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Heat Transfer Enhancement in Solar Air Heaters Using Porous Ribs

Kherrour Sofiane[1]*, Boutina Lyes[1], Lebbi Mohamed[1], Zarrit Rida[1], Khelifi Reski[1], Dadda Bachir[1]

[1] Unité de Recherche Appliquée en Energies Renouvelables URAER, Centre de Développement des Energies Renouvelables, CDER 47133, Ghardaïa, Algeria,

*Corresponding author: (Sofiane Kherrour), *Email Address*: sofianekherrour@gmail.com

Abstract

Solar air heaters (SAHs) suffer from inherently low thermal efficiency due to poor convective heat transfer between the absorber plate and airflow. This study numerically investigates a novel passive enhancement technique: integrating periodically placed porous ribs on the absorber plate. A comprehensive computational fluid dynamics (CFD) analysis was conducted using ANSYS Fluent to model turbulent flow (Reynolds number range: $3000 \leq Re \leq 12000$) and heat transfer within a rectangular SAH duct. The finite volume method (FVM) solved the governing equations, employing the validated RNG k-ε turbulence model and the Darcy-Brinkman-Forchheimer model for the porous rib regions. The primary objectives were to quantify the impact of porous ribs on heat transfer and fluid flow characteristics compared to both smooth ducts and conventional solid ribs, and to optimize key geometric parameters: relative rib height (e/D), relative rib pitch (P/e), and rib porosity. Results demonstrate that porous ribs significantly enhance thermal performance.

Keywords: Porous Ribs; Solar Air Heater; Heat Transfer Enhancement; Thermo-hydraulic Performance.

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1. Introduction

Solar air heaters (SAHs) represent a vital technology for harnessing renewable energy for applications like space heating, drying, and ventilation. However, their widespread adoption is often limited by inherently low thermal efficiency, primarily stemming from poor heat transfer characteristics between the absorber plate and the flowing air stream [1]. To address this, numerous passive heat transfer enhancement techniques have been investigated over decades.

One of the most prevalent and effective passive methods involves artificially roughening the absorber surface, typically by attaching periodic ribs, baffles, or turbulators. These structures disrupt the laminar sub-layer, promote turbulence, and enhance fluid mixing near the hot surface, significantly improving convective heat transfer [2, 3]. Extensive research, such as the foundational work by Promvonge and Thianpong [4], has demonstrated the effectiveness of solid ribs (e.g., transverse, V-shaped, arc-shaped) in augmenting the Nusselt number (Nu), a key indicator of convective heat transfer performance. However, a critical drawback of solid ribs is the concurrent substantial increase in friction factor (f), leading to higher pumping power requirements [5]. Optimizing rib geometry (pitch (P), height (e), shape) is therefore essential to achieve the best thermo-hydraulic performance (THP), often characterized by parameters like the Performance Evaluation Criterion (PEC) balancing Nu and f enhancements [6, 7]. The computational domain and approach adopted in this work build upon established methodologies like those used by Yadav and Bhagoria [3].

Recently, the integration of porous media within thermal systems, including SAHs and solar dryers, has gained significant attention due to its potential for superior heat transfer augmentation [8, 9]. Porous materials (e.g., metal foams, packed beds, porous blocks) offer a high surface area-to-volume ratio and promote intense local mixing and thermal dispersion within the flow [10]. This characteristic makes them particularly attractive for enhancing energy absorption and transfer efficiency in solar thermal applications [11, 12]. Studies have suggested that porous inserts can lead to significant performance gains, potentially offering a more favorable trade-off between heat transfer enhancement and pressure drop compared to some solid turbulator configurations [13].

Despite the effectiveness of porous media, the specific application of porous ribs as discrete, periodic roughness elements on the SAH absorber plate remains relatively underexplored compared to continuous porous inserts or solid ribs. Porous ribs offer the potential to combine the boundary layer disruption mechanism of traditional ribs with the intense local mixing and extended heat transfer surface inherent to porous structures, potentially leading to superior thermal performance without a proportional penalty in friction loss.

Motivated by this gap, the present study employs a rigorous numerical approach to investigate the heat transfer and fluid flow characteristics in a rectangular SAH duct equipped with porous ribs on the absorber plate. Where, the primary objectives are:

- To analyze the impact of porous ribs on key performance parameters: average Nusselt number (Nu), convection heat transfer coefficient, and friction factor.

- To systematically investigate the influence of critical geometric parameters, including relative rib height (e/D), relative rib pitch (P/e), and rib porosity.

