

Geometric optimisation of volumetric solar receivers: a study of polygonal cavity configurations

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ABSTRACT

Volumetric solar receivers integrated with reticulated porous ceramics (RPCs) are a key component in high-temperature concentrated solar power (CSP) systems, enabling efficient radiative absorption and convective heat transfer. While prior studies have largely focused on material properties and operating conditions, the influence of cavity geometry on thermal performance remains underexplored. This study presents a systematic computational investigation of four polygonal cavity configurations: hexagonal, heptagonal, octagonal, and nonagonal, using high-fidelity CFD simulations in ANSYS Fluent coupled with the Monte Carlo radiation model. All designs were evaluated under consistent geometric constraints and two solar heat flux inputs (4.1 kW and 4.9 kW), with varying air mass flow rates. The nonagonal receiver achieved the highest thermal efficiencies of 75 % and 73 % at the respective flux levels, outperforming conventional designs. This improvement is attributed to its compact internal structure and increased edge count, which enhance surface energy density and fluid-wall interaction. The findings demonstrate that geometric optimisation, particularly through polygonal cavity design, offers a viable pathway to enhance the thermal performance of volumetric receivers. This work provides new design insights for next-generation CSP applications requiring compact, high-efficiency thermal energy conversion.

1. Introduction

The rising global demand for clean, high-temperature energy has accelerated the development of solar thermal technologies, particularly concentrated solar power (CSP) systems. These systems are essential in electricity generation, industrial heating, and hydrogen thermochemical production, where efficient heat capture and transfer are key to performance [1–3]. At their core lies the solar receiver, which absorbs concentrated solar radiation and transfers heat to the working fluid. Improving receiver efficiency is thus central to enhancing CSP viability [4–6]. Volumetric receivers using reticulated porous ceramics (RPCs) have shown strong potential in high-temperature applications [7]. The porous structure allows deep radiation penetration and airflow, enabling volumetric absorption and efficient heat transfer [8]. This makes cavity geometry a critical factor affecting both radiative and

convective performance [9–12].

Several studies have focused on hybrid CSP systems for thermochemical processes, such as integrating cylindrical receivers with porous catalytic reactors for solar-driven propane reforming [13,14]. These systems demonstrate the promise of porous media but typically rely on standard cavity shapes, neglecting the geometry's effect on thermal performance. Most optimisation efforts target reaction kinetics or heat transfer fluid (HTF) circulation, with limited exploration of geometric enhancements. Material-level improvements have included composite absorbers with graded porosity to reduce temperature gradients and boost outlet temperatures [15–17].

Microstructural changes, such as those proposed by Capuano et al. [18] and Luque et al. [19], enhanced heat absorption via optimised layouts and fractal geometries, achieving outlet temperatures up to 1200 K and 805 K, respectively. However, thermal efficiencies remained close

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to conventional levels. Similarly, Meng et al. [20] showed that a cup-shaped cavity using Al_2O_3 foam could reach 83.7 % efficiency, but their study was limited to axisymmetric designs and did not explore polygonal cross-sections. Other improvements, such as double-layer ceramic foams [14], helped control radiation losses and boost outlet temperatures but introduced higher pressure drops and remained focused on material design, not cavity geometry. Moreover, simplified radiation models restrict their predictive accuracy for real CSP environments. A few studies have explored cavity shape using ray tracing and CFD. Daabo et al. [21] compared cylindrical, conical, and spherical cavities using Monte Carlo methods, showing differences in optical efficiency but ignoring heat transfer. Others proposed new shapes like the upside-down pear cavity [22] or used CFD to assess chamfered geometries [23,24]. In a related experimental study, V. Patil et al. [25] investigated an octagonal cavity receiver with RPCs, achieving outlet air temperatures up to 1406 K and an efficiency of 69 % under a 4.9 kW input. Building on this validated baseline geometry, the present study explores alternative polygonal configurations, ranging from hexagonal to decagonal, to investigate whether geometric optimisation can further enhance thermal efficiency and surface energy distribution. By extending Patil's work, this paper aims to uncover the performance benefits of increasing or decreasing the number of cavity faces while maintaining compactness.

Complementary strategies like heliostat aiming optimisation e.g., Besarati et al. [26] used a genetic algorithm to help manage external heat flux, but does not address internal geometric losses. More accurate radiation models have emerged: Wu et al. [27] introduced a two-band Monte Carlo method validated against experiments, showing grey models underestimate radiative heat transfer. However, their focus was on photothermal catalysis, not CSP receiver geometry. Despite these advancements, many studies continue to rely on simplified models (e.g., P1, Rosseland) or neglect fluid flow, limiting their ability to predict convective losses and real operating temperatures. Furthermore, outlet temperatures often remain below 1300 K, which is insufficient for advanced CSP processes like hydrogen production, which demand >1300 K [28]. This study fills a key gap by systematically analysing how polygonal cross-sectional cavity geometries, specifically hexagonal, heptagonal, octagonal, and nonagonal, affect the thermal performance of volumetric solar receivers. Using high-fidelity CFD simulations in ANSYS Fluent, the study couples the Monte Carlo radiation model with convective heat transfer in air. Simulations were conducted at two solar inputs (4.1 kW and 4.9 kW), across different mass flow rates, while maintaining constant parameters (aperture diameter, wall thickness, air gap) to ensure fair comparison. The nonagonal design achieved the best performance, with a peak efficiency of 75 % and outlet temperatures exceeding 1500 K. This result shows that increasing the number of polygonal faces while maintaining compactness enhances surface energy density, radiative absorption, and convective performance. Unlike previous studies, this work captures the coupled radiation convection effects in complex geometries using advanced numerical modelling.

The key contribution of this paper lies in introducing cavity geometry optimisation as a new, effective design strategy for CSP receivers. It demonstrates that polygonal shapes can significantly boost thermal efficiency in high-temperature CSP applications, particularly for compact integrated systems like receiver-reactors or receiver-storage units. The remaining part of this paper is structured as follows. Section 2 introduces the solar receiver design and describes the polygonal cavity configurations analysed in the study. Section 3 outlines the numerical modelling methodology, including mesh generation, boundary conditions, and the integration of radiative and convective heat transfer models. Section 4 presents the validation of the simulation results against experimental data to ensure model accuracy. Section 5 provides a detailed analysis of thermal performance across the different cavity geometries and operating conditions. Section 6 concludes the study by summarising the key findings and offering recommendations for future research in the optimisation of high-temperature CSP receiver designs.

2. Solar receiver design

The solar receiver in this study is based on a cavity-type volumetric receiver concept, designed to efficiently capture and transfer heat from concentrated solar radiation. It features a reticulated porous ceramic (RPC) absorber, which allows the incident solar flux to penetrate deep into the porous matrix, enhancing volumetric absorption and minimising radiative losses. Unlike conventional flat-plate receivers, this design ensures that the highest temperatures occur within the cavity, reducing energy loss to the surroundings. The receiver operates in an open-airflow configuration, where ambient air enters through the aperture, passes through the porous structure, absorbs heat, and exits at elevated temperatures.

Structurally, as shown in Fig. 1, the receiver is designed to accommodate multiple cavity configurations formed by arranging modular trapezoidal RPC bricks to create the respective polygonal geometry. The aperture span of the SiSiC RPC, defined as the internal width across the cavity through its centre, is fixed by setting a 40 mm distance from the cavity centre to each inner wall face. This ensures a consistent internal volume across all geometric configurations, while the wall thickness is maintained at 24 mm across all configurations. The cavity dimensions are carefully set, with the distance from the cavity centre to the inner edge fixed at 40 mm, ensuring comparable internal volume across different geometric configurations. Additionally, a 5 mm air gap is maintained between the RPC structure and the surrounding insulation, allowing for effective convective heat transfer while minimising thermal losses. To further enhance thermal containment, the cavity is lined with ceria laminate insulation, reducing heat dissipation while maintaining high-temperature stability. The front face of the cavity is sealed using alumina-silica insulating material, leaving only the aperture open for radiation intake. The entire system is enclosed within a stainless-steel housing, providing mechanical protection and thermal stability, while an aluminium shield surrounds the aperture to minimise re-radiation losses and enhance energy retention. The selected porosity of the reticulated porous ceramic (RPC) in this study is PPI10, chosen for its superior ability to absorb and transfer solar heat compared to higher porosity structures like PPI20 and PPI30 [25]. This design ensures that the receiver maintains high efficiency under high-temperature conditions, making it well-suited for CSP applications requiring enhanced volumetric heat absorption.

