



Computational Fluid Dynamics-Based Characterization of Heat Transfer and Airflow in an Indirect Solar Fish Dryer Under Tropical Conditions

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Abstract: Indirect solar drying is a widely adopted fish preservation technique in tropical regions, valued for its low energy requirement and capacity to enhance product quality. However, non-uniform airflow and temperature distribution within drying chambers remain persistent design challenges that compromise drying efficiency and product consistency. This study investigated the airflow characteristics and convective heat transfer performance of an indirect solar fish dryer through computational fluid dynamics (CFD) simulation and experimental validation. A three-dimensional CFD model was developed to simulate temperature distribution and airflow patterns under forced convection, with an inlet air velocity of $1.2 \text{ m}\cdot\text{s}^{-1}$ and an inlet air temperature of 60°C . Experimental air temperatures were recorded inside the drying chamber during actual drying operations and compared with simulated values using linear regression and root-mean-square error (RMSE). The regression analysis yielded a strong linear relationship between simulated and experimental temperatures ($R^2 = 0.9413$), with an RMSE of 3.41°C , indicating reasonable agreement in absolute temperature prediction. These results confirm that the CFD model accurately represents the thermal behavior and convective heat transfer characteristics of the dryer. The validated CFD framework provides a reliable, cost-effective tool for evaluating and optimizing the design and operational performance of indirect solar fish dryers under tropical conditions, thereby reducing reliance on iterative physical prototyping.

Keywords: Computational fluid dynamics; convective heat transfer; experimental validation; fish drying; indirect solar dryer

1. Introduction

The use of computational fluid dynamics (CFD) in the design and optimization of indirect solar dryers has become increasingly significant for enhancing the efficiency and reliability of drying processes for fish and other agricultural products. CFD allows for a detailed examination of airflow patterns, temperature distribution, and convective heat transfer within drying systems, processes that are often challenging to assess using experimental methods alone. When combined with experimental validation, CFD provides a robust framework for understanding thermal–fluid interactions and for enhancing drying uniformity and product quality. Indirect solar dryers are designed to

separate solar radiation collection from the drying chamber, thereby shielding products from direct sunlight exposure. This configuration reduces the risk of contamination and deterioration in quality while providing improved control over key drying parameters, such as airflow rate and temperature. In these systems, solar energy heats the air, which is then circulated throughout the drying chamber, where convective heat transfer drives moisture removal. Previous studies have emphasized that optimizing airflow and temperature distribution within indirect solar dryers is critical to achieving uniform drying and maintaining product integrity [1–4]. Solar drying has also been shown to yield product quality comparable to conventional electrical drying for a range of agricultural commodities, including silk cocoons [4].

Several researchers have demonstrated the effectiveness of CFD in analyzing the thermal and fluid behavior of indirect solar dryers. Noh et al. [5] developed a CFD model that integrated airflow, temperature, and moisture interactions to evaluate solar dryer performance, highlighting the strong influence of airflow distribution on drying behavior. Carrera et al. [6] used CFD to study heat distribution in an indirect solar chili dryer and found that airflow behavior plays a crucial role in achieving uniform temperature in the drying chamber. Similarly, Villagrán et al. [7] showed that optimized airflow patterns in greenhouse solar dryers improved the drying performance of aromatic plants. Similarly, Srirugsa et al. [8] demonstrated that ventilated solar technology significantly improved drying efficiency for red chili peppers by enhancing forced airflow within the drying chamber. Despite the strengths of numerical modeling, experimental validation remains essential to ensure that CFD predictions accurately represent real drying conditions. Gunathilaka et al. [9] validated a CFD model of an indirect solar dryer using experimental temperature and air velocity measurements and reported close agreement between simulated and measured values. Mellalou et al. [10] likewise observed a strong correlation between experimental and simulated temperature profiles in a modified greenhouse solar dryer, supporting the reliability of CFD techniques for dryer performance evaluation.

Although earlier studies have established the usefulness of CFD for analyzing solar drying systems, relatively few have combined CFD with experimental methods to evaluate convective heat transfer in indirect solar fish dryers operating under tropical conditions. Therefore, this study seeks to develop and validate a three-dimensional CFD model of an indirect solar fish dryer by comparing simulated airflow patterns and temperature distributions with experimentally measured temperatures inside the drying chamber. The findings are expected to contribute to understanding convective heat transfer mechanisms and to support the optimization of the design and operation of indirect solar dryers for fish drying applications.