- To compare the performance of porous ribs directly against conventional solid ribs under identical operating conditions.

- To identify optimal geometric configurations for maximizing the thermo-hydraulic performance of the SAH.

Utilizing the Finite Volume Method (FVM) within ANSYS Fluent, the study solves the governing Navier-Stokes equations coupled with the energy equation. Turbulence is modeled using the RNG k-ε model, validated against literature data for accuracy. Flow within the porous ribs is modeled using the Darcy-Brinkman-Forchheimer (DBF) approach. The computational domain follows established practices for SAH simulation, including entrance, test, and exit sections [3].

This work aims to provide valuable insights and design guidelines for the implementation of porous ribs as a novel and potentially highly efficient heat transfer enhancement technique for next-generation solar air heaters.

2. Methods and modelisation

2.1. System Presentation

The problem to be considered in this study is presented schematically in Figure 1. It involves the 2-D heat transfer and fluid flow characteristics of turbulent flow in rectangular duct of solar air heater with solid ribs and porous ribs, respectively.

The computational domain of an artificially roughened solar air heater is represented in 2D form by a rectangle. The computational domain used in this work has been adopted from the authors' previous study Yadav and Bhagoria (2013). The domain consisted of three sections, namely entrance section (L1), test section (L2), and exit section (L3). The internal duct cross section is 50 mm × 10 mm. The rib pitch (P) and rib height (e) were varied. The rib pitch to rib height ratio, P/e, varied from 4.46 to 8.57. The rib height to hydraulic diameter ratio, e/D, varied from 0.049 to 0.11, and Reynolds number (Re) varied from 3000 to 12000.

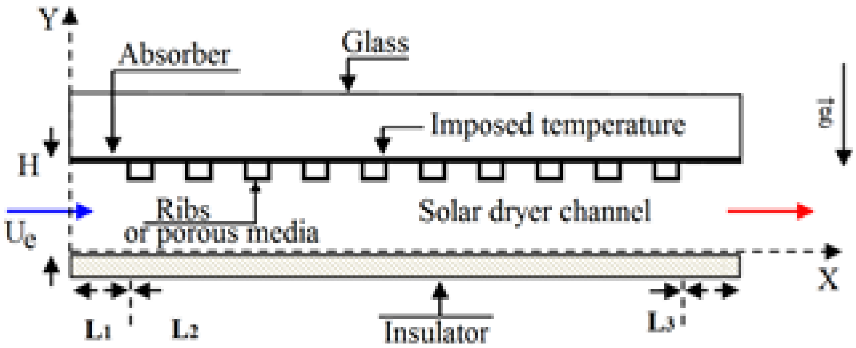

Figure 1. Schematic of 2D computational domain

2.2. Model Development

In this study, we based on the numerical optimization of geometric and physical parameters to improve the performance of a horizontal solar collector by integrating, on the one hand, solid ribs and on the other hand porous ribs. To achieve this objective, the flow and heat transfer in turbulent forced convection in the solar collector channel were analysed.

Consider a forced convection with turbulent flow inside a parallel plate heater. A porous rib is placed at the plate with thickness (e). The fluid (air) enters the domain with a uniform velocity distribution (Ue) and constant temperature (Te). A constant and hot temperature (Tc) is used at the plate and it is uniform along the channel. Computational domain and coordinate system for the heater are shown in Figure 02. Numerical analysis was performed using ANSYS to solve the governing equations, Navier Stokes equations, and the Darcy- Brinkman Forchheimer model in the porous domain