3. Receiver geometry configurations

To investigate the influence of cavity geometry on the thermal performance of solar volumetric receivers, this study examines four distinct polygonal configurations: hexagonal (6-sided), heptagonal (7-sided), octagonal (8-sided), and nonagonal (9-sided). Each configuration is constructed by assembling modular trapezoidal RPC bricks to form the corresponding polygonal cavity. This approach enables a systematic evaluation of how the number of faces affects radiative heat absorption, convective flow behaviour, and re-radiation losses. The cross-sectional view of all four cavity configurations studied in this work is illustrated in Fig. 2. Polygonal geometries were selected because they provide a modular and practical structure for assembling SiSiC RPCs while offering a wide range of surface areas and angular arrangements that significantly influence radiative view factors and flow paths inside the cavity. To ensure a consistent and fair comparison across all geometries, several key design parameters were kept constant:

1. The radial distance from the cavity centre to the inner edge was fixed at 40 mm for all configurations, ensuring comparable exposure to solar flux while allowing the natural variation in internal volume to reflect the influence of geometry.
2. The wall thickness was maintained at 24 mm, thereby preserving the thermal conduction path across all designs.

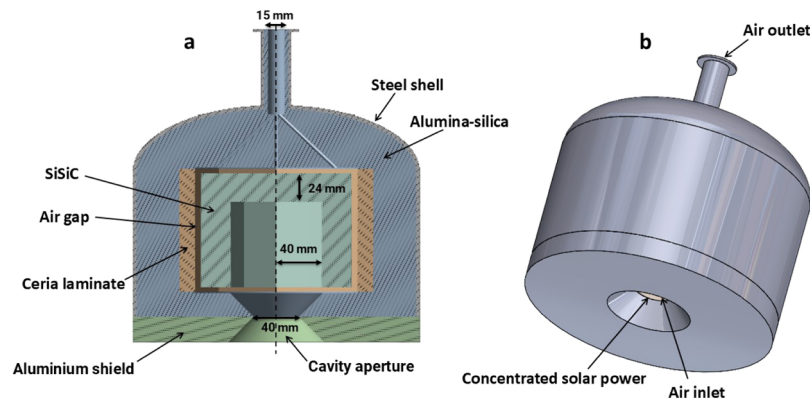


Fig. 1. a. Cross-section view showing dimensions and materials, b. 3D external view of the receiver.

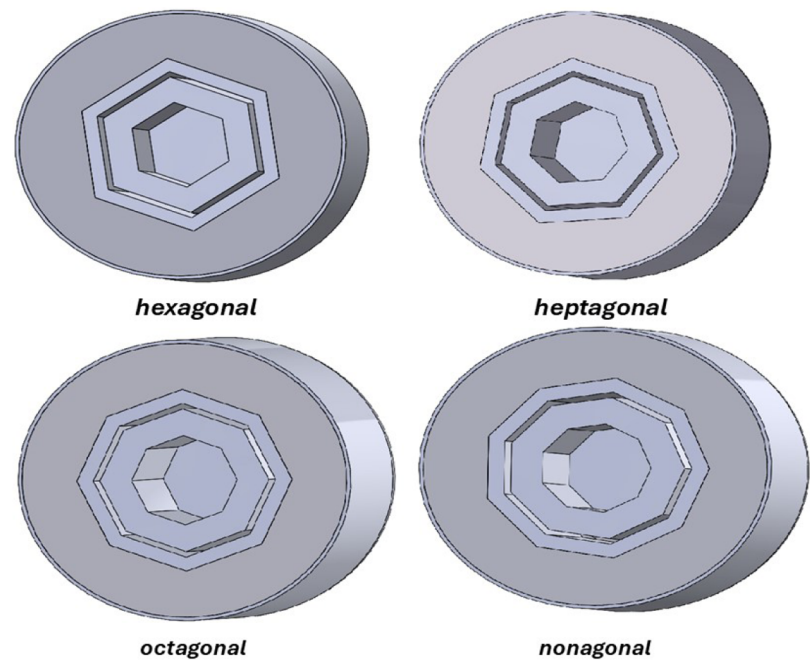


Fig. 2. Cross-sectional view of the receiver showing different geometries: hexagonal (6-sided), heptagonal (7-sided), octagonal (8-sided), and nonagonal (9-sided).

- 3. The air gap between the RPC structure and insulation layer was maintained at 5 mm, facilitating uniform convective heat transfer.
- 4. The aperture diameter was fixed at 40 mm, ensuring that each configuration received the same amount of concentrated solar radiation.

By keeping these design and simulation parameters consistent, the influence of cavity geometry on performance could be clearly evaluated. Each geometric configuration yields measurable variations in key physical attributes, including the internal total surface area, the width of the cavity faces, the base area, the total cavity volume, and the overall surface area. These differences are directly linked to the heat absorption, conduction, and convective behaviour within the receiver. [Table 1](#)

provides a comparative overview of these geometric parameters, which are essential for interpreting the thermal performance of each configuration.

4. Numerical modelling of the solar receiver

This study evaluates the impact of internal cavity geometry on the receiver’s thermal performance by numerically simulating solar irradiation and airflow under controlled conditions. The parametric analysis of thermal efficiency in this study focuses on the effects of applied heat input and varying air mass flow rates. The results are then compared to available experimental data in the literature to validate the accuracy of the simulation outcomes. [Fig. 2](#) highlights the internal structural

Table 1
Comparison of Receiver Geometry Configurations.

Geometry	Faces	Internal Total Area mm2	Face Width mm	Base Area mm2	Total Cavity Volume (mm³)	Total Surface Area mm2
Hexagonal	6	26,343.8	46.2	5542.6	1003,722.6	107,402.7
Heptagonal	7	25,635.9	38.5	5393.6	976,752.9	104,516.8
Octagonal	8	25,200.1	33.1	5301.9	960,146.2	102,739.8
Nonagonal	9	24,911.3	29.1	5241.2	949,142.6	101,562.4

differences between the receiver designs, showing how the shape of the RPC cross-section varies across the studied configurations. These geometric differences influence how solar radiation is absorbed and distributed within the porous region. The simulations apply identical heat inputs (4.1 kW and 4.9 kW) and mass flow rates across all cases to ensure fair comparison. Radiation is absorbed volumetrically within the RPC, with each shape altering how solar rays are distributed, reflected, and reabsorbed within the cavity. The polygonal geometry also influences airflow through the porous media, affecting the velocity profile and convective heat transfer. These simulations are structured to isolate the effects of geometric shape, ensuring that any variation in thermal response is solely due to internal structural differences.

Air was selected as the working fluid due to its compatibility with high-temperature applications and its ability to operate safely under atmospheric pressure. It was modelled as an ideal gas, with temperature-dependent thermophysical properties including density, specific heat capacity, viscosity, and thermal conductivity. These properties were defined through user-defined polynomial functions manually entered in ANSYS Fluent, based on values obtained from the COMSOL material library [29]. The porous region, representing the RPC, was assigned a porosity of 0.9 and a calculated permeability of $4.92 \times 10^6 \text{ m}^2$, in accordance with structural characteristics of 10 PPI SiSiC foam. The solid phase was defined with temperature-dependent thermal conductivity appropriate for SiSiC materials. A thermal equilibrium model was assumed for the porous zone, enabling a single energy equation to be solved for both the solid and fluid phases, which simplifies the analysis while maintaining sufficient accuracy for high-temperature receiver simulations. The thermal energy Q absorbed by the receiver can be estimated using Eq. (1).

$$Q = c_p \cdot m \cdot \Delta T \quad (1)$$

Where Q is the heat absorbed by air in (J), c_p is the specific heat capacity of air, m is the mass of the air, and ΔT in (K) is the temperature difference between outlet and inlet given by Eq. (2). The surface applied heat flux used as the boundary condition in the simulation is derived by dividing the total solar input power by the internal surface area of the RPC as given by Eq. (3).

