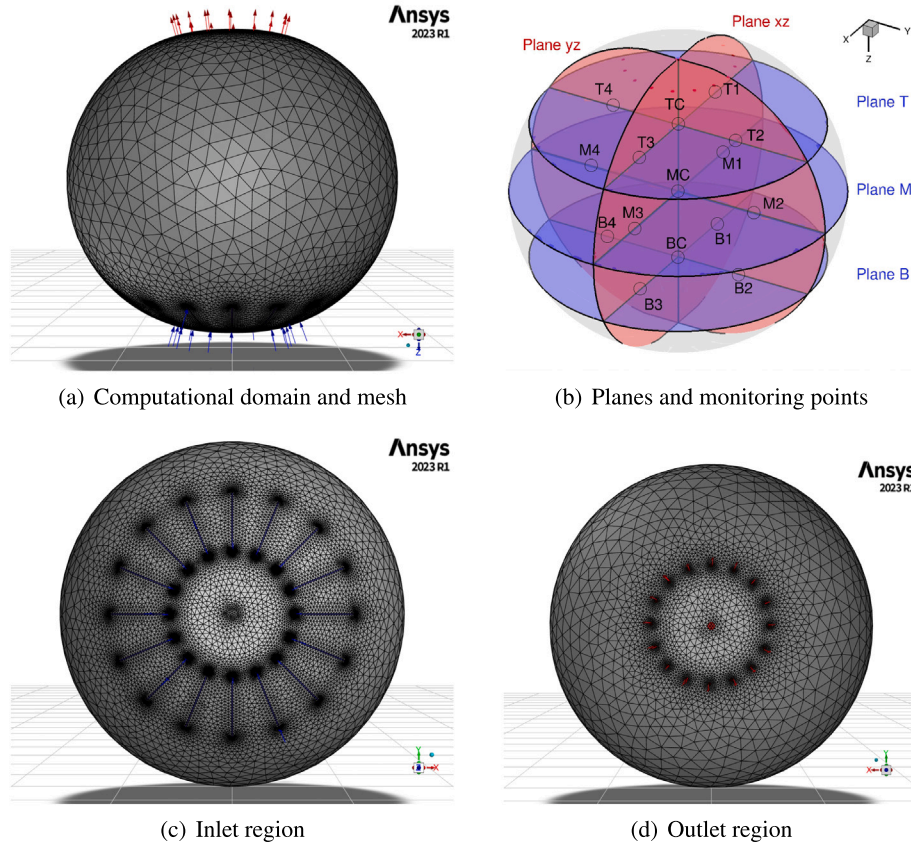


Fig. 8. Computational domain, mesh, and monitoring points used for the CFD simulations.

Table 8

Mesh Independence study: absolute relative deviations in temperature ($ARD(T)$) between meshes at the experimental monitoring points.

M1 (254,000 elements) vs. M2 (503,000 elements)															
Point	BC	B1	B2	B3	B4	MC	M1	M2	M3	M4	TC	T1	T2	T3	T4
$ARD(T)[\%]$	5.7	5.9	6.1	5.7	5.9	3.5	3.5	3.5	3.7	3.1	3.4	3.6	3.0	2.3	3.4
M2 (503,000 elements) vs. M3 (981,000 elements)															
Point	BC	B1	B2	B3	B4	MC	M1	M2	M3	M4	TC	T1	T2	T3	T4
$ARD(T)[\%]$	0.4	0.2	0.2	0.0	0.2	0.8	0.8	0.8	0.8	0.8	1.9	1.7	1.5	1.1	1.8

The mesh used for the simulations, labelled M2 in the following, is shown in Fig. 8, together with a 3D schematic representation of the planes and points adopted for the CFD analysis. The monitoring points correspond to those used in the experimental measurements (see Fig. 2). The computational grid is a conformal hybrid mesh composed of tetrahedral and wedge elements. The aspect ratio ranges from 1.2 to 55.3, with an average value of 3.5, while the skewness varies between 5.6×10^{-4} and 0.91, with an average value of 0.2.