2. Materials and Methods

2.1 Description of the indirect solar dryer

The indirect solar dryer examined in this study comprises a solar air heater coupled to a vertical drying chamber fitted with multiple perforated trays (Figure 1). The system is designed to separate solar heat collection from the drying chamber, preventing direct exposure of fish samples to solar radiation while allowing controlled convective heat transfer. The solar air heater is installed at a lower elevation and inclined to enhance solar energy absorption. It is enclosed with a transparent glass cover that permits solar radiation to enter the collector while minimizing convective heat losses. Solar energy transmitted through the glass cover warms the air inside the collector, which is then directed to the drying chamber via a connecting duct. The heated air enters the drying chamber through a lower inlet positioned at the base of the chamber. The drying chamber is a vertical enclosure supported by a rigid frame and fitted with multiple perforated horizontal trays arranged at uniform spacing. The trays and internal structural components are made of aluminum to prevent rust and corrosion, ensuring durability and suitability for food-drying applications. Fish samples are placed on the trays during operation. The perforations in the trays allow heated air to pass uniformly across and through the product layers, thereby enhancing convective heat transfer and promoting uniform moisture removal. As the heated air enters the drying chamber through the lower inlet, it rises and flows across the trays under forced convection, transferring heat to the fish samples. Moist air exits the chamber through an upper vent located beneath the roof. This airflow configuration establishes a continuous upward flow path that supports stable drying conditions and a relatively uniform temperature distribution within the chamber.

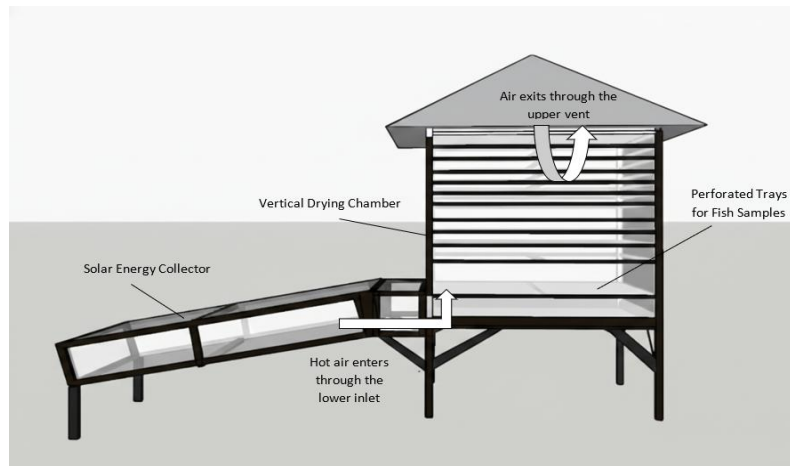


Figure 1. Schematic diagram of the indirect solar dryer.

2.2 CFD geometry and computational domain

The computational geometry of the indirect solar dryer was created using the actual dimensions of the fabricated system, as illustrated in the front and left-side elevation views in Figure 2. The geometry represents the solar air heater, connecting duct, and vertical drying chamber as a single continuous airflow domain for numerical simulation. The vertical drying chamber has an overall height of approximately 1.99 m, measured from the base support to the topmost section. The chamber has a width of 0.90 m, as indicated in the front elevation, and contains multiple perforated aluminum trays arranged at uniform vertical spacing of approximately 0.14 m. A lower clearance of approximately 0.80 m is provided beneath the chamber to accommodate structural supports and airflow passage. The drying chamber is surmounted by a pitched roof with an effective roof length of approximately 0.90 m and a roof height of 0.52 m. This configuration protects from environmental exposure and aids in the discharge of moist air through the upper vent. The exhaust opening is located directly beneath the roof, establishing an upward airflow path through the chamber.