$$\Delta T = T_{out} - T_{in} \quad (2)$$

$$q'' = Q / A \quad (3)$$

Where Q is the total power input (e.g., 4.1 or 4.9 kW) and A is the internal surface area of the RPC receiving the flux. To characterise the airflow through the porous medium and evaluate the flow regime, the Reynolds number is given by Eq. (4).

$$Re = (\rho \cdot V \cdot D_h) / \mu \quad (4)$$

Where ρ is air density, V is the velocity of flow, D_h is the hydraulic diameter, and μ is the dynamic viscosity. It determines the flow regime. Given the dominance of radiative heat transfer in high-temperature cavity receivers, the net radiative flux absorbed by the porous media was indirectly represented via the surface-applied heat flux based on the Monte Carlo ray tracing results. In future refinement, the net radiative heat transfer rate Q_{rad} can be expressed using Eq. (5).

$$Q_{rad} = \varepsilon \cdot \sigma \cdot A \cdot T_{wall}^4 - T_{Surroundings}^4 \quad (5)$$

Where ε is the surface emissivity, σ is the Stefan-Boltzmann constant, and T values are the surface and ambient temperatures. Additionally, for post-processing and performance assessment, the Nusselt number N_u may be used to estimate the convective heat transfer coefficient inside the porous region given by Eq. (6).

$$N_u = \left(\frac{h}{D_h} \right) / k \quad (6)$$

Where h is the convective heat transfer coefficient and k is the thermal conductivity of air. To assess the influence of cavity shape on the thermal performance of volumetric solar receivers, mathematical models were developed for four regular polygonal cross-sections: hexagonal (6-sided), heptagonal (7-sided), octagonal (8-sided), and nonagonal (9-sided). Each geometry was assumed to be regular, with equal side lengths and internal angles, and the porous absorber was modelled as a cylindrical volume defined by its cross-sectional shape and height H . The circumradius R , defined as the distance from the centre to a vertex, was held constant across all configurations to ensure a fair comparison. For a regular n -sided polygon to characterise the receiver geometry, the cross-sectional area, perimeter, volume of the porous region and hydraulic diameter are given by Eqs. (7), (8), (9) and (10).

$$A_c = \frac{1}{2} n R^2 \sin\left(\frac{2\pi}{n}\right) \quad (7)$$

$$P = 2nR \sin\left(\frac{\pi}{n}\right) \quad (8)$$

$$V = A_c \cdot H = \frac{1}{2} n R^2 \sin\left(\frac{2\pi}{n}\right) \cdot H \quad (9)$$

$$D_h = \frac{4A_c}{P} \quad (10)$$

The hydraulic diameter D_h is particularly relevant in evaluating convective heat transfer, as it appears in dimensionless numbers such as the Reynolds and Nusselt numbers. The internal surface area of the cavity was also computed using the perimeter and height to determine the surface-applied heat flux in boundary conditions. Substituting the number of sides n for each configuration into Eqs. (7), (8), (9), and (10) yields the design characterisation for each design. Given that hexagonal design $n = 6$, substituting into the equations and simplifying gives Eqs. (11), (12), and (13). Similarly, for heptagonal design $n = 7$, therefore the characteristics of the design are given by Eqs. (14), (15), and (16). The same applies to octagonal design ($n = 8$) and nonagonal design ($n = 9$) to have Eqs. (17), (18), (19) and (20), (21), (22), respectively.

$$A_{c,6} = \frac{1}{2} \cdot 6 \cdot R^2 \cdot \sin\left(\frac{2\pi}{6}\right) = 3R^2 \cdot \sin\left(\frac{\pi}{3}\right) = 3R^2 \cdot \frac{\sqrt{3}}{2} = \frac{3\sqrt{3}}{2} R^2 \quad (11)$$

$$P_6 = 2 \cdot 6 \cdot R \cdot \sin\left(\frac{\pi}{6}\right) = 12R \cdot \frac{1}{2} = 6R \quad (12)$$

$$D_{h,6} = \frac{4A_{c,6}}{P_6} = \frac{4 \cdot \frac{3\sqrt{3}}{2} R^2}{6R} = \frac{2\sqrt{3}}{1} R = 2\sqrt{3}R \quad (13)$$

$$A_{c,7} = \frac{7}{2} R^2 \sin\left(\frac{2\pi}{7}\right) \quad (14)$$

$$P_7 = 14R \cdot \sin\left(\frac{\pi}{7}\right) \approx 14R \cdot 0.4339 = 6.0746R \quad (15)$$

$$D_{h,7} = \frac{4A_{c,7}}{P_7} \quad (16)$$

$$A_{c,8} = 4R^2 \cdot \sin\left(\frac{\pi}{4}\right) = 4R^2 \cdot \frac{\sqrt{2}}{2} = 2\sqrt{2}R^2 \quad (17)$$

$$P_8 = 2 \cdot 8 \cdot R \cdot \sin\left(\frac{\pi}{8}\right) \approx 16R \cdot 0.3827 = 6.1232R \quad (18)$$

$$D_{h,8} = \frac{4A_{c,8}}{P_8} \quad (19)$$

$$A_{c,9} = \frac{9}{2} R^2 \sin\left(\frac{2\pi}{9}\right) \quad (20)$$

$$P_9 = 18R \cdot \sin\left(\frac{\pi}{9}\right) \approx 18R \cdot 0.3420 = 6.156R \quad (21)$$

$$D_{h,9} = \frac{4A_{c,9}}{P_9} \quad (22)$$

The numerical values of sine, open paren 2 pi over 7 close paren and sine, open paren 2 pi over 9 close paren are approximately 0.7818 and 0.6840, respectively. The numerical approach is employed to evaluate the thermal performance of the receiver under various geometric and operating conditions. All simulations were conducted using ANSYS Fluent 2024 R2 to solve the governing equations under specified thermal and flow conditions. The energy equation was activated to calculate heat transfer within the domain. The simulation was carried out under steady-state conditions. Turbulence was modelled using the $k-\omega$ SST model, which provides accurate near-wall treatment and is well-suited for capturing complex flow behaviour in confined geometries. Furthermore, the simulations were carried out based on the assumptions that the applied heat flux is uniformly distributed over the internal surface area of the porous region, and the inlet air temperature was fixed at 293 K for all simulation cases.

4.1. Governing equations

The computational model employed in this study solves the governing fluid flow and heat transfer equations using ANSYS Fluent. The flow in the free air domain is described by the incompressible Navier–Stokes equations, while the airflow within the porous reticulated porous ceramic (RPC) domain is modelled using the Brinkman-extended Darcy formulation. These equations collectively describe the conservation of mass, momentum, and energy, enabling the simulation of both convective flow and heat exchange mechanisms in and around the porous medium. The continuity equation, representing mass conservation in the absence of any mass sources or sinks, applies to both the free-flow region and the RPC domain.

$$\nabla \cdot (\rho_{air} \mathbf{u}) \quad (23)$$

Where ρ_{air} represents the density of air. The conservation of momentum in the region of free flow is described by

$$\rho_{air} \mathbf{u}(\mathbf{u} \cdot \nabla) = \nabla \cdot [\boldsymbol{\tau} - p\mathbf{I}] + \vec{F} \quad (24)$$

Where $\mathbf{I}$ refer to the identity tensor, $\vec{F}$ is a vector representing the effect of internal forces per unit volume acting on the fluid, and $\boldsymbol{\tau}$ stands for the viscous stress tensor

$$\boldsymbol{\tau} = \mu_{air}((\nabla \cdot \mathbf{u})^T + \nabla \mathbf{u}) - \frac{2}{3}\mu_{air}(\nabla \cdot \mathbf{u})\mathbf{I} \quad (25)$$

Where μ_{air} represents the dynamic viscosity of the air. In the RPC domain, the conservation of momentum is described by an equation:

$$\frac{1}{\epsilon_{RPC}} \rho_{air}(\mathbf{u} \cdot \nabla) \mathbf{u} \frac{1}{\epsilon_{RPC}} = \nabla \cdot [-p\mathbf{I} + \boldsymbol{\tau}_{RPC}] - \mu_{air}\tau_{RPC}^{-1} + \vec{F} \quad (26)$$