A mesh independence study was carried out in order to assess the sensitivity of the numerical results to grid refinement and to identify a suitable compromise between accuracy and computational cost. Three different meshes were considered, namely M1, M2, and M3, characterized by a progressively increasing level of spatial refinement. The refinement from M1 to M3 was obtained by doubling the number of elements along the edges defining the openings, halving the first cell height of the boundary layer mesh (composed of ten layers), and reducing the overall element size. Temperature values at the experimental monitoring points were extracted for all three meshes and compared in order to quantify grid sensitivity. Table 8 reports the absolute relative deviations in temperature ($ARD(T)$) between two consecutive meshes

computed as:

$$ARD(T)[\%] = \frac{|T_{\text{fine}} - T_{\text{coarse}}|}{T_{\text{fine}}} \cdot 100 \quad (9)$$

where T_{fine} and T_{coarse} denote the temperatures obtained using the finer and coarser meshes, respectively. The comparison highlights a significant sensitivity to grid refinement for the coarsest mesh (M1), whereas a much better agreement is observed between meshes M2 and M3. In particular, the maximum and average temperature variations decrease from 6.1% and 4.2% (M1–M2) to 1.9% and 0.9% (M2–M3), respectively, indicating that grid-independent results are essentially achieved for mesh M2. Moreover, the largest discrepancies are observed on the top plane, which is not directly involved in the drying process, while very good agreement is found on the middle plane, corresponding to the actual drying region. Based on these results, mesh M2 is selected for all subsequent simulations.

5.2. Validation and analysis of CFD results

To assess the accuracy of the numerical model, Table 9 reports the predicted temperatures (T_{num}) and experimental temperatures (T_{exp}) at

Table 9
Comparison between numerical and experimental temperatures at the monitoring points used for CFD validation.

Numerical vs. Experimental temperatures															
Point	BC	B1	B2	B3	B4	MC	M1	M2	M3	M4	TC	T1	T2	T3	T4
T_{exp} [°C]	46.2	47.2	47.8	47.8	47.3	48.5	50.1	49.6	50.3	50.8	53.3	55.6	55.3	56.6	56.1
T_{num} [°C]	49.1	49.0	48.9	49.1	49.0	51.2	51.2	51.1	51.2	51.2	52.9	53.1	53.1	53.3	53.1
$ARD(T)$ [%]	6.3	3.9	2.4	2.8	3.6	5.5	2.2	3.0	1.8	0.9	0.8	4.4	3.9	4.1	5.4