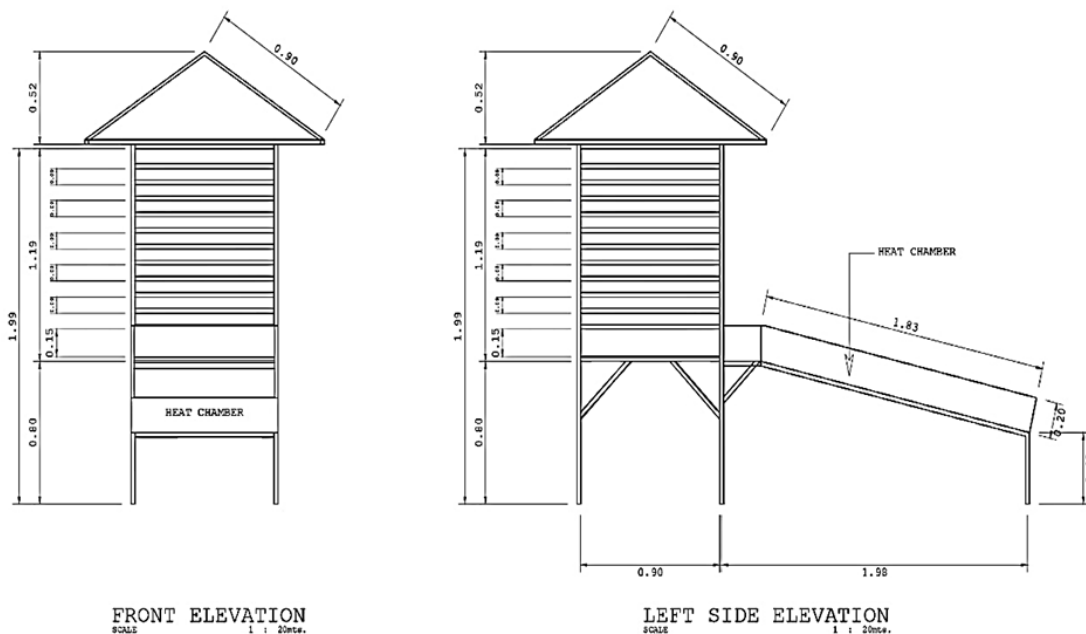


Figure 2. Technical elevations of the solar drying system.

Airflow from the solar air heater enters the drying chamber through a horizontally oriented inlet located near the chamber base. The left-side elevation shows that the heater is mounted at an incline, with an

effective collector length of approximately 1.83 m and a horizontal projection of about 1.98 m. The air gap height within the collector varies along its length, with an approximate height of 0.27 m near the inlet and 0.40 m near the downstream end, following the inclined geometry.

2.3 Tray configuration and computational representation

Each drying tray has a square plan dimension of 0.90 m $\times$ 0.90 m and is constructed using a fine aluminum wire mesh, as shown in Figure 3. The wire mesh provides structural support for the fish samples while allowing heated air to pass through the tray surface, thereby enhancing convective heat transfer and promoting uniform airflow distribution within the drying chamber.

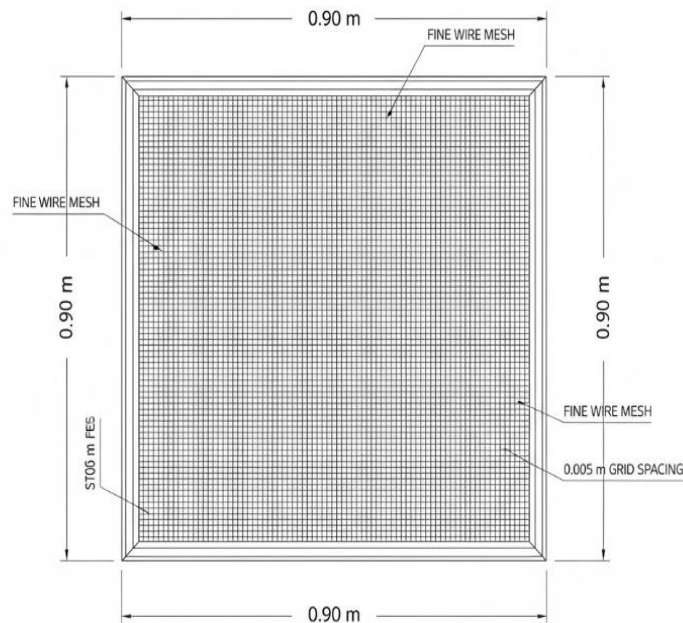


Figure 3. Technical specifications of the 0.9 m $\times$ 0.9 m drying tray mesh.

The mesh consists of a uniform grid with a spacing of approximately 0.005 m, which permits adequate airflow while supporting the product during drying. The use of aluminum minimizes corrosion and ensures suitability for food-drying applications. In the CFD model, the trays were represented as porous surfaces to account for airflow resistance while avoiding excessive mesh refinement that would significantly increase computational cost [11–13]. This approach preserves the overall flow behavior through the tray region while maintaining numerical efficiency. The porous resistance parameters were selected to reflect the fine-wire-mesh structure, allowing air to pass through the trays with minimal pressure drop. This simplified representation enables accurate simulation of airflow and temperature distribution across the tray levels without explicitly resolving individual wire elements, which would be computationally prohibitive. The use of porous materials in solar air collectors has also been shown to improve thermal efficiency under tropical conditions [13], further supporting the representational approach adopted here.