The RPC's viscous stress tensor is defined as:

$$\boldsymbol{\tau}_{RPC} = \mu_{air} \frac{1}{\epsilon_{RPC}} (\nabla \mathbf{u} + (\nabla \cdot \mathbf{u})^T) - \frac{2}{3}\mu_{air} \frac{1}{\epsilon_{RPC}} (\nabla \cdot \mathbf{u})\mathbf{I} \quad (27)$$

The temperature of the RPC material (T_{RPC}) is determined by solving an energy conservation equation below, distinct from the one applied to the air

$$(1 - \epsilon_{RPC})\nabla \cdot (-k_s \nabla T_{RPC}) = Q_{rad} - Q_{conv} \quad (28)$$

Q_{rad} denotes the volumetric radiative heat source, which is obtained from the Monte Carlo radiation model implemented in ANSYS Fluent and Q_{conv} represents the convective heat exchange between the porous solid and the surrounding air due to local temperature differences.

$$\dot{q}_{conv} = h_{s-f} A_{sp} (T_{RPC} - T_{air}) \quad (29)$$

Where A_{sp} refer to the RPC specific surface area. Determining the air temperature (T_{air}) is by applying the following equation.

$$\epsilon_{RPC} \rho_{air} C_{p,air} \mathbf{u} \cdot \nabla T_{air} + \nabla \cdot (-\epsilon_{RPC} \tau_{air} \nabla T_{air}) = Q_{conv} + \dot{q}'' \quad (30)$$

The radiative heat flux $\dot{q}''$ applied to the cavity boundaries is also determined using the Monte Carlo radiation model embedded in the CFD simulation.

4.2. Boundary conditions, material properties and simulation settings

The simulation setup was conducted using ANSYS Fluent to investigate the thermal performance of the solar receiver. The viscous model was configured using the $k-\omega$ turbulence model, which is suitable for capturing near-wall effects. The energy equation was activated to account for heat transfer, and radiation effects were included by enabling the Monte Carlo radiation model, which offers accurate modelling of radiative heat transport within the RPC cavity. Material thermal properties for all domains were defined as detailed in Table 2. The cell zone conditions were assigned appropriately, with the porous zone enabled in the RPC wall, specifying a porosity of 0.9 to represent the reticulated porous ceramic accurately. For the boundary conditions, a mass flow rate was applied at the inlet, while an outlet pressure condition was specified at the exit. A solar heat flux was imposed on the internal SiSiC RPC wall to simulate concentrated solar input, and heat losses through the outer aluminium shield and steel shell were also considered to ensure realistic thermal boundary behaviour. To enhance understanding of the simulated flow configuration and applied boundary conditions, Fig. 3 illustrates the airflow path through the cavity receiver and the defined boundary zones.

5. Mesh independence study

A mesh independence study was carried out for each cavity configuration: hexagonal, heptagonal, octagonal, and nonagonal. The average outlet temperature was selected as the evaluation parameter due to its importance in determining thermal performance. Inflation layers were applied to the air and porous RPC domains to improve mesh accuracy in high-gradient regions. The final mesh element counts were 7699,519 for the hexagonal, 11,316,693 for the heptagonal, 1580,645 for the octagonal, and 11,109,467 for the nonagonal geometry. All mesh setups were tested and selected based on stability in the predicted temperature results. The results of the mesh independence tests are illustrated in Fig. 4 for all geometries. All mesh independence studies were performed at an input power of 4.1 kW to ensure consistent comparison across geometries.

6. Validation of numerical model

To validate the numerical model used for the simulation of the solar receiver, a comparison was performed between the simulated outlet air temperatures and experimental results for the octagonal receiver geometry at two input power levels: 4.1 kW and 4.9 kW. The experimental data were obtained from the literature and digitised for comparison [25]. The validation aims to assess the model's accuracy in replicating real system behaviour under varying mass flow rates and two heat inputs.

As shown in Fig. 5, the simulation results exhibit a strong agreement with the experimental measurements at 4.1 kW, with minor deviations across the full range of mass flow rates. At 4.9 kW, the simulation also follows the trend of the experimental data, although the deviation increases at higher flow rates. This discrepancy is expected due to enhanced thermal and radiative losses at elevated input power, which are more difficult to capture precisely in the numerical model. To

Table 2

Thermal properties of materials used in CFD simulation of heat and mass transfer.

Property	Value/correlation	Ref.
SiSiC RPC		
Porosity	$\varepsilon = 0.9$	Manufacturer
RPC Type	$PPI = 10$	Manufacturer
Average Pore Diameter (m)	$d_m = \left(\frac{357}{PPI}\right) \times (5.3022 \times \varepsilon + 2.1549) \times 10^{-5}$	[30]
Extinction Coefficient (m^{-1})	$\beta = (1229.36 - 120.060\varepsilon - 630.674\varepsilon^2)/(1000d_m)$	[30]
Strut Reflectivity	$r = 0.1$ (dimensionless, for SiSiC material)	[31]
Absorption coefficient (m^{-1})	$\alpha = \beta \times (1 - r)$	[30]
Scattering Coefficient (m^{-1})	$\sigma = \beta \times r$	[30]
Specific Heat Capacity ($\frac{J}{Kg \cdot K}$)	0.94	[32]
Thermal conductivity ($\frac{W}{m \cdot K}$)	<div> <div>T (K)</div> <div>K_{SiSiC}</div> </div> <div> <div>293</div> <div>110</div> </div> <div> <div>473</div> <div>85</div> </div> <div> <div>773</div> <div>60</div> </div> <div> <div>1273</div> <div>42</div> </div> <div> <div>1773</div> <div>38</div> </div> <div> <div>2830</div> <div></div> </div>	[33]
Density ($\frac{Kg}{m^3}$)	2830	[32]
Ceria Laminate (CeO_2)		
Specific Heat Capacity ($\frac{J}{Kg \cdot K}$)	$-0.000127 \times T^2 + 0.269765 \times T + 299.8$ (when $T \leq 1100$ K) 444.27 (when $T > 1100$ K)	[34]
Thermal conductivity ($\frac{W}{m \cdot K}$)	$2.2 \times 10^{-7} \times T^2 - 2.8387 \times 10^{-4} \times T + 0.176786$	
Density ($\frac{Kg}{m^3}$)	504.4	
Alumina-Silica ($Al_2O_3 - SiO_2$)		
Specific Heat Capacity ($\frac{J}{Kg \cdot K}$)	$4 \times 10^{-7} \times T^3 - 1.3797 \times 10^{-3} \times T^2 + 1.5987289 \times T + 477.7$ (when $T \leq 1480$ K) 1118.44 (when $T > 1480$ K)	[34]
Thermal conductivity ($\frac{W}{m \cdot K}$)	$0.00012926 \times T + 0.019654$	
Density ($\frac{Kg}{m^3}$)	560.65	
Steel Shell		
Specific Heat Capacity ($\frac{J}{Kg \cdot K}$)	$0.2827 \times T + 327.29$	[34]
Thermal conductivity ($\frac{W}{m \cdot K}$)	$0.0158 \times T + 10.169$	
Density ($\frac{Kg}{m^3}$)	8470	
Aluminium Shield		
Specific Heat Capacity ($\frac{J}{Kg \cdot K}$)	898	COMSOL material library [29]
Thermal conductivity ($\frac{W}{m \cdot K}$)	237	
Density ($\frac{Kg}{m^3}$)	2700	
Air		
Specific Heat Capacity ($\frac{J}{Kg \cdot K}$)	$1047.63657 - 0.372589265 \times T + 9.45304214 \times 10^{-4} \times T^2 - 6.02409443 \times 10^{-7} \times T^3 + 1.2858961 \times 10^{-10} \times T^4$	COMSOL material library [29]
Thermal conductivity ($\frac{W}{m \cdot K}$)	$-0.00227583562 + 1.15480022 \times 10^{-4} \times T - 7.90252856 \times 10^{-8} \times T^2 + 4.11702505 \times 10^{-11} \times T^3 - 7.43864331 \times 10^{-15} \times T^4$	
Density ($\frac{Kg}{m^3}$)	$\frac{p}{R_{sp} \times T}$ (where $R_{sp} = 287 \frac{J}{Kg \cdot K}$)	
Dynamic viscosity ($Pa \cdot s$)	$-8.38278 \times 10^{-7} + 8.35717342 \times 10^{-8} \times T - 7.69429583 \times 10^{-11} \times T^2 + 4.6437266 \times 10^{-14} \times T^3 - 1.06585607 \times 10^{-17} \times T^4$	

quantify the accuracy of the simulation, the Root Mean Square Error (RMSE) was calculated for both cases. The RMSE values were 29.4 K at 4.1 kW and 101.9 K at 4.9 kW. These values quantify the average discrepancy between the simulated outlet temperatures and the experimental measurements. The corresponding temperature deviations ranged from 0 % to 5 % at 4.1 kW and from 0 % to 12 % at 4.9 kW, demonstrating a strong agreement and reliability of the simulation model. The higher RMSE and percentage deviation at 4.9 kW are expected due to stronger thermal gradients and enhanced radiative losses under higher power conditions.