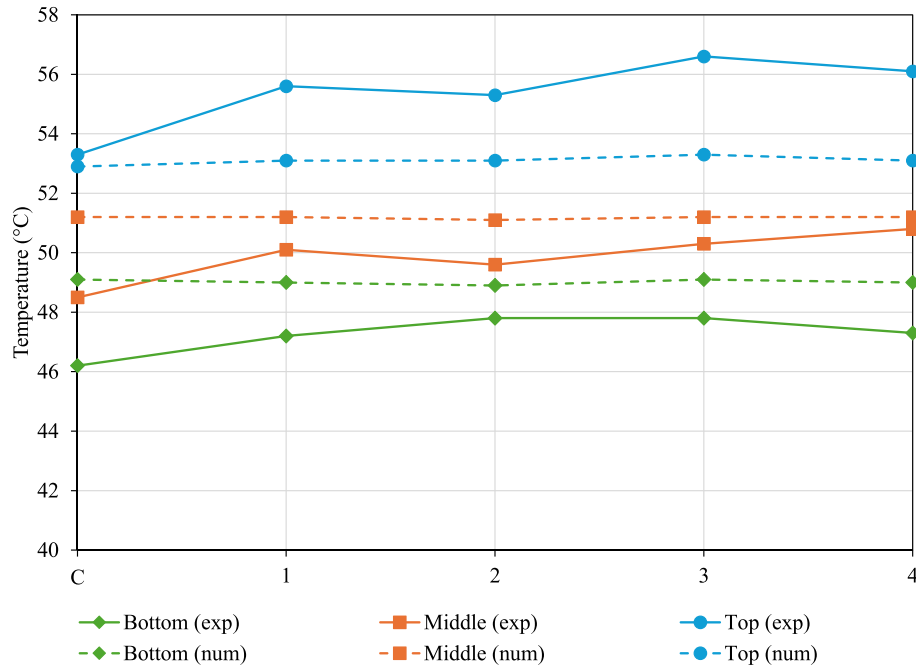


Fig. 9. Numerical and experimental temperatures at the monitoring points used for CFD validation. The solid and dashed lines correspond to the experimental and numerical data, respectively.

the monitoring points, together with the absolute relative deviation between numerical and experimental temperature values, computed as:

$$ARD(T) [\%] = \frac{|T_{num} - T_{exp}|}{T_{exp}} \cdot 100. \quad (10)$$

The comparison indicates that the CFD predictions are in good agreement with the experimental measurements. The mean absolute relative deviation is equal to 3.4%, corresponding to a mean absolute error of approximately 1.7 °C, which is comparable to the measurement uncertainty of the thermocouples employed in the experimental campaign (± 1 °C). This level of agreement confirms the capability of the numerical model to reliably reproduce the thermal behaviour of the solar dryer under the tested operating conditions. The largest discrepancies are observed at the central monitoring points BC and MC, where the absolute percentage differences reach 6.3% and 5.5%, respectively. These deviations can be attributed to the presence of the vertical steel rod used to position the thermocouples and to support the circular mesh tray used for food placement. As described in Section 2, the central rod has a diameter of 10 mm, which is significantly larger than that of the lateral supporting rods (5 mm) arranged perpendicularly. Due to its larger size, the central rod locally alters the heat transfer mechanisms and absorbs part of the thermal energy, leading to experimentally measured temperatures that are slightly lower than those predicted by the numerical model.

The numerical results reported in Table 9 and graphically presented in Fig. 9 indicate, consistently with the experimental observations, that the temperature within the drying chamber falls within the typical

range required for drying fruits and vegetables, generally between 40 and 70 °C. Under the considered operating conditions, this confirms that the entire internal volume of the proposed dryer is suitable for drying applications. Furthermore, the results show that, at each horizontal plane, the temperature distribution is relatively uniform, which is a desirable feature to ensure a homogeneous drying process for food placed at the same vertical level. In this regard, Fig. 9 highlights that temperature uniformity appears more pronounced in the CFD prediction than in the corresponding experimental measurements. The slightly larger fluctuations observed in the experimental data can reasonably be attributed to experimental uncertainties, including thermocouple positioning and local temperature variability under real-world operating conditions. Conversely, a clear temperature gradient is observed along the vertical (z) direction, as highlighted in the top plots of Fig. 10. Such a non-uniform temperature distribution along the height of the dryer is generally undesirable, since food located at different horizontal planes may experience different drying rates. It should be noted that the following considerations strictly apply to the configuration analysed in this study, corresponding to zenithal or high solar elevation irradiation conditions. For generic positions of the Sun, additional temperature non-uniformities may arise also within the horizontal planes. Nevertheless, the nearly spherical geometry of the dryer tends to promote a smoother distribution of incident solar radiation compared to flat surfaces. This behaviour is related to the continuous variation of the local surface normal over the curved geometry, which leads to a gradual change in the angle of incidence of solar radiation. As a result, highly localized irradiated regions are mitigated, contributing to a more spatially distributed thermal input to the structure.

Table 10

Thermo-physical properties of aluminium adopted in the CFD simulations.