2.4 CFD model development

A three-dimensional CFD model of the indirect solar dryer was established to represent the airflow field and the system's thermal behavior. The geometric model was constructed based on the physical dimensions of the fabricated dryer, including the air inlet, solar air heater connection, vertical drying chamber, perforated trays, and air outlet. These components were treated as a single continuous airflow domain to capture the coupled effects of airflow and heat transfer processes throughout the system [14, 15]. The CFD model was established to simulate transient forced-convection airflow and temperature distributions within the drying chamber using ANSYS Fluent 2022 R2 with a pressure-based solver and the finite-volume method for spatial discretization. Air was assumed to behave as an incompressible fluid with constant thermophysical

properties evaluated at the operating temperature range. Given the inlet velocity of $1.2 \text{ m}\cdot\text{s}^{-1}$ and chamber width of 0.9 m , the approximate Reynolds number is $\sim 70,000$, indicating turbulent flow. Accordingly, the realizable k - ϵ turbulence closure model was employed, with standard wall functions applied at solid boundaries. The simulation was run as a transient (unsteady) model with a time step of 1 s and 20 inner iterations per time step to ensure convergence. Convergence was assessed using residual criteria of 10^{-4} for the continuity, momentum, and turbulence transport equations and 10^{-6} for the energy equation. Radiation heat transfer and moisture evaporation effects were not included in the present model [16,17]. The governing equations solved in the CFD simulation consist of the transient conservation equations for mass, momentum, and energy for incompressible flow, with time-derivative terms included to capture the unsteady thermal behavior of the drying chamber. These equations are expressed as follows:

Continuity equation:

$$\nabla \cdot v = 0 \quad (1)$$

Momentum equation:

$$\rho(v \cdot \nabla)v = -\nabla p + \mu \nabla^2 v \quad (2)$$

Energy equation:

$$\rho c_p(v \cdot \nabla T) = k \nabla^2 T \quad (3)$$

where v is the air velocity vector ($\text{m}\cdot\text{s}^{-1}$), p is pressure (Pa), T is temperature ($^{\circ}\text{C}$), ρ is air density ($\text{kg}\cdot\text{m}^{-3}$), μ is dynamic viscosity ($\text{Pa}\cdot\text{s}$), c_p is the specific heat capacity of air ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$), and k is the thermal conductivity of air ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$). The above equations were solved simultaneously to predict airflow patterns and temperature distribution inside the drying chamber. Heat transfer within the system was governed primarily by convection between the heated air and the tray surfaces. The perforated trays were represented as porous media to account for airflow resistance while maintaining numerical efficiency [18].

2.5 Boundary Conditions and Assumptions

The boundary conditions and modeling assumptions adopted in this study are summarized in Table 1 and were used to represent airflow and convective heat transfer within the indirect solar dryer. These parameters were defined using measured operating conditions obtained during experimental trials and were consistently imposed throughout the entire computational domain. Air enters the drying chamber through the lower inlet at a prescribed uniform velocity and temperature, representing the forced convection airflow generated by the solar air heater. The inlet air temperature was specified as 60°C , corresponding to the average heated air temperature observed during dryer operation. The inlet air velocity was fixed at $1.2 \text{ m}\cdot\text{s}^{-1}$, based on the measured airflow rate supplied to the drying chamber. A pressure-based boundary condition was applied at the outlet, with the gauge pressure specified as atmospheric to permit unrestricted air discharge from the drying chamber. All solid walls were treated as no-slip boundaries with adiabatic thermal conditions. Such simplifying assumptions are widely used in CFD analyses of indirect solar drying systems [19, 20].

2.6 Experimental validation

An experimental validation procedure was performed to evaluate the predictive capability of the CFD model in representing the thermal response of the indirect solar fish dryer. Temperature measurements were collected at multiple locations across four tray levels inside the drying chamber. At each tray level, sensors were positioned at two horizontal locations — one near the center and one near the lateral edge — to capture spatial temperature variation and to verify the predicted non-uniform temperature distribution. A total of 10 calibrated K-type thermocouples ($\pm 0.5^{\circ}\text{C}$ accuracy) were used, connected to a Graphtec GL840 multichannel data logger with a 60 s recording interval. Sensors were calibrated against a reference thermometer before deployment. Data were collected on 12–14 March 2024 between 08:00 and 16:00 local time under clear-sky conditions representative of the operating conditions imposed in the CFD simulation. The coefficient of determination (R^2) was used to quantify the degree of correlation between simulated and experimental

temperatures, while the root mean square error (RMSE) was calculated to quantify the average deviation between the two datasets. The RMSE was calculated using the following expression:

$$\text{RMSE} = \sqrt{1/n \cdot \sum (T_{\text{sim},i} - T_{\text{exp},i})^2} \quad (4)$$

where $T_{\text{sim},i}$ and $T_{\text{exp},i}$ denote the simulated and experimental temperatures at time step i , respectively, and n is the total number of data points considered. A scatter plot of simulated versus experimental temperatures was generated to visually assess model performance, and a linear regression line was fitted to the data [21].