Additionally, error bars were added to the experimental data to represent uncertainty. A ± 5 % error margin was applied for the 4.1 kW case and ± 10 % margin for the 4.9 kW case. These ranges account for measurement tolerances and potential digitisation inaccuracies. Comparable levels of deviation have been reported in similar high-temperature solar receiver studies [35,36]. The validated settings

were therefore used to evaluate the remaining geometrical configurations.

7. Results and discussion

This section systematically analyses the thermal performance of four cavity-type receiver geometries, hexagonal, heptagonal, octagonal, and nonagonal, subjected to two different solar power inputs, namely 4.1 kW and 4.9 kW. Key performance metrics, including air outlet temperature, thermal efficiency, and internal heat distribution, are presented and compared across geometries. To facilitate a robust evaluation, results are further normalised by volume and internal surface area, accounting for differences in receiver size and heat-absorbing surface. The simulation data are interpreted through parametric plots and thermal contour maps, providing comprehensive insights into the radiative and convective heat transfer characteristics associated with each design.

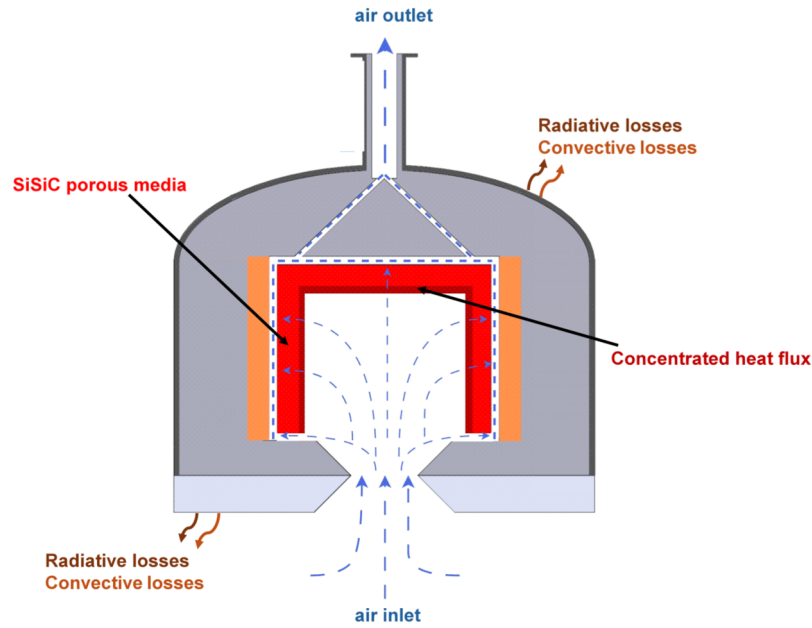


Fig. 3. Schematic illustration of airflow paths and boundary conditions in the porous cavity receiver.

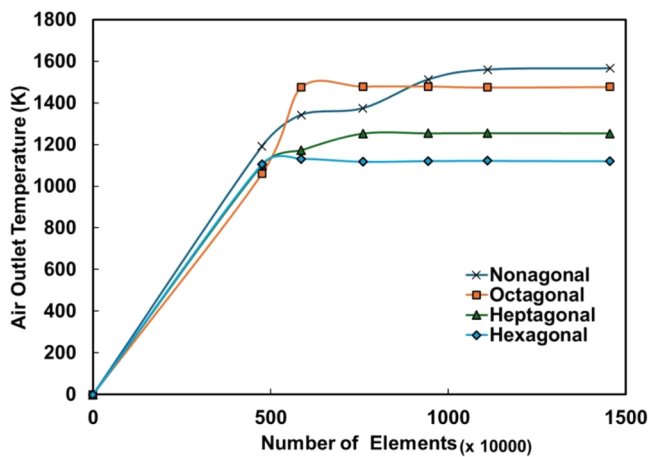


Fig. 4. Mesh independence study for the four receiver geometries.

7.1. Thermal performance analysis at 4.1 kW

Fig. 6 presents a comparative analysis of the thermal performance of four different receiver geometries, hexagonal, heptagonal, octagonal, and nonagonal, under varying air mass flow rates at a fixed solar input of

4.1 kW. Fig. 6(a) illustrates the outlet temperature profiles, while Fig. 6 (b) displays the corresponding heat extracted by the working fluid. The experimental data for the octagonal geometry, previously used for validation, is also included here as a benchmark reference to contextualise the comparative trends. In Fig. 6(a), the nonagonal geometry demonstrates the highest outlet temperatures across the flow range, followed closely by the octagonal geometry. These elevated temperatures are attributed to the compact configuration and reduced internal surface area of the nonagonal and octagonal cavities, which enhance local heat flux density and radiation absorption. The hexagonal and heptagonal geometries, with relatively larger internal areas and more spread-out wall distribution, yield lower outlet temperatures due to more distributed heat absorption and diminished radiative intensity per unit area.

The thermal energy extracted by the air, as depicted in Fig. 6(b), further substantiates these observations. The nonagonal geometry achieves the highest peak heat generation at approximately 3.06 kW, significantly outperforming the octagonal (2.64 kW), heptagonal (2.32 kW), and hexagonal (2.29 kW) designs. The simulation results for the octagonal geometry closely match the experimental reference value (2.69 kW), reinforcing the model's validity and its capacity to predict thermal trends across varying geometries. This progressive increase in thermal performance with the number of faces can be attributed to the geometric influence on internal cavity characteristics. As the number of faces increases, while maintaining a constant radial distance from the

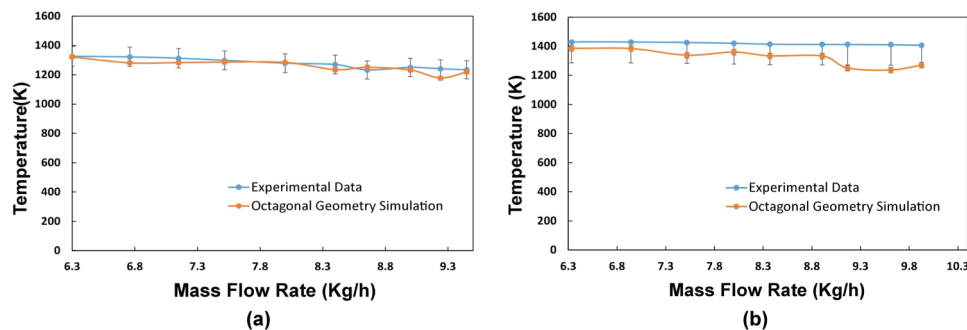


Fig. 5. Comparison between experimental and simulated outlet air temperatures for the octagonal receiver configuration at two solar input powers: (a) 4.1 kW with $\pm 5\%$ error bars, and (b) 4.9 kW with $\pm 10\%$ error bars.