Material	Property	Value
Aluminium shell	Density ρ	2719 kg m ⁻³
	Specific heat c_p	871 J kg ⁻¹ K ⁻¹
	Thermal conductivity k	202.4 W m ⁻¹ K ⁻¹

In this context, the presence of a fixed reflector can further enhance both the magnitude and the spatial distribution of the incident solar radiation. In general, such a reflector is oriented towards the equator (south in the northern hemisphere and north in the southern hemisphere). Accordingly, in the experimental campaign described in Section 3, the reflector was oriented towards the south, consistent with the geographical location of Ancona (Italy). However, due to the continuous variation of solar altitude and azimuth, a fixed reflective surface cannot ensure a uniform redistribution of solar radiation under all operating conditions. Although mirrors can increase the overall thermal input, their effect on the spatial distribution of absorbed energy remains strongly dependent on solar position, which continuously varies throughout the day and across seasons. In contrast, passive thermal design strategies, such as the adoption of highly conductive absorbent materials and an optimized geometry, act through intrinsic mechanisms. While a reflector modifies the incoming radiation field, high-conductivity materials promote internal heat spreading, resulting in a more stable reduction of temperature gradients. Compared to reflective solutions, this approach results in a simpler, more robust, and low-maintenance system, which is particularly advantageous for decentralized and off-grid drying applications. Therefore, although fixed mirrors may represent an optional enhancement under specific conditions, the present study prioritizes intrinsic

design solutions that improve thermal performance while preserving system simplicity.

For this reasons, the validated CFD model is employed as a design tool to investigate the influence of the absorbent surface material on the thermal behaviour of the dryer. In particular, aluminium is considered as an alternative material for the absorbent surface, with thermo-physical properties, adopted in the simulations, reported in Table 10.

As shown by the CFD results in Fig. 10, the aluminium configuration leads to a markedly reduced vertical thermal gradient, with a temperature difference between the top and bottom planes not exceeding 2 °C. In contrast, the corresponding CFD simulation performed with a steel absorbent surface (see also Table 9) results in a temperature difference of about 4 °C, a value that is even larger when considering the experimental measurements.

However, it should be noted that, while the use of aluminium significantly improves the uniformity of the temperature field inside the drying chamber, it also affects the buoyancy-driven airflow responsible for natural ventilation, which in turn results in reduced airflow velocities through the openings. This behaviour can be consistently analysed within the framework of the Boussinesq approximation, in which the buoyancy term in the momentum equations is proportional to $g \beta \Delta T$. The CFD model therefore predicts a less intense natural ventilation, with the mass flow rate through the openings decreasing by about 14% compared with the configuration employing the steel absorbent surface. From a drying perspective, this highlights a trade-off between improved thermal uniformity and reduced convective air renewal. The actual impact on drying rate and final product moisture content cannot be inferred from the present CFD model alone, since it does not include coupled heat and mass transfer within the product. For this reason, the drying performance of the aluminium configuration requires dedicated load tests, which are currently under investigation.