Table 1. Boundary conditions, material properties, and modeling assumptions used in the CFD simulation.

Parameter	Value / Description
Inlet air temperature [°C]	60
Inlet air velocity [m/s]	1.2
Outlet condition	Pressure outlet (atmospheric pressure)
Initial air temperature [°C]	30
Flow regime	Transient (unsteady)
Flow type	Forced convection
Air density (ρ) [kg/m ³]	1.060 (at 60 °C operating temperature; corrected from standard sea-level value)
Dynamic viscosity of air (μ) [Pa·s]	1.97×10^{-5} (at 60 °C operating temperature; corrected from standard sea-level value)
Specific heat capacity (cp) [J/kg·K]	1006
Thermal conductivity of air (k) [W/m·K]	0.026
Tray material	Aluminum (corrosion-resistant)
Tray modeling approach	Porous media
Wall boundary condition	No-slip
Wall thermal condition	Adiabatic
Radiation heat transfer	Neglected
Evaporative cooling	Neglected
Gravity effects	Included
Solver type	Pressure-based
CFD software [name and version]	ANSYS Fluent 2022 R2
Turbulence model	Realizable k- ϵ with standard wall functions
Mesh type / total element count	Unstructured tetrahedral, ~1,800,000 cells (selected from mesh-independence study, Section 2.4)
Porous media viscous resistance coefficient [m ⁻²]	3.5×10^7 (derived analytically from Ergun-type formulation for perforated tray geometry)
Porous media inertial resistance coefficient [m ⁻¹]	25 (derived analytically from Ergun-type formulation for perforated tray geometry)

3. Results and Discussion

3.1 Transient heat transfer of drying air

The temperature contour plots of the drying air predicted by the CFD model at selected time intervals within the indirect solar fish dryer are presented in Figure 4. At the early stage of operation (1–32 s), relatively low air temperatures are observed throughout most of the drying chamber, with localized regions of elevated temperature near the lower inlet. As drying time increases (128–512 s), a continuous upward propagation of heated air develops, forming a thermal plume extending toward the upper section of the chamber. At longer operating times (1024–2048 s), the temperature field becomes increasingly uniform, with air temperatures throughout the chamber approaching the inlet temperature of approximately 60 °C. Similar transient-to-steady thermal behavior has been reported in three-dimensional CFD studies of forced-air and indirect solar dryers [16, 17].

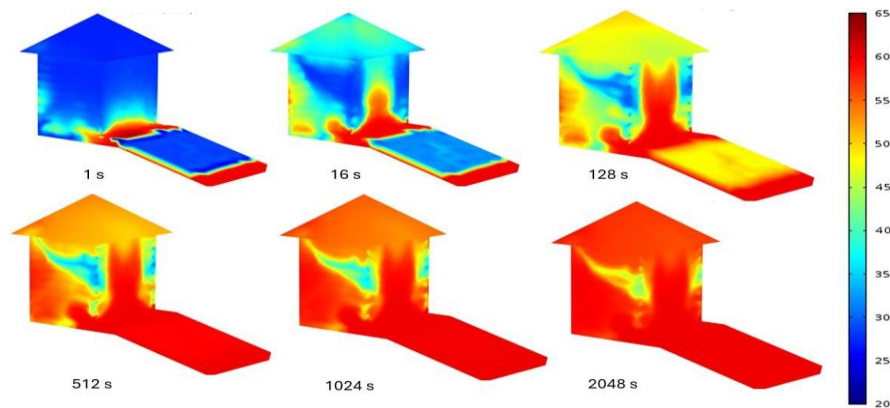


Figure 4. Time-dependent temperature contours of drying air inside the indirect solar fish dryer.