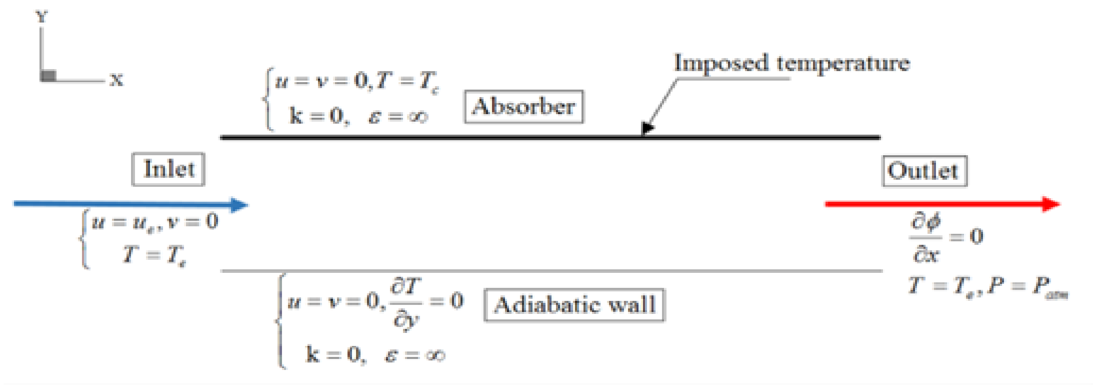

Figure 2. Computational domain for the heater

TABLE I. GOVERNING EQUATIONS

Equations	ϕ	Γ	S_ϕ
Continuity equation	1	0	0
Momentum equation	U	$\frac{(1+\upsilon_t^*)}{Re}$	$-\frac{\partial P^*}{\partial x^*}+\frac{1}{\varepsilon Re}\left[\frac{\partial}{\partial x_j^*}\left((1+\upsilon_t^*)\frac{\partial u_j^*}{\partial x^*}\right)\right] -\frac{1}{ReDa}u_i^*-\frac{C_F}{\sqrt{Da}}\sqrt{u_i^2+u_j^2}u_j$
	V		$-\frac{\partial P^*}{\partial y^*}+\frac{1}{\varepsilon Re}\left[\frac{\partial}{\partial x_j^*}\left((1+\upsilon_t^*)\frac{\partial u_j^*}{\partial y^*}\right)\right] -\frac{1}{ReDa}u_i^*-\frac{C_F}{\sqrt{Da}}\sqrt{u_i^2+u_j^2}u_j$
Energy	θ	$\frac{Rc(1+\alpha_t^*)}{Pe}$	0
Turbulent energy	k	$\frac{1}{Re}\left(1+\frac{\upsilon_t}{\sigma_K}\right)$	$\frac{1}{Re}(P_K^*+G_K^*-\varepsilon^*)$
Turbulent dissipation	ϵ	$\frac{1}{Re}\left(1+\frac{\upsilon_t}{\sigma_\varepsilon}\right)$	$\frac{1}{Re}\left[C_{\varepsilon 1}(P_K^*+C_{\varepsilon 3}G_K^*)-C_{\varepsilon 2}\varepsilon^*\right]\frac{\varepsilon^*}{k^*}$

3. Results and discussions

Before presenting and discussing the results obtained, some simulations were carried out, in order to validate the calculation model considered in our numerical simulations, by comparing its results to the experimental data presented in the literature to give credibility to the performed work. Figure 03 shows the digital code validation. Noting that the deviation of the Nusselt number reaches 13.02% for the standard k-ε model, while it does not exceed 6.63% for the RNG k-ε model. Therefore, it is recommended to use the RNG k-ε model exclusively in the rest of the simulation process.

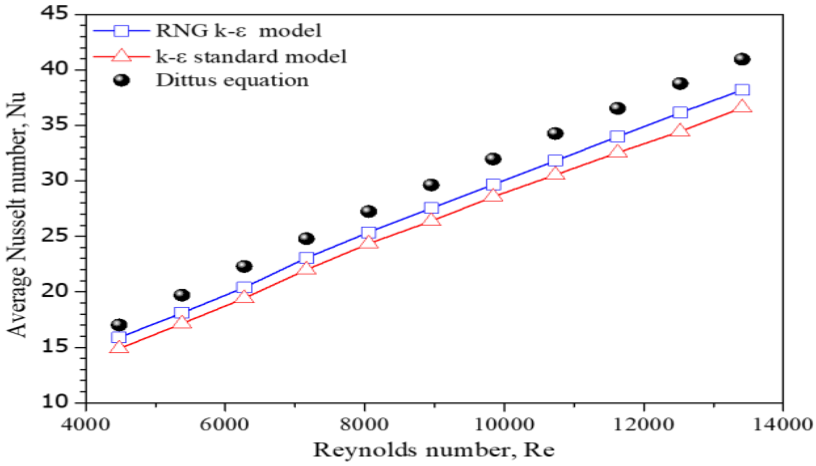

Figure 3. Validation of our numerical model

Figure 4 illustrates the evolution of the average Nusselt number as a function of mesh density, showing that the accuracy of the numerical results improves with an increase in the number of elements. Mesh independence is achieved at 93,092 elements, beyond which the Nu values stabilize (with variations of less than ±1%). The selected non-uniform quadrilateral mesh, characterized by a first near-wall cell size of 1.46 mm, optimizes both the resolution of boundary layer effects and computational efficiency, thereby ensuring the numerical robustness of the simulations.