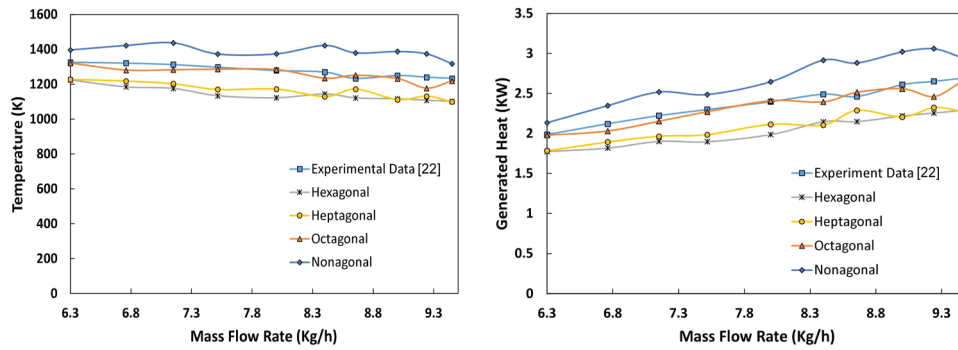


Fig. 6. (a) Outlet temperature profiles and (b) corresponding extracted thermal energy for all the receiver geometries under varying air mass flow rates at a fixed solar input of 4.1 KW.

cavity centre, the internal surface area and volume of the RPC structure decrease. This geometric compactness leads to a higher surface energy density and enhances the local radiative heat flux. Additionally, the increase in the number of edges in the receiver cross-section may contribute to improved interaction between the heat transfer fluid and the RPC surfaces. These additional surfaces and corners likely increase turbulence intensity and mixing within the flow, thereby strengthening the convective heat transfer mechanism. Consequently, the nonagonal geometry outperforms the octagonal, which in turn exceeds the performance of the heptagonal and hexagonal designs, as clearly observed in the generated heat trends in Fig. 6(b).

7.1.1. Thermal efficiency analysis at 4.1 kW

This relationship between all the designs becomes clearer when evaluating the relative gain in thermal efficiency as the number of faces increases. Holding the radius constant, each incremental increase in the number of cavity faces results in improved thermal performance. From the hexagonal to the nonagonal geometry (i.e., from 6 to 9 faces), the peak thermal efficiency increases from 56 % to 75 %, representing a 19 % improvement. Increasing from 6 (hexagonal) to 7 (heptagonal) faces results in a 1 % gain in peak efficiency. Similarly, increasing 7 to 8 (octagonal) results in a gain of 8 %, while increasing 8 to 9 (nonagonal), the gain reaches 10 %. This progressive improvement is not linear but suggests an accelerating benefit as the cavity becomes more compact and edge dense. The efficiency jump from heptagonal to octagonal is more substantial than from hexagonal to heptagonal, and the largest gain is observed between octagonal and nonagonal geometries, illustrated in Fig. 7.

The explanation for this lies in the geometry's influence on radiative

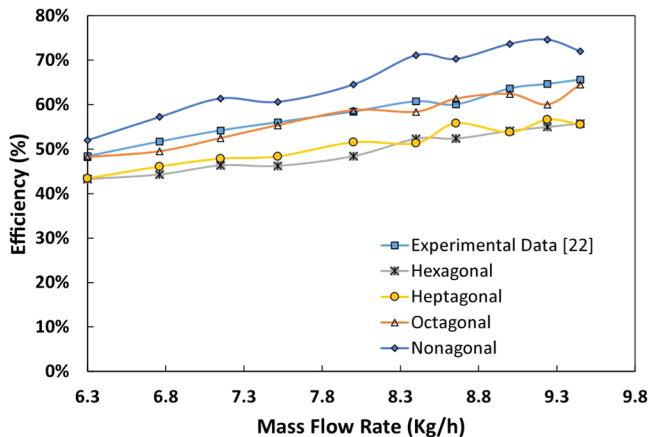


Fig. 7. Efficiency comparison of hexagonal, heptagonal, octagonal, and nonagonal receiver geometries under varying air mass flow rates at a fixed solar input of 4.1 KW.

and convective coupling. As the number of faces increases, the internal surface area and total volume of the cavity decrease slightly, concentrating incident radiation and increasing the surface heat flux density. Additionally, increasing the number of polygonal corners creates more regions of intensified thermal gradients, which can induce local turbulence and enhance convective heat transfer between the RPC walls and the airflow. This trend indicates that receiver designs with a higher number of faces, not merely smoother or larger surfaces, can deliver superior thermal efficiency due to enhanced surface interaction and tighter geometric enclosure of the working fluid. However, it also suggests diminishing returns may occur beyond a certain face count, where the benefits of compactness may be offset by potential fabrication complexity or flow resistance.

7.1.2. Thermal distribution at 4.1 kW

Fig. 8 presents the temperature distribution contours for each receiver geometry, hexagonal, heptagonal, octagonal, and nonagonal, captured at the maximum thermal efficiency condition for a solar input of 4.1 kW. Each geometry is shown in two orthogonal views: XY (cross-sectional front face) and XZ (longitudinal vertical section), offering a full visualisation of the heat propagation throughout the receiver volume. In all geometries, the highest temperatures are observed at the central region of the Reticulated Porous Ceramic (RPC) cavity, where concentrated solar radiation is initially absorbed. This hot zone appears as a deep red region in the XY view and is consistently encased by concentric isothermal layers of decreasing temperature gradients toward the outer insulation, indicated by a shift from red to green and blue. The XZ view reveals the thermal penetration depth into the cavity structure and the flow behaviour of the heated air toward the outlet.

Among the geometries, the nonagonal receiver exhibits the most compact and symmetric high-temperature core, with a peak static temperature approaching 1672 K, and superior thermal uniformity along both radial and axial directions. This behaviour aligns with its previously reported peak thermal efficiency of 75 %, the highest among all tested designs. The octagonal geometry also shows strong performance, with a relatively wide and well-defined hot core, confirming its efficient heat capture and transfer characteristics. In contrast, the hexagonal and heptagonal designs demonstrate more diffuse heat zones, particularly in the XY contours, where the thermal concentration is visibly lower, and the transition from hot to cool zones is more abrupt. This reflects their lower convective coupling and reduced thermal efficiencies of 56 % and 57 %, respectively. Additionally, the XZ contours in these geometries indicate slower heat propagation along the flow path, with cooler zones persisting near the outlet. Overall, the temperature contours substantiate the quantitative findings presented earlier: increasing the number of faces results in more efficient radiative heat absorption and a more uniform distribution of thermal energy within the receiver. This enhanced thermal stratification minimises localised losses and supports a higher heat transfer rate to the working fluid, especially