3.2 Temperature Distribution of Fish Samples

The temperature distribution on the fish samples placed on the perforated trays at selected drying times is presented in Figure 5. During the initial drying phase (0.5–32 s), the fish samples remain at relatively low temperatures. As drying time progresses (128–512 s), the fish samples on the lower and middle trays exhibit a noticeable increase in temperature. At extended drying times (2048–8400 s), temperature differences among tray levels diminish, and the fish sample temperatures converge to 55–60 °C across all trays. The progressive increase and eventual uniformity of fish sample temperatures across tray levels highlight the strong influence of airflow distribution on product heating. Fish samples located closer to the lower inlet experienced earlier heating due to higher local air temperatures and velocities, while samples on upper trays warmed more gradually during the initial stages. This trend is consistent with previous numerical and experimental investigations, which have shown that sustained and uniform airflow is essential for achieving homogeneous drying conditions and minimizing product quality variation [1,2].

3.3 Airflow Velocity Distribution

The airflow velocity contours inside the drying chamber at selected time intervals are shown in Figure 6. At early times (1–32 s), airflow is concentrated near the lower inlet region, with relatively low velocities observed in the upper sections of the chamber. As the operation continues (512–1024 s), a continuous upward airflow path forms from the inlet toward the upper exhaust vent. At longer times (2048 s), the airflow distribution becomes relatively uniform across all tray levels, with no significant stagnant or recirculation zones observed. Perez et al. [16] demonstrated that airflow channeling and non-uniform velocity fields can significantly influence convective heat transfer in shallow-depth drying systems. The gradual stabilization of airflow across tray levels in the present study contributed to improved heat transfer and temperature uniformity, underscoring the importance of chamber design and tray configuration in indirect solar dryers [22, 23].

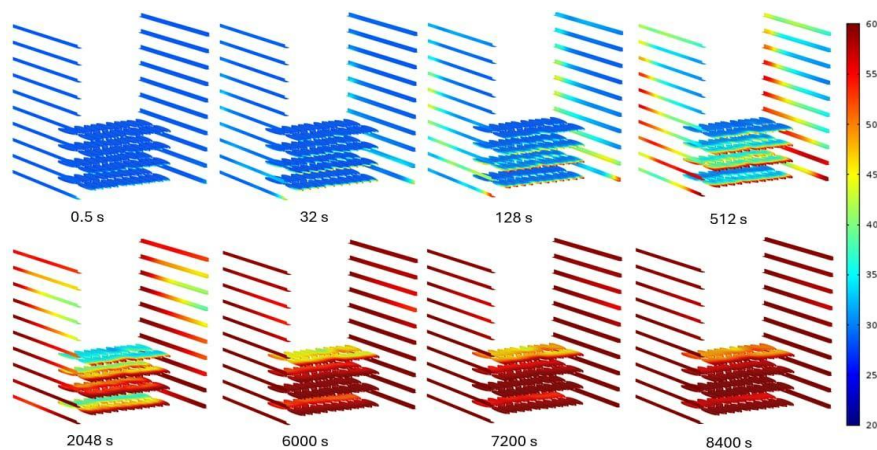


Figure 5. Temperature evolution of fish samples on perforated trays during drying.

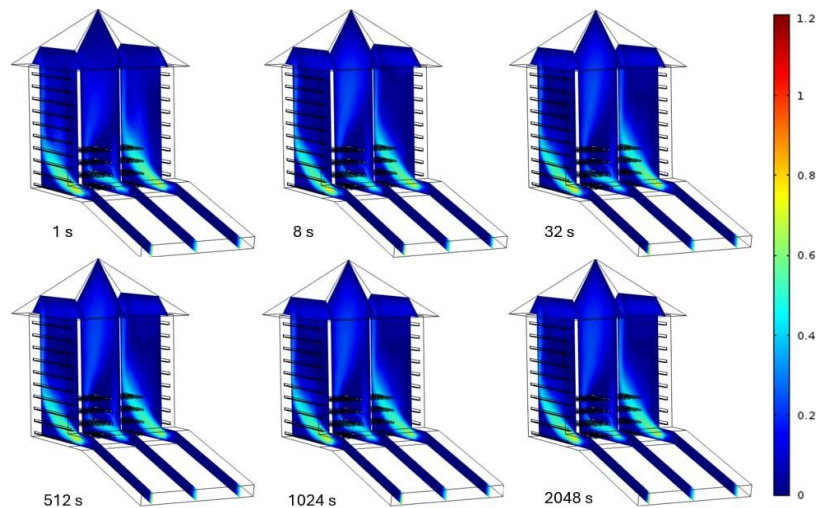


Figure 6. Airflow velocity contours showing the establishment of uniform upward flow in the drying chamber.

3.4 Temporal Evolution of Simulated and Experimental Drying Air Temperature

The temporal evolution of the simulated and experimentally measured drying air temperatures inside the indirect solar fish dryer is shown in Figure 7. Both datasets exhibit a similar overall trend, characterized by an initial rapid temperature rise, followed by a near-steady region, and a gradual decline at extended drying times.