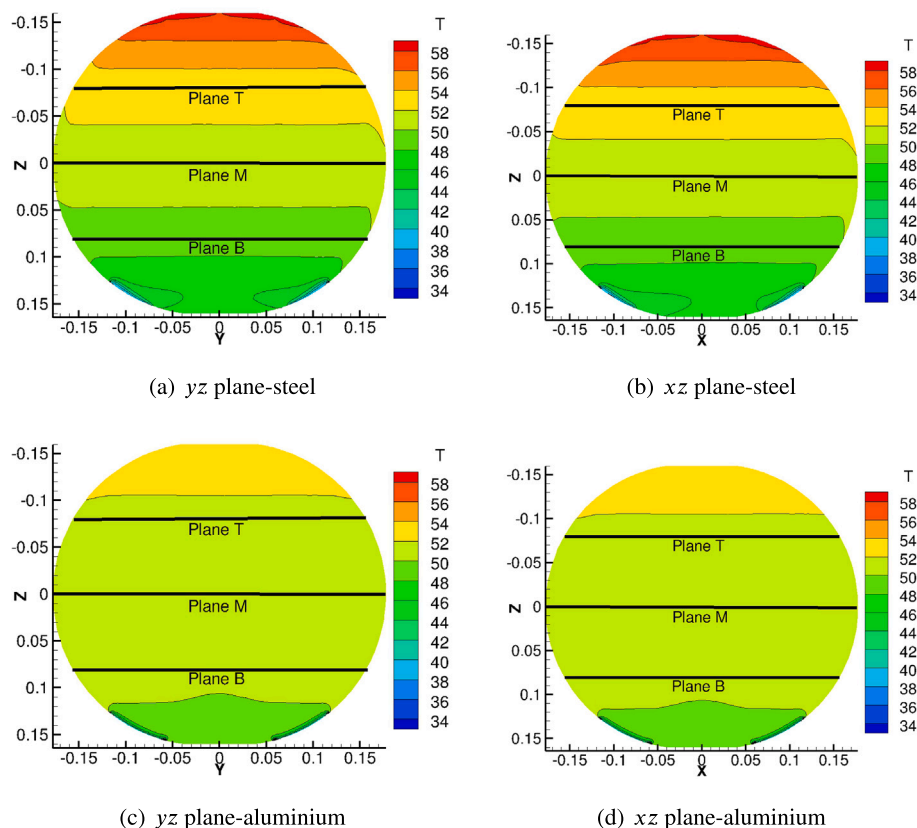


Fig. 10. Contours of temperature distribution along the yz and xz planes (see Fig. 8(b)) with steel (top row) and aluminium (bottom row).

6. Conclusion

A compact nearly spherical solar dryer was experimentally and numerically characterized, enabling assessment of its thermal behaviour under both controlled and real operating conditions. The experimental campaign demonstrated that the prototype achieves temperature levels suitable for fruit drying, while indoor no-load thermal tests enabled detailed characterization of the internal temperature field and provided reference data for CFD validation. In particular, the mass reduction and drying rate obtained during the drying tests indicated the prototype's capability to dry apples both under indoor and outdoor conditions. However, to more accurately characterize the performance of the proposed solar dryer, future studies will focus on drying kinetics and product quality. The developed three-dimensional CFD model, accounting for buoyancy-driven airflow, conjugate heat transfer, and radiative effects, showed good agreement with the measurements, with average deviations comparable to the experimental uncertainty. The validated model was then employed as a design tool to investigate the influence of material properties on the internal thermal field. The results indicate that the nearly spherical geometry contributes to a smooth spatial distribution of absorbed solar energy, while the use of highly conductive absorbent materials significantly improves vertical temperature uniformity. At the same time, the reduction in temperature gradients leads to weaker buoyancy-driven ventilation, highlighting a trade-off between thermal homogeneity and natural airflow intensity in passive solar dryers. These findings provide quantitative insight into the thermofluid behaviour of compact naturally ventilated solar dryers and support informed design choices based on geometry and material selection. Ongoing work aims to extend the analysis to loaded conditions in order to assess the influence of the absorbent material on drying performance.

CRedit authorship contribution statement

Alessandra Nigro: Writing – review & editing, Writing – original draft, Visualization, Validation, Software, Methodology, Data curation, Conceptualization. **Sebastiano Tomassetti:** Writing – review & editing, Writing – original draft, Validation, Investigation, Data curation. **Tariku Negash Demissie:** Visualization, Investigation. **Giovanni Di Nicola:** Writing – review & editing, Supervision, Project administration.

Declaration of Generative AI and AI-assisted technologies in the writing process

During the preparation of this work, the authors used ChatGPT (OpenAI) to improve the readability of the manuscript and to refine the English language. After using this tool, the authors carefully reviewed and edited the content as needed and take full responsibility for the content of the published article.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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