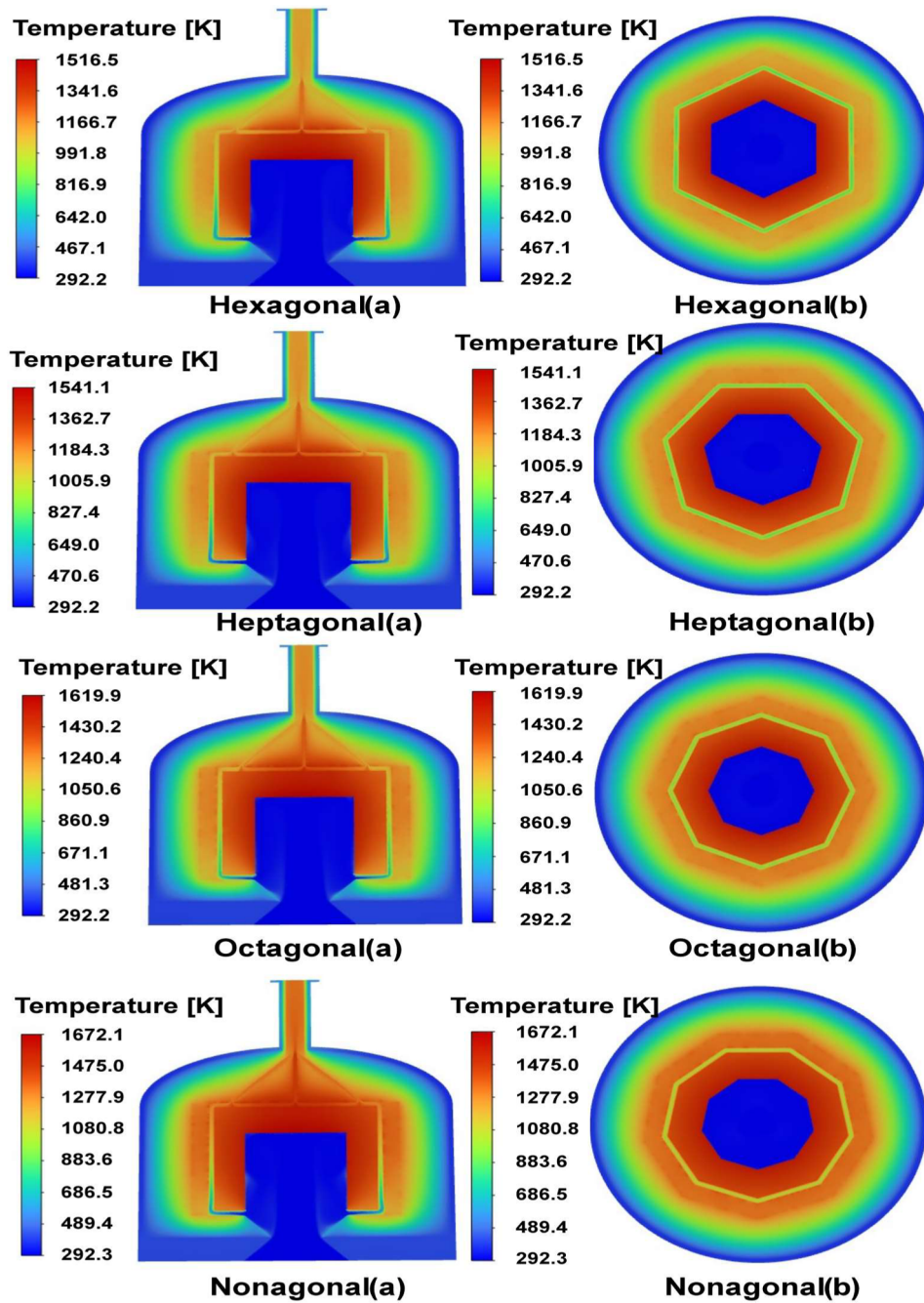


Fig. 8. Temperature contours of all receiver configurations at maximum thermal efficiency under 4.1 kW input. (a) XY cross-section showing lateral gradients. (b) XZ section illustrating axial heat propagation.

evident in the nonagonal and octagonal configurations.

7.1.3. Thermal performance analysis at 4.9 kW

To strengthen the comparative analysis and validate the consistency of observed trends, a second heat input condition of 4.9 kW was simulated. This higher input not only allows for evaluating the thermal performance under increased flux conditions but also provides a more robust basis for confirming the influence of cavity geometry on heat transfer mechanisms. Fig. 9 illustrates the outlet temperature and the total heat extracted by the air across a range of mass flow rates for each cavity geometry. The experimental data for the octagonal receiver is again included as a reference benchmark. As shown in Fig. 9(a), the nonagonal geometry maintains the highest outlet temperature range across most flow conditions, followed by the octagonal, with the

heptagonal and hexagonal trailing behind. Fig. 9(b) further confirms that the nonagonal receiver delivers the highest peak heat generation, reaching 3.56 kW, compared to 2.94 kW for the octagonal, 2.64 kW for the heptagonal, and 2.53 kW for the hexagonal configuration. Similar to the 4.1 kW case, the enhanced thermal performance of the nonagonal and octagonal receivers is attributed to their compact design, which enhances radiative heat absorption and facilitates stronger convective interactions with the heat transfer fluid. However, at 4.9 kW, this geometric advantage becomes even more significant. The steeper thermal gradients introduced by the higher heat flux amplify the sensitivity of heat transfer to the receiver shape. In particular, the nonagonal geometry, with its smaller internal volume and more significant number of flow-interacting edges, is better positioned to convert the input energy into usable heat, as evidenced by its superior heat extraction across all

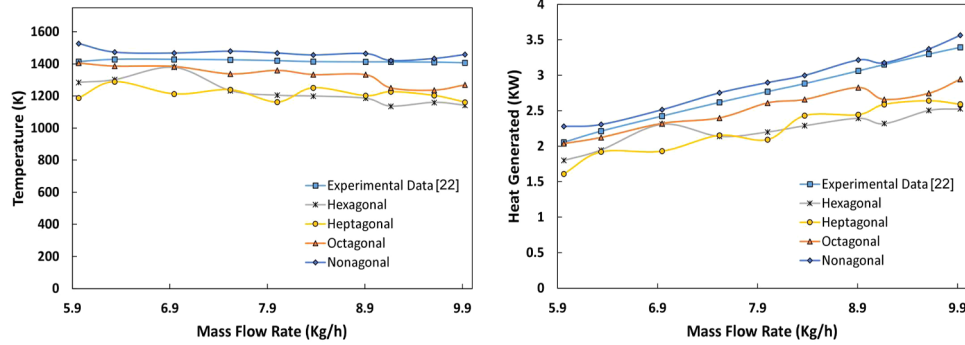


Fig. 9. (a) Outlet temperature profiles and (b) corresponding extracted thermal energy for all tested receiver geometries under varying air mass flow rates at a fixed solar input of 4.9 kW.

flow rates.

7.1.4. Thermal comparing 4.1 KW and 4.9 KW efficiencies

To further consolidate the thermal performance evaluation, a comparison of peak thermal efficiencies at both 4.1 kW and 4.9 kW heat input levels was conducted. Interestingly, while one might expect that increased thermal input would enhance the efficiency due to higher temperature gradients, the results suggest a more nuanced relationship. For the hexagonal, heptagonal, and octagonal geometries, the peak efficiency actually declined as the power input increased, dropping from 56 % to 52 %, 57 % to 54 %, and 65 % to 60 %, respectively. This reduction can be attributed to the geometrical limitations in surface area exposure and convective heat extraction, which may not scale proportionally with the increased radiative heat input, leading to localised thermal saturation and less effective heat transfer to the fluid. The efficiency of each geometry at 4.9 KW can be shown in Fig. 10.

The nonagonal geometry maintained a strong performance across both conditions, with a marginal decrease from 75 % to 73 %. Its superior ability to accommodate increased flux stems from its enhanced surface energy density and improved internal aerodynamics, which support more uniform heat absorption and distribution. This sustained performance reinforces the nonagonal cavity's robustness under varying thermal loads. These observations underscore the importance of optimising geometry for baseline efficiency and stability and adaptability under variable operating conditions, which are crucial considerations for real-world concentrated solar power applications where solar input fluctuates.

7.1.5. Temperature contour analysis at 4.9 kW

The increase in solar input to 4.9 kW further accentuates the thermal

performance differences among the cavity geometries. Fig. 11 captures the temperature contours at each geometry's peak efficiency, using orthogonal views (XY and XZ) to visualise lateral and axial heat distribution. The nonagonal receiver stands out with a sharply defined, symmetric thermal core reaching approximately 1816 K. This concentrated hot zone suggests enhanced radiative absorption and superior internal thermal management, promoting efficient heat transfer to the working fluid. The octagonal configuration also demonstrates strong thermal performance, with temperatures peaking around 1706 K and a stable axial heat profile. The contours reveal deeper penetration and smoother gradients along the flow path, reinforcing its effectiveness in converting incident solar flux into usable thermal energy. Conversely, the heptagonal and hexagonal designs exhibit broader, less focused heat zones. Peak temperatures remain lower, near 1634 K and 1582 K, respectively, accompanied by weaker axial thermal penetration and less distinct core regions. These characteristics reflect lower energy density and suboptimal radiative interaction due to their geometric spread. Compared with the 4.1 kW case, the elevated input intensifies heat flux disparities, widening the performance gap among geometries. The results confirm that increasing the number of receiver faces not only enhances radiative concentration but also promotes more efficient thermal distribution, most notably in the nonagonal and octagonal cases.

Comparing four geometrical configurations, hexagonal, heptagonal, octagonal, and nonagonal, under controlled conditions, the results revealed how internal geometry affects heat transfer characteristics, energy absorption, and outlet temperature. The nonagonal receiver consistently outperformed the others, achieving the highest thermal efficiency at both input levels (75 % at 4.1 kW and 73 % at 4.9 kW), attributed to its compact structure, higher surface energy density, and stronger convective mixing facilitated by the increased number of internal edges. Moreover, by keeping the aperture radius constant while increasing the number of faces, the internal volume and absorbing surface area are reduced, thereby concentrating the incoming solar energy over a smaller area. This geometric compactness leads to greater localised heat flux, enhancing radiative absorption and further improving thermal performance.