During the initial stage of drying (≤ 3000 s), the drying air temperature increases rapidly from approximately 30°C to above 45°C , corresponding to the initial warm-up period during which heated air supplied by the solar dryer induces forced convection within the drying chamber. Simulated temperatures closely match the experimental measurements during this period, indicating that the CFD model reliably captures the system's transient thermal response. As drying time progresses (approximately 5000 – $18,000$ s), both simulated and experimental temperatures approach a quasi-steady state, with peak values of 58 – 61°C , reflecting the establishment of thermal equilibrium among the heated air, the drying chamber, and the surrounding environment [10, 24]. Similar temperature–time responses have been reported in validated CFD studies of indirect and solar-assisted dryers [22, 23]. At extended drying times ($>18,000$ s), the experimental temperature decreases gradually, whereas the simulated temperature remains relatively stable. This divergence may be attributed to factors not explicitly accounted for in the numerical model, such as variations in solar radiation intensity, ambient conditions, and evaporative cooling associated with moisture removal from fish samples [25, 26]. Despite this deviation, the overall agreement between simulated and experimental temperature trends remains strong across the drying duration.

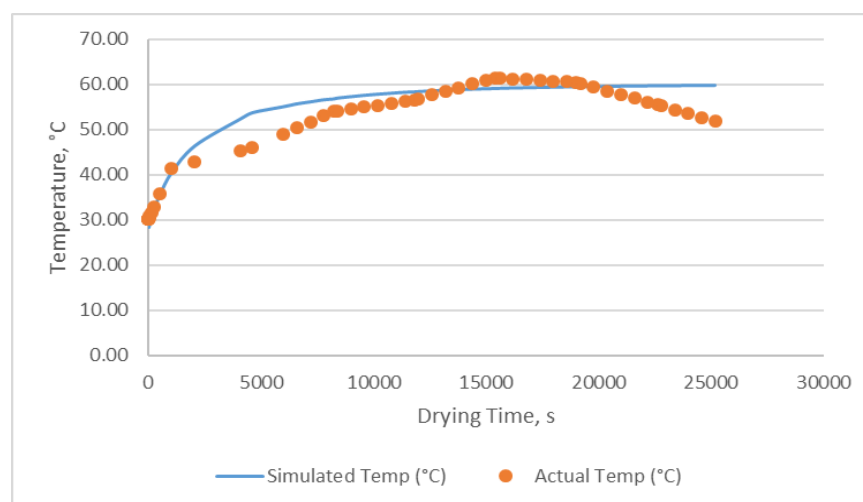


Figure 7. Simulated and experimental drying air temperature profiles as a function of drying time.

3.5 Correlation and Validation between Simulated and Experimental Temperatures

The predictive capability of the CFD model was quantitatively evaluated by correlating simulated drying air temperatures with experimentally measured values. A scatter plot comparing predicted and experimental temperatures is shown in Figure 8, and linear regression was used to assess the agreement between the two datasets.

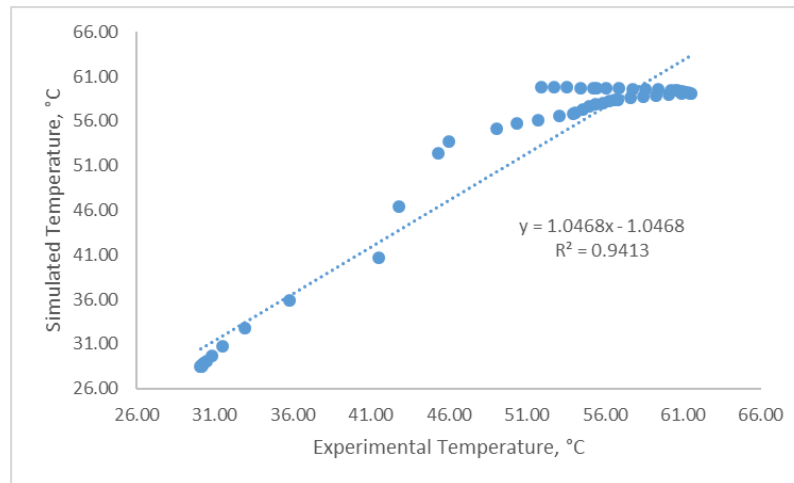


Figure 8. Correlation between simulated and experimental drying air temperatures.