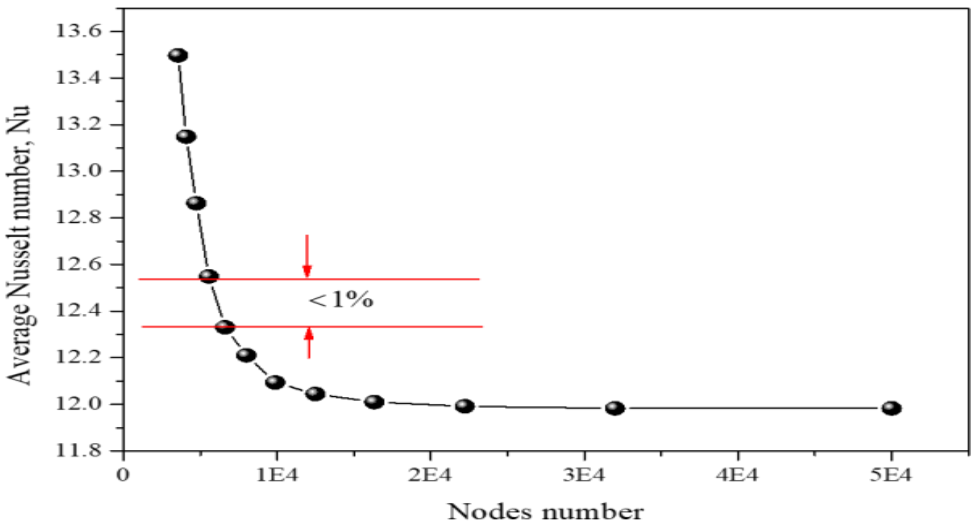

Figure 4. Evolution of the average Nusselt number according to the different meshes.

Figure 5 shows the variation of the heat transfer rate as a function of the relative roughness height (e/D). it is found that the Nusselt number (Nu) peak corresponds to e/D = 0.055 (Re=8000, ΔT=73.7°C). Below this height, ribs inadequately disrupt the thermal boundary layer. Above it, excessive flow blockage and recirculation zones reduce heat transfer. This identifies e/D=0.055 as the optimal height for maximizing thermal performance.

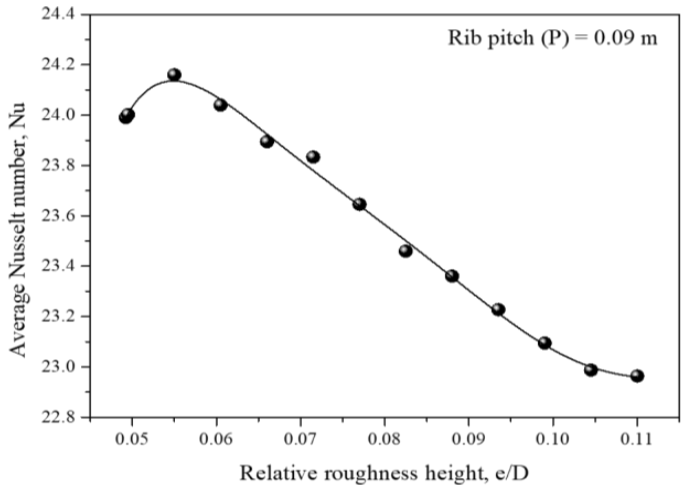

Figure 5. Variation of the average Nusselt number (Nu) as a function of the relative roughness height (e/D), (ΔT = 73.7°C and Re= 8000)

Figure 6 shows the evolution of the average Nusselt number as a function of the relative roughness pitch (P/e). It is observed that the average Nusselt number (Nu) increases with P0.05 (Re=8000), achieving a 48.5% improvement compared to the smooth channel SAH. At lower P/e, the ribs are too close together, causing problems with flow reconnection; whereas, at higher P/e, turbulence generation is reduced. P/e=0.05 optimally balances boundary layer disruption and flow redevelopment.