In addition, it should be noted that this study is limited to small-scale receiver models and does not address economic viability or scalability to large-scale CSP systems. The primary focus is on analysing the thermal performance impact of cavity geometry, particularly the number of faces. Future research may consider the influence of scale and assess the economic feasibility of implementing such geometries in commercial-scale receivers. This study focuses on the influence of face number under idealised conditions and uniform solar flux. It does not account for optical losses, structural constraints, or economic factors. These limitations can be addressed in future work to assess the practical scalability of the proposed designs.

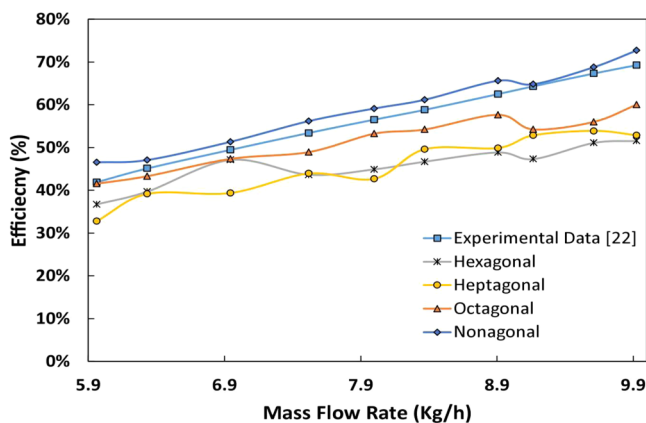


Fig. 10. Variation in thermal efficiency with air mass flow rate for all tested receiver geometries at a constant solar input of 4.9 kW.

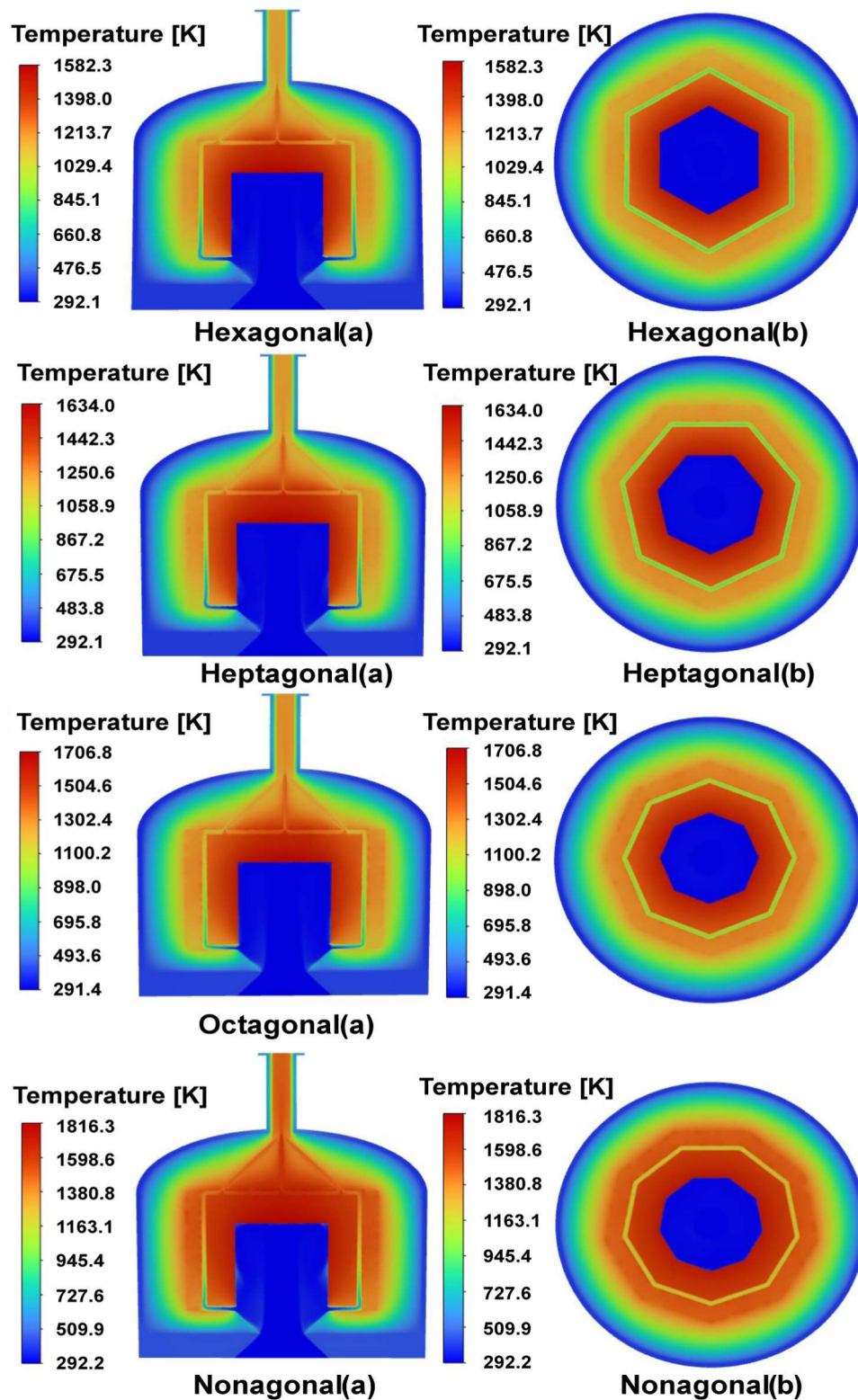


Fig. 11. Temperature distribution contours for each receiver geometry at 4.9 kW solar input: (a) XY cross-sectional view, (b) XZ longitudinal view.

8. Conclusion

This study conducted a detailed numerical investigation of volumetric solar receivers with four different polygonal cavity geometries: hexagonal, heptagonal, octagonal, and nonagonal, under two solar input conditions (4.1 kW and 4.9 kW). Using validated CFD simulations incorporating the Monte Carlo radiation model, the analysis focused on

key thermal performance indicators, including outlet temperature, heat absorption, and thermal efficiency. Among the evaluated configurations, the nonagonal cavity consistently outperformed the others, delivering the highest outlet temperatures and peak thermal efficiencies across all tested conditions. This enhanced performance is attributed to the geometric optimisation of the cavity, which increases the number of internal edges and promotes greater surface energy density. The

integration of reticulated porous ceramics (RPCs) further amplified heat transfer, acting as an efficient medium for both radiative absorption and convective exchange. The open-cell structure and extended surface area of the RPC enabled deeper thermal penetration, more uniform airflow, and improved dispersion, particularly in geometries with higher face counts where flow recirculation and turbulence are naturally intensified. These findings demonstrate the synergistic potential of cavity geometry and porous media design. Increasing the number of cavity faces, when combined with an effective RPC structure, significantly improves both radiative and convective heat transfer, enabling higher efficiency and more compact receiver designs suitable for high-flux CSP applications. Future work should investigate the influence of porous media parameters such as pore density, layer thickness, and material composition on both thermal performance and mechanical reliability under dynamic operating conditions. This approach will further support the development of high-efficiency, next-generation CSP receivers tailored for thermochemical and power generation applications.

CRedit authorship contribution statement

Abdullah Ayed Alrwili: Writing – original draft, Validation, Software, Resources, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Qing Qin:** Writing – review & editing, Supervision, Software. **Khalifa Aliyu Ibrahim:** Writing – review & editing, Supervision, Methodology, Formal analysis. **Peter King:** Supervision. **Thomas Delaney:** Supervision. **Ahmed Ali Aldubayyan:** Writing – review & editing, Software. **Yousef Lafi A Alshammari:** Writing – review & editing. **Zhenhua Luo:** Supervision, Software, Investigation.

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.

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Data availability

Data will be made available on request.

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Geometric optimisation of volumetric solar receivers: a study of polygonal cavity configurations

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