

Geometrical comparison of inline and staggered stack wire mesh absorbers for solar volumetric receivers

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ABSTRACT

Open volumetric receivers using air as the heat transfer fluid can operate at higher temperatures and thermal efficiencies than the current state of the art in central receiver systems. Optimising their design requires detailed understanding of two critical operational attributes: the heat transfer coefficient (HTC) and the pressure drop. This work examines these attributes in two dense wire mesh absorber arrangements, inline (IL) and staggered (ST), with single-mesh porosities of 80 % and wire diameters of 0.7, 0.4 and 0.1 mm. A 2D porous model or homogeneous equivalent model (HEM) with local thermal non-equilibrium (LTNE) is developed, using a flux density of 600 kW/m² and an air inlet velocity of 1 m/s. The model is experimentally validated and subsequently used to evaluate the generalised correlations (GCs) obtained for the thermo-fluid-dynamic attributes with those derived from the individual correlations (ICs). In addition, the model assesses the impact of the different wire diameters and arrangements on thermal and fluid flow performance. The findings reveal that the ST arrangement generally outperforms the IL arrangement in thermal efficiency, except for the 0.1 mm wire diameter. Conversely, the IL arrangement demonstrates superior hydrodynamic performance. Finally, the results are corroborated with existing literature, which further validates the reliability of the numerical model and its conclusions.

1. Introduction

Within the past few years, concentrated solar power technologies have reduced their levelised-cost of electricity down to 0.118 USD/kWh, a 69 % decline from 2010. However, concentrated solar power technologies need to reduce the initial investment as well in order to be attractive to the industry. For central receiver systems (CRS) or concentrated solar tower (CST), the total installed costs are 6,354 USD/kWh with the solar field being the largest driver of cost, followed by the power block and the receiver [1]. Open volumetric receivers (OVRs) offer the best prospects for achieving cost reduction through increased operating temperatures among current receiver technologies [2]. The OVR, heated by the reflected rays of the heliostat, transfers energy to air being drawn through it. This heated air can be used in heat processes or electrical generation [3,4].

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OVRs have been typically made of ceramic and metals. In the early stages, the OVRs evaluated were plain and knitted wire meshes, corrugated metal foil named “Emitec”, SiC honeycombs and ceramic foams [5–8]. The OVR technology envisaged an air outlet temperature of 1,000 °C in order to couple the volumetric receivers with highly efficient power cycles. However, the results presented during this early evolution of the technology had air outlet temperatures below this target temperature. As a result, newer absorbers were mainly made of ceramic material, due to its capability to withstand higher temperatures. Some of the best ceramic volumetric absorbers developed are covered in the studies summarised below.

In 1998, Karni et al. [9] presented a novel receiver nicknamed the “Porcupine” made mostly of alumina-silica (Al_2O_3). In the study, four configurations were analysed at a solar furnace, concluding that the “Porcupine” withstood flux densities higher than honeycombs and foam matrices. The “Porcupine” outperformed them under similar operating conditions, around 79 % thermal efficiency at 800 °C gas temperature at the outlet, and allowed wider working conditions. In 1999, Hoffschmidt et al. [10] developed the HiTRec, a SiSiC cup with a SiC honeycomb structure of 49.5 % porosity as absorber. This receiver was tested at the Plataforma Solar de Almeria (PSA) reaching air outlet temperatures near 1,000 °C. In 2003, Hoffschmidt et al. [11] presented HiTRec II, an evolution of HiTRec, which was designed to be manufactured in mass. HiTRec II was tested at PSA and withstood 700 °C average air temperature at the outlet, presenting a $76 \% \pm 7 \%$ thermal efficiency. In 2004, Fend et al. [12] presented two novel absorbers with high porosity. A ceramic foam combining two foam materials of distinct cell density and a cellular SiC receiver with parallel channels. Both absorbers met the general requirements for use as high-performance solar receivers. Also, in the same year, Fend et al. [13] presented a comparison between different mixtures of absorber structures, such as, “Fraunhofer IKTS” foam ceramics, SiC fiber mesh, the cellular SiC receiver with parallel channels [12] and the “Emitec” metal foil absorber [7]. It was concluded that there was potential in mixing different structures, and more importantly, this study provided a basis for numerical simulations. Later on, many other studies have evaluated volumetric absorbers, mostly ceramic absorbers, through numerical simulations under the local thermal non-equilibrium (LTNE) state [14–20], obtaining outcomes closer to the experimental results.

Recent studies have sparked interest in metallic OVRs, as advances in additive manufacturing techniques have facilitated intricate designs with the availability of new materials [17,21–24]. Some of the most recent metal volumetric absorbers experimentally evaluated are covered in the studies summarised below.