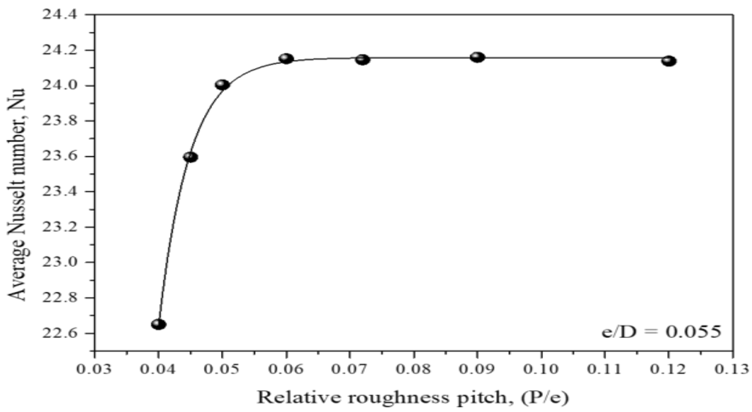

Figure 6. Variation of the average Nusselt number (Nu) as a function of the relative roughness pitch (P/e), (ΔT = 73.7°C and Re= 8000)

Figure 7 shows the effect of porosity on the Nusselt number for the solar air heater absorber. This indicates that the average Nusselt number (Nu) decreases as porosity increases. Low- porosity ribs (high solid fraction) maximize conductive heat transfer through the solid matrix. High porosity (>75%) permits significant flow bypassing, reducing fluid-solid interaction. This highlights a trade-off, lower porosity boosts Nu but may increase weight/cost.

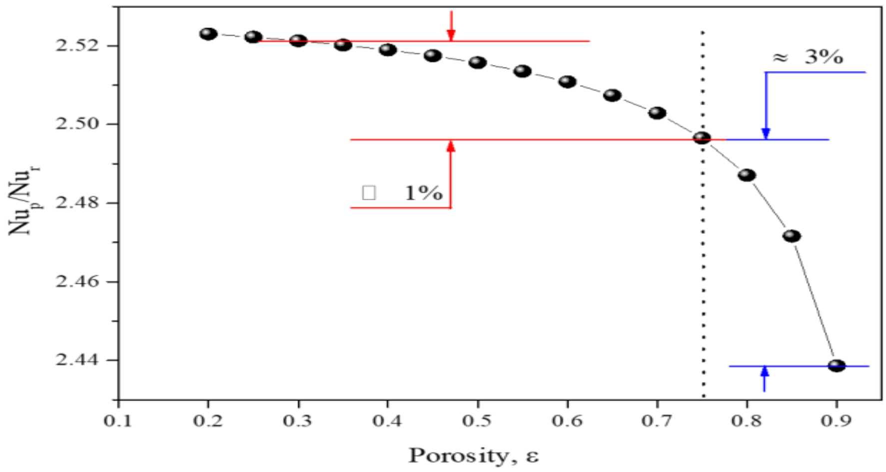

Figure 7. Variation of the average Nusselt number (Nu) as a function of the porosity, (ΔT = 73.7°C and Re= 8000)

4. Conclusion

The numerical simulation of a solar air heater whose absorber plate has porous ribs is carried out in the present study. It is observed that the implementation of a porous ribs over the absorber plate significantly increases the heat transfer rate in the solar air heater duct. These results can be attributed to the turbulence effects within the porous ribs.

This numerical study demonstrates that integrating porous ribs onto the absorber plate of solar air heaters (SAHs) is a highly effective strategy for enhancing thermal performance while mitigating friction losses. Using a rigorously validated CFD model (RNG k-ε turbulence and Darcy-Brinkman-Forchheimer porous media treatment), we analyzed turbulent airflow (Re: 3000–12000) and optimized critical geometric parameters. Key findings reveal:

- Porous ribs increase the average Nusselt number by 20–50% across the Reynolds number range compared to conventional smooth-channel SAHs, confirming superior heat transfer intensification.
- Thermal performance peaks at the optimal geometric ratios, where the relative rib height *e/D*

= 0.055 and the relative rib pitch *P/e* = 0.05, achieving a 48.5% Nu enhancement at Re=8000.

- Porosity has an inverse relationship with heat transfer. As porosity increases Nu decreases, which suggests that lower porosity ribs (higher solid fraction) are best for maximizing thermal gain.

- Most importantly porous ribs have a lower friction when compared to solid rib turbulators at the same level of heat transfer enhancement, highlighting their superior thermo- hydraulic performance (THP).

These results provide actionable design guidelines for implementing porous ribs in SAHs. By strategically optimizing rib geometry (*e/D*, *P/e*) and porosity, engineers can develop significantly more efficient solar heating systems with lower pumping power requirements. This work advances passive heat transfer enhancement techniques for sustainable energy applications.

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