The regression analysis shows a strong linear relationship between simulated and experimental temperatures, with $R^2 = 0.9413$. The fitted regression equation derived from the data is:

$$T_{\text{sim}} = 1.0468 \cdot T_{\text{exp}} - 1.0468 \quad (5)$$

The high R^2 value indicates that the CFD model accurately represents the dominant thermal behavior of the indirect solar fish dryer over the investigated temperature range. The regression slope of 1.0468, while close to unity, represents a systematic overestimation of approximately 4.7% across the operating range. At a representative operating temperature of 60 °C, this translates to a systematic bias of approximately 2.8 °C, which is non-negligible relative to the RMSE of 3.41 °C and should be considered when applying the model for design optimization. This overestimation may be attributed to the exclusion of evaporative cooling effects, which would otherwise suppress predicted temperatures during active drying. [Note: authors must also verify the intercept value in Eq. (5), as the identical slope and intercept values (both 1.0468) are statistically anomalous.] Discrepancies between simulated and experimental results are primarily attributed to the exclusion of evaporative cooling effects and the use of adiabatic wall conditions [23,25]. Evaporative cooling during the constant-rate drying period suppresses fish surface temperatures through latent heat exchange, an effect not captured in the present model. A rough estimation suggests that latent heat fluxes associated with moisture removal from fish can be on the order of 200–400 W·m⁻² during the constant-rate drying period (based on typical fish drying rates of 0.3–0.6 kg·m⁻²·h⁻¹ and a latent heat of vaporization of ≈ 2.4 MJ·kg⁻¹), which may account for a portion of the observed RMSE of 3.41 °C. While these simplifications are commonly adopted in thermal CFD models to reduce computational complexity [4, 26, 27], their impact is most pronounced at extended drying times (>18,000 s), as shown in Figure 7. The RMSE of 3.41 °C represents approximately 9.7% of the full operating temperature range (30–65 °C). For fish drying quality assurance, temperature uniformity within ± 5 °C is generally considered adequate to prevent microbial growth and ensure product safety, consistent with the Codex Alimentarius Code of Practice for Fish and Fishery Products (CXC 52-2003). On this basis, the model accuracy is considered acceptable for comparative design evaluation, though not for precise thermal optimization without further refinement. Overall, the validation results confirm that the developed CFD model provides a robust representation of the thermal performance of the indirect solar dryer and can serve as a useful predictive tool for dryer analysis and design optimization.

4. Conclusions

This study integrated CFD analysis with experimental validation to investigate airflow characteristics and convective heat transfer in an indirect solar fish dryer. The CFD results demonstrated that the indirect solar dryer effectively establishes an upward airflow path, promoting convective heat transfer within the drying chamber. Simulated temperature distributions showed a transition from localized heating near the inlet to a more uniform thermal field as drying progressed. The corresponding temperature evolution of fish samples indicated improved thermal uniformity across tray levels at extended drying times, which is essential for consistent drying performance and product quality. Experimental measurements of drying air temperature exhibited trends similar to the numerical predictions, characterized by an initial rapid heating phase followed by a quasi-steady thermal regime. The close correspondence between simulated and measured data was verified through regression analysis, which produced a high coefficient of determination ($R^2 = 0.9413$). These findings confirm that the developed CFD model reliably represents the dominant convective heat transfer processes governing the thermal behavior of the indirect solar dryer. For instance, a pulse-copper pipe configuration integrated into the drying chamber has been shown to extend effective drying periods during low-solar-radiation periods, and such hybrid thermal approaches represent a promising avenue for improving dryer performance under variable weather conditions. The findings highlight the usefulness of CFD as an effective tool for analyzing and optimizing indirect solar drying systems under tropical conditions. The validated model provides a foundation for evaluating design modifications to improve thermal uniformity and drying efficiency. The CFD results indicate that temperature non-uniformity between the lower and upper tray levels is most pronounced during the initial warm-up phase, suggesting that repositioning the inlet to a mid-chamber height or introducing flow deflectors at the lower inlet could promote earlier and more uniform heating across all tray levels. Additionally, the convergence of tray temperatures observed at extended drying times supports the potential benefit of multi-speed fan control strategies that reduce airflow velocity during the falling-rate drying period, thereby conserving energy without sacrificing product quality. These design insights, derived directly from the validated CFD model, offer practical pathways to improve the performance of indirect solar fish dryers under tropical operating conditions. Future studies may extend the present work by incorporating moisture transport, evaporative cooling, and transient solar radiation to enhance model accuracy further and support comprehensive dryer optimization.

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Conflicts of Interest: The authors declare that there are no conflicts of interest associated with the conduct of this research, the preparation of the manuscript, or its publication.

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