In 2017, Pabst et al. [21] developed a new metal OVR based on pairs of planar and corrugated metal sheets connected by soldered joints at specific points, called MetRec (Fig. 1a). Moreover, the authors modified the absorbers by cutting the plane and corrugated sheets to enhance radial flow and turbulence. Then, the MetRec absorber was thermally evaluated in laboratory scale and in 500 kW scale tests, showing an improvement of around 5 % with respect the state-of-the-art absorber. In the same year, Capuano et al. [25] presented a new OVR made of a titanium-aluminium alloy. The concept included an absorber with two different geometries, a front one made of pins and a rear one following a honeycomb structure (Fig. 1b). The absorber was 3D numerically simulated and also tested at lab scale. The experimental tests validated the numerical simulations with good agreement. The authors concluded that the new OVR improved thermal efficiency by 12 % compared to the state-of-the-art absorber. In 2018, Luque et al. [22] published the results of

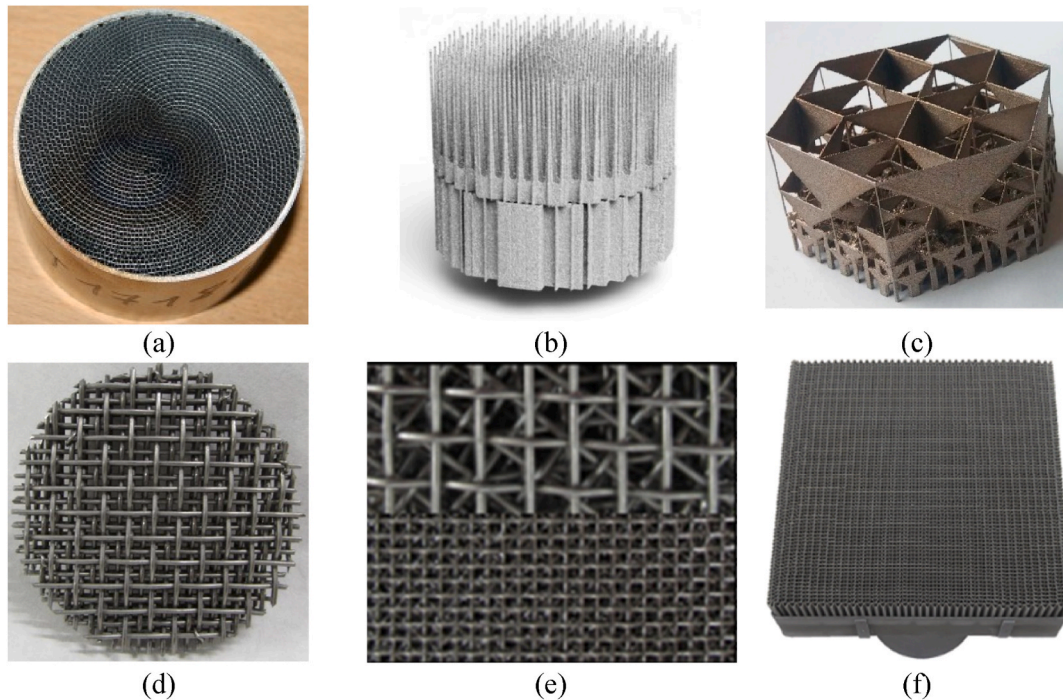


Fig. 1. Metal volumetric absorbers tested experimentally in recent studies: (a) MetRec [21], (b) Ti6Al4V demonstrator [25], (c) Fractal-like [22], (d) Staggered stack wire meshes [26,30], (e) Axial varied porosity [28], (f) Emitec/EmiRec demonstrator [24,29].

a new OVR made of stainless steel and following a hierarchically-layered fractal-like concept (Fig. 1c). This concept demonstrated that the absorbers were able to produce the volumetric effect, shifting the radiation absorption and enhancing the convective heat transfer through its intricate geometry. However, the solar to thermal conversion efficiency had the same order as in state-of-the-art geometries. In the same year, Avila-Marin et al. [26,27] analysed 26 dense wire mesh OVR with single (Fig. 1d) and gradual porosities (Fig. 1e) made of 310-alloy. The results were compared with state-of-the-art absorbers showing that 6 absorbers had similar or even better results than the baseline receivers. These experimental results were used to validate computational fluid dynamic (CFD) models using staggered stack wire mesh OVRs with different configurations [23,28]. In 2022, Broeske et al. [29] developed a 1D continuum model to analyse the thermal performance of three volumetric receiver geometries. The HiTRec which is the reference SiC volumetric absorber [11], the StepRec which is based on the development by Capuano [25], but built out of SiSiC, and the Emitec absorber made of a steel alloy with smooth and curled steel sheets (Fig. 1f) similarly to that proposed by Pabst [21]. The 1D model was validated against 3D simulations, and it was concluded that the StepRec and the Emitec absorbers performed better than the HiTRec, the StepRec being the best one in the different cases studied. Later, in 2023, Broeske et al. [24] carried out an experimental test campaign of these new absorbers in the Synlight solar simulator, concluding that the new designs outperform the HiTRec performance, the Emitec (called EmiRec in this article) having the highest recorded efficiency for 300 cpsi with a 91.8 %.

The present work is focused on the comparison of the two extreme arrangements, inline and staggered, of woven wire meshes defined by the geometrical parameters. The two arrangements have contrasting thermo-fluid-dynamic behaviour. The inline arrangement, with its higher porosity and greater radiative penetration has a lower convective heat transfer coefficient and lower hydrodynamic resistance, while the staggered arrangement with lower porosity and less radiative penetration has a higher convective heat transfer coefficient and higher hydrodynamic resistance. Thus, the aim of this study is to evaluate the inline and the staggered arrangements with 0.7, 0.4 and 0.1 mm wire diameters and 80 % porosity to seek the optimal geometrical parameters and configuration. Both arrangements have not been comprehensively evaluated in the literature before and the comparison could guide future developments and optimisations of wire mesh absorbers.

This research exhibits a comparison based on the thermo-fluid-dynamic attributes, the convective heat transfer coefficient (HTC) and the pressure drop, for each arrangement, inline (IL) and staggered (ST), and each wire diameter presented. The comparison of each attribute is made through individual and generalised correlations [30–32] summarised in sections 2.1, 2.2 and 2.3. The results implemented a homogeneous equivalent model (HEM), which is validated as shown in section 4.1.1, and later used to: i) corroborate the generalised correlations with the individual correlations, explained in section 4.1.2.1, ii) evaluate the influence of the wire diameter (0.7, 0.4, 0.1 mm) and arrangement (staggered and inline) on the temperature profiles and thermal and hydrodynamic performance of OVRs, detailed in sections 4.1.2.2 and 4.1.2.3, and iii) in section 4.2, the ST results obtained are compared with similar one found in the literature.

2. Characterisation of the inline and staggered arrangements

The inline (IL) and staggered (ST) stacks studied in this work have quite distinct geometrical and thermo-fluid-dynamic attributes [33,34]. Fig. 2 depicts both patterns, IL and ST, which are defined by the following geometrical parameters: the wire diameter (d), the spacing between the wires or pitch (w) and the mesh count ($M = 1/(w + d)$) [35]. The difference in the structural pattern is based on

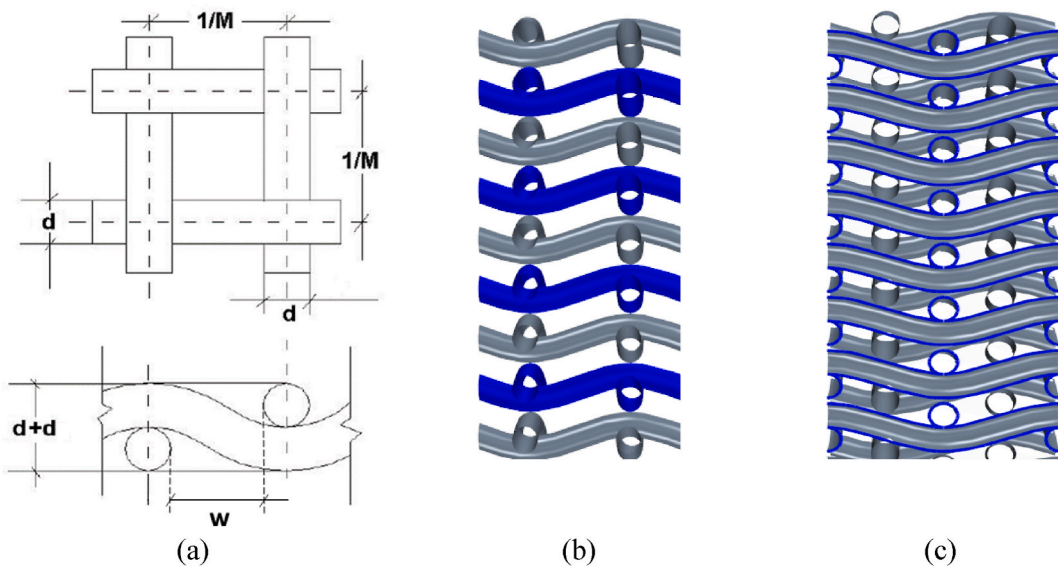


Fig. 2. (A) Top and frontal view of a single screen woven wire mesh, (b) frontal view of an IL square woven wire mesh lattice and (c) frontal view of an ST square woven wire mesh lattice. Odd meshes are outlined in blue.

the meshes, for the IL pattern, Fig. 2b, the meshes are aligned perfectly, while, for the ST pattern, Fig. 2c, the position of the uneven meshes is displaced a value of $1/(2M)$ regarding the even meshes [36]. Additionally, these geometrical parameters are used to defined geometrical properties in dense wire meshes such as the specific surface area, the porosity and the hydraulic diameter as detailed in the literature [37].

In previous studies [38–41], the characterisation of the IL and ST patterns was established considering three geometrical properties: the porosity (ϕ), the specific surface area (a_v) and the hydraulic diameter (d_h) which are based on the geometrical parameters, d and w .

The first part of this work exhibits the main thermo-fluid-dynamic attributes, the convective HTC at thermal equilibrium and the pressure drop, of the IL and ST patterns of wire meshes with a fixed 80 % single-mesh porosity and wire diameters from 0.1 to 0.7 mm, based on published studies [30–32]. Those studies, already simulated using 3D detailed models, presented individual correlations (ICs), one for each wire diameter, arrangement and thermo-fluid-dynamic attribute, and generalised correlations (GCs), one for each thermo-fluid-dynamic attribute and arrangement. The main advantage offered by the generalised correlations is in OVR optimisation. Section 2.1 presents the main coefficients of the ICs and GCs that estimate the thermo-fluid-dynamic attributes of the IL arrangement, and section 2.2 presents analogous results for the ST arrangement. The comparison between the arrangements in terms of thermo-fluid-dynamic attributes is shown in section 2.3. The structure of the results, section 4, was described at the end of the Introduction.

2.1. Inline pattern

In the inline arrangement, the porosity of the stack is defined by the single-mesh porosity as each mesh is aligned with the one below, resulting in a compactness factor, c_f , of 1 (Fig. 2b). The cases studied here had wire diameters of 0.7, 0.4 and 0.1 mm, all with 80 % porosity.

2.1.1. Individual correlations (ICs)

ICs generate predicted values of the two thermo-fluid-dynamic attributes, convective HTC and pressure drop, for each wire diameter analysed.

The HTC is expressed by the Nusselt number, which is associated with the Reynolds and Prandtl numbers. This correlation is shown in Eq. (1), where a , b and c are constants that depend on the geometry, mainly characteristics d and w [42], and the Reynolds number is inferred through the superficial velocity at the inlet of the volumetric absorber.

$$Nu = a \cdot Re^b \cdot Pr^c \quad (1)$$

And, the ICs that estimate the $\Delta P/L$ are based on Forchheimer-extended Darcy's law as presented in Eq. (2), where K_1 and K_2 are the viscous and the inertial permeability coefficients respectively [43].

$$\frac{-dp}{dx} = \frac{\mu}{K_1} \cdot v + \frac{\rho}{K_2} \cdot v^2 \quad (2)$$

Table 1 presents the constants (a , b , c) that with Eq. (1) yield the value of the convective HTC for each wire diameter considered. And, Table 2 shows K_1 and K_2 coefficients that with Eq. (2) predict the $\Delta P/L$ for each wire diameter considered [32].

2.1.2. Generalised correlations (GCs)

The GCs describe the convective HTC and the pressure drop of the fluid across a range of wire diameters with a constant 80 % single-mesh porosity. Regarding the GC for the convective HTC, the main assumptions implemented were:

- A dimensionless geometrical factor, F , added to Eq. (1). This geometrical factor was defined by " $a_v d_h$ ".
- The Pr number was treated as a constant, with a value of 0.7, and inserted in coefficient " m_1 " in Eq. (3).
- The Reynolds number implemented was obtained through the superficial velocity at the inlet of the volumetric absorber.

Eq. (3), based on these assumptions, yields the value of the convective HTC for wire diameters, from 0.1 to 0.7 mm with 80 % single-mesh porosity.

$$Nu_{lv} = (m_1) \cdot Re^{m_2} + F^{m_3} \quad (3)$$

Eq. (4) presents the constants of the convective HTC implemented in the GC (Eq. (3)) for the IL pattern with the geometrical parameters and properties considered in the study [32]. In addition, the range of validity of the Reynolds number was 1–222, and, specifically for this pattern and single-mesh porosity, the dimensionless geometrical factor, F , had a value of 3.19.

Table 1

Constants (a , b , c) of the Nusselt in each IC for the IL arrangement with 80 % single-mesh porosity [32].

Case	d , mm	"a"	"b"	"c"	Re range	Pr range
I1	0.7	8.284	0.2665	1.423	6–222	0.69–0.71
I2	0.4	10.01	0.1031	0.3852	4–133	
I3	0.1	9.525	−0.00044	−0.326	1–31	

Table 2

Viscous (K_1) and inertial (K_2) permeability coefficients in each IC for the IL arrangement with 80 % single-mesh porosity [32].

Case	d, mm	K_1 , m ²	K_2 , m
I1	0.7	2.50E-05	6.72E-03
I2	0.4	3.85E-08	5.38E-03
I3	0.1	2.51E-09	1.21E-03

$$Nu_{lv} = 0.00875 \cdot Re^{1.352} + F^{2.051} \quad (4)$$

The $\Delta P/L$ of the fluid in an OVR is usually evaluated by means of the Forchheimer-extended Darcy's law as defined in Eq. (2). The influence of the geometrical parameters in the $\Delta P/L$ was added to Eq. (2) resulting in the GC defined by Eq. (5), a modification of Ergun equation. The GC yields the values of the $\Delta P/L$ for the IL dense wire mesh stacks with wire diameters from 0.1 to 0.7 mm and 80 % single-mesh porosity [32].

$$-\frac{dp}{dx} = 67.95 \cdot M^2 \cdot \mu \cdot v + 0.3274 \cdot M \cdot \rho \cdot v^2 \quad (5)$$

2.1.3. Corroboration of the GCs and the ICs with the CFD results

This section evaluates the accuracy of the IC and GC methods for an IL pattern with $d = 0.4$ mm, using CFD results as a reference. Tables 3 and 4 summarise the convective HTC and $\Delta P/L$ outcomes for three different velocities, based on: (i) results from detailed 3D numerical simulations, (ii) IC results, and (iii) GC results. The coefficients used to calculate HTC values for the IC and GC methods are listed in Table 1 and Eq. (4) respectively. For the $\Delta P/L$ outcomes, the coefficients for the IC method are provided in Table 2, while the GC method and its associated coefficients are outlined in Eq. (5). Appendix 1 and Appendix 2 shows similar results for $d = 0.1$ mm and $d = 0.7$ mm.

The results of the convective HTC obtained with the ICs and GC are close to the results of the convective HTC obtained with the 3D numerical simulation, presenting similar deviations. The average deviations for ICs and the GC to the 3D numerical simulation are 4.4 and 3.2 %. Consequently, the ICs and the GC have good agreement with the CFD results for the IL pattern. Appendix 1 presents the results for the remaining IL cases, for case I1 (IL-0.7 mm), the average deviations were 9.3 and 5.6 % with the IC and GC respectively, and, for case I3 (IL-0.1 mm), were 0.4 and 3.6 % with the IC and GC respectively.

The $\Delta P/L$ results obtained with the ICs and GC show good agreement with the $\Delta P/L$ results obtained with the CFD. The IC with a deviation of 0.5 % is closer to the CFD results than the GC with an average deviation of 2.4 %. Appendix 2 shows the values of $\Delta P/L$ for the IL remaining cases. For case I1 (IL-0.7 mm), the average deviations were 20.5 and 6.2 % with the IC and GC respectively, and, for case I3 (IL-0.1 mm), were 0.8 and 1.3 % with the IC and GC respectively.

2.2. Staggered pattern

When perfectly stacked as shown in Fig. 2c, the ST pattern for meshes with 80 % single-mesh porosity achieves a volumetric porosity of 65 % obtained with STAR-CCM + [31,30,44]. For the three wire diameters studied, 0.7, 0.4 and 0.1 mm, the resulting compactness factors are 0.56, 0.57 and 0.58, respectively.

2.2.1. Individual correlations (ICs)

ICs predict each thermo-fluid-dynamic attribute for the three wire diameters considered: 0.7, 0.4 and 0.1 mm. The ICs for the convective HTC in the staggered arrangement follow Eq. (1), with Table 5 presenting the constants (a, b, c) associated with each wire diameter considered [31].

Likewise, the ICs of the $\Delta P/L$ follow the Forchheimer-extended Darcy's law presented in Eq. (2). Table 6 presents the viscous and inertial permeability coefficient, K_1 and K_2 respectively, which estimate the $\Delta P/L$ for each wire diameter considered [30,44].

2.2.2. Generalised correlations (GCs)

In this case, the GCs predict the convective HTC and the pressure drop of the fluid in the ST arrangement of wire meshes with wire

Table 3

Convective HTC obtained with the CFD simulation, the IC and the GC for case I2 (IL-0.4 mm).

v m/s	CFD			IC			GC		
	h_{lv} , kW/m ³ ·K			h_{lv} , kW/m ³ ·K			h_{lv} , kW/m ³ ·K		
	700 K	1100 K	1500 K	700 K	1100 K	1500 K	700 K	1100 K	1500 K
1	229	317	408	244	321	393	234	316	403
3	258	339	423	270	357	437	278	337	415
5	312	386	457	280	371	453	339	365	428
Average deviation, %					4.4			3.2	

Table 4Numerical $\Delta P/L$ results obtained with the CFD simulation, the IC, and the GC for I2 (IL pattern with 0.4 mm wire diameter).

v m/s	I2		
	CFD	IC	GC
	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m
1	700	699	702
3	3,412	3,408	3,517
5	7,878	7,868	8,211
Average deviation, %	–	0.5	2.4

Table 5

Constants (a, b, c) of the Nusselt for each IC following ST arrangement with 80 % single-mesh porosity.

Case	d, mm	“a”	“b”	“c”	Re range	Pr range
S1	0.7	12.15	0.3884	3.059	3–100	0.69–0.71
S2	0.4	11.51	0.1425	1.055	2–58	
S3	0.1	6.359	–0.04724	–1.486	0.5–15	

Table 6Viscous (K_1) and inertial (K_2) permeability coefficients for each IC following ST arrangement with 80 % single-mesh porosity.

Case	d, mm	K_1 , m ²	K_2 , m
S1	0.7	2.21E-08	1.16E-03
S2	0.4	7.35E-09	6.57E-04
S3	0.1	3.87E-10	1.86E-04

diameters from 0.1 to 0.7 mm and 80 % single-mesh porosity. The GCs followed the same methodology explained in section 2.1.2.

Concerning the GC for the convective HTC, the dimensionless geometrical factor, F , had a value of 2.57 and, the same assumptions applied in section 2.1.2 were implemented in the ST arrangement as well. Eq. (6) presents the constants of the convective HTC implemented in the GC (Eq. (3)) for the ST pattern with the geometrical parameters and properties considered in the study [31]. In addition, the range of validity of the Reynolds number is 0.5–100.

$$Nu_{bv} = 0.015477 \cdot Re^{1.519} + F^{2.457} \quad (6)$$

The GC used to estimate the $\Delta P/L$ of the fluid through the wire meshes with ST arrangement and 80 % single-mesh porosity is defined with Eq. (7) [30,44], a modification of Ergun equation based on the Forchheimer-extended Darcy's law.

$$-\frac{dp}{dx} = 425.04 \cdot M^2 \cdot \mu \cdot v + 2.292 \cdot M \cdot \rho \cdot v^2 \quad (7)$$

2.2.3. Corroboration of the GCs and the ICs with the CFD results

This section evaluates the accuracy of the IC and GC methods for an ST pattern with $d = 0.4$ mm, using CFD results as a reference. Tables 7 and 8 summarise the convective HTC and $\Delta P/L$ outcomes for three different velocities, based on: (i) results from detailed 3D numerical simulations, (ii) IC results, and (iii) GC results. The coefficients used to calculate HTC values for the IC and GC methods are listed in Table 5 and Eq. (6) respectively. For the $\Delta P/L$ outcomes, the coefficients for the IC method are provided in Table 6, while the GC method and its associated coefficients are outlined in Eq. (7). Appendix 1 and Appendix 2 shows similar results for $d = 0.1$ mm and $d = 0.7$ mm.

The results of the convective HTC obtained with the ICs and GC are close to the results of the convective HTC obtained with the 3D

Table 7

Convective HTC obtained with the CFD simulation, the IC and the GC for S2 (ST-0.4 mm).

v m/s	CFD			IC			GC		
	h_{lv} , kW/m ³ ·K			h_{lv} , kW/m ³ ·K			h_{lv} , kW/m ³ ·K		
	700 K	1100 K	1500 K	700 K	1100 K	1500 K	700 K	1100 K	1500 K
1	1,011	1,388	1,773	1,073	1,397	1,677	1,059	1,430	1,822
3	1,168	1,518	1,878	1,243	1,610	1,937	1,311	1,527	1,859
5	1,523	1,782	2,102	1,332	1,721	2,061	1,694	1,698	1,941
Average deviation, %					4.8			5.3	

Table 8Numerical $\Delta P/L$ results obtained with the CFD simulation, the IC, and the GC for S2 (ST pattern with 0.4 mm wire diameter).

v m/s	S2		
	CFD	IC	GC
	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m
1	4.5	4.3	4.6
3	23	24	24
5	58	57	57
Average deviation, %	–	2.6	2.6

numerical simulation, presenting similar deviations. The average deviations for ICs and the GC to the 3D numerical simulation are 4.8 and 5.3 %. Consequently, the ICs and the GC are in good agreement with the CFD results for the ST pattern. [Appendix 1](#) presents the results for the remaining ST cases, for case S1 (ST-0.7 mm), the average deviations were 12.9 and 7.7 % with the IC and GC respectively, and, for case S3 (ST-0.1 mm), were 1.3 and 4.7 % with the IC and GC respectively.

The $\Delta P/L$ results obtained with the ICs and GC show good agreement with the $\Delta P/L$ results obtained with the CFD. Both ICs and GC present the same 2.6 % average deviation. [Appendix 2](#) shows the values of $\Delta P/L$ for the ST remaining cases. For case S1 (ST-0.7 mm), the average deviations were 1.0 and 2.5 % with the IC and GC respectively, and, for case S3 (ST-0.1 mm), were 0.5 and 0.4 % with the IC and GC respectively.

2.3. Summary of the pattern matching and uncertainty in correlations

A summary of the outcomes gather from literature [[30–32](#)] is presented in this subsection. The significance of these results are a foundation for the results presented in section 4.

2.3.1. Convective HTC

Analysing the two mesh patterns, the convective HTC for the ST pattern is approximately 4.5 times higher than for the IL pattern for each wire diameter. Furthermore, the convective HTC decreases significantly with the increment of the wire diameter, irrespective of the pattern. Specifically, it reduces by a factor of 12–16 times as the wire diameter increases from 0.1 to 0.4 mm, an approximately 3 times when the wire diameter increases to 0.7 mm. In summary, the ST pattern has a higher convective HTC compared to the IL pattern, and smaller wire diameters yield higher convective HTCs regardless the pattern.

Considering all the cases studied, including those presented in [Appendix 1](#), for both arrangements, the average deviations are less than 13 % for the ICs and less than 8 % for the GCs. The key difference between the two correlations lies in their methodology: ICs calculate the convective HTC for each wire diameter and configuration using a literature-based correlation (Eq. (1)), whereas GCs utilise a modified correlation specifically developed for wire meshes, including the geometrical parameters d and w (Eq. (3)). For the IL pattern, the GC to calculate the convective HTC is Eq. (4) and, for the ST pattern, the GC is Eq. (6).

The detailed results for the other two cases are provided in [Appendix 1](#). Overall, the findings indicate that either correlation can reliably predict the convective HTC.

2.3.2. $\Delta P/L$

The evaluation of the CFD $\Delta P/L$ outcomes depends on the arrangement, and it reveals that the $\Delta P/L$ is higher for the ST pattern than for the IL pattern, with values approximately 6.5–7 times higher across all wire diameters. Furthermore, as the wire diameter increases from 0.1 to 0.4 mm, the $\Delta P/L$ decreases between 7 and 13 times, regardless of the pattern, and, it further decreases by a factor of 2–2.5 times when the wire diameter increases to 0.7 mm. In summary, the ST pattern has a higher $\Delta P/L$ compared to the IL pattern, smaller wire diameters result in higher $\Delta P/L$, irrespective of the pattern.

Considering the cases studied, including those in [Appendix 2](#), the $\Delta P/L$ results from the ICs and GCs closely align with the CFD simulations outcomes. The exception was case I1, where the IC showed a larger average deviation (20.5 %) compared to the GC (6.2 %). At low velocities, the deviations are larger for the ICs than the GCs because ICs define the $\Delta P/L$ for each wire diameter and configuration using a literature-based correlation (Eq. (2)). However, the GCs are yielded on a modified correlation developed specifically for wire meshes and the geometrical parameters that defined each case, d and w (Eq. (5) for the IL pattern and Eq. (7) for the ST pattern). The detailed results for the other two cases are presented in [Appendix 2](#).

3. Numerical methodology

The HEM is a numerical methodology used to analyse the complex physics in a volumetric absorber. This methodology takes into account the radiation, convection and conduction mechanisms together with the fluid flow within an absorber. The accuracy of the method is directly related to the accuracy of the effective properties implemented in the porous model. The HEM has been adopted and validated in the literature [[45,46](#)]. This research implements the HEM with the commercial software STAR-CCM + v18.02.008, which was used in a CETA Supercomputer. The cluster partition used has 5 nodes of 36 processors each, clock speeds of 3.1 GHz and 192 GB of memory per node to carry out this work.

3.1. Physics of the porous medium method

The physics of a volumetric absorber or porous region are described by the following equations [47–49]:

- a) Mass conservation equation for the porous region in steady state conditions:

$$\nabla \cdot (\rho \cdot \mathbf{v}) = 0 \quad (8)$$

- b) Momentum equation for the porous region in steady state conditions:

$$\frac{1}{\varnothing^2} \nabla \cdot (\rho \cdot \mathbf{v} \cdot \mathbf{v}) = -\nabla \langle P_f \rangle^f + \frac{\mu}{\varnothing} \cdot \nabla^2 \mathbf{v} - \left[\frac{\mu}{K_1} - \frac{\rho}{K_2} [\mathbf{v}] \right] \cdot \mathbf{v} \quad (9)$$

where ρ is the density of the fluid, $\mathbf{v}$ is the superficial velocity, $\varnothing$ is the volumetric porosity, P_f is the differential pressure of the fluid, μ is the viscosity of the fluid, and K_1 and K_2 are the viscous and inertial permeability coefficients defined in the main correlations and tables presented in section 2.

- c) Energy equations for the fluid and porous regions in steady state conditions:

The local thermal non-equilibrium (LTNE) model accounts for thermal interactions between the fluid and solid phases. It uses the convective HTC, h_v , defined in the main correlations and tables presented in section 2, and adds a solar radiation source term to the solid energy equation.

$$\nabla \cdot (\langle \rho \cdot c_{pf} \cdot T_f \rangle^f \cdot \mathbf{v}) = \nabla \cdot (k_{eff,f} \cdot \nabla \langle T_f \rangle^f) + h_v \cdot (\langle T_s \rangle^s - \langle T_f \rangle^f) \quad (10)$$

$$0 = \nabla \cdot (k_{eff,s} \cdot \nabla \langle T_s \rangle^s) + h_v \cdot (\langle T_f \rangle^f - \langle T_s \rangle^s) - S_{rad} \quad (11)$$

here c_{pf} is the specific heat of the fluid, T_f is the temperature of the fluid, $k_{eff,f}$ is the effective conductivity of the fluid, T_s is the temperature of the solid, $k_{eff,s}$ is the effective conductivity of the solid, h_v is the convective HTC and S_{rad} is the radiative source term.

The solar radiation source term, S_{rad} , is dependent on the emissivity, ε , and the extinction, β , within the OVR, as well as the incident solar radiation, I . It follows Beer's Law [50,51]:

$$S_{rad} = \varepsilon \cdot \beta \cdot I_{z=0} \cdot e^{-\beta \cdot z} \quad (12)$$

where z refers to fluid direction and $I_{z=0}$ is the initial incident solar radiation at the frontal surface of the volumetric absorber.

The thermal radiation within the porous region in steady state conditions is defined as:

$$q_{rad} = -k_{rad} \cdot \nabla \langle T_s \rangle^s \quad (13)$$

where k_{rad} is the radiative conductivity.

3.2. Effective properties

The HEM with the LTNE state requires the effective properties of the porous media and the fluid to analyse their thermal and hydrodynamic behaviour [52]. Moreover, these properties directly influence the fluid flow and the heat transfer mechanisms in the OVRs.

The behaviour of the fluid flow was determined by the viscous and inertial permeability coefficients (K_1 and K_2) and the mesh arrangement. As discussed in section 2, for the 0.7, 0.4 and 0.1 wire diameters, the ICs were defined by the Forthheimer-extended Darcy's law (Eq. (2)) with the coefficients shown in Table 2 for the IL pattern and Table 6 for the ST pattern. And, the GCs were defined by a modified Ergun equation [30,32], with the constants shown in Eq. (5) for the IL pattern and Eq. (7) for the ST pattern.

The heat transfer mechanisms included in the HEM with LTNE state were: the convective HTC (h_v), the radiative properties (β, k_{rad}), and the effective fluid and porous material conductivities ($k_{eff,f}, k_{eff,s}$).

The convective HTC, h_v , is also dependent on the morphology and arrangement considered but not the material. Thus, for the IL pattern, the h_v could be inferred with the ICs and the GC defined with Eq. (1) and Eq. (3) and the dimensionless constants presented in Table 1 and Eq. (4) respectively. Analogously, the same occurs with the ST pattern, the dimensionless constants showed in Table 5 and Eq. (6) are the ones used with the ICs (Eq. (1)) and the GC (Eq. (3)).

The radiative properties are based on both, the extinction coefficient and the radiative conductivity dependent on the geometry. For dense wire meshes, the extinction coefficient, β , is determined by Ref. [23]:

$$\beta = \frac{8}{\pi} \cdot \frac{1 - \varnothing}{d} \quad (14)$$

And, the radiative conductivity is simplified with the Rosseland approximation, k_{rad} , which is defined by Ref. [53]:

$$k_{rad} = \frac{16 \cdot \sigma \cdot T^3}{3 \cdot \beta} \quad (15)$$

where σ is the Stefan Boltzmann constant.

The effective thermal conductivity of the fluid and the porous absorber are function of the porosity and the conductivity of the fluid and the solid material [54]. For the fluid, it is as follows:

$$k_{eff,f} = \phi \cdot k_f \quad (16)$$

where k_f is the conductivity of the fluid and is defined by:

$$k_f = -2.28 \cdot 10^{-3} + 1.155 \cdot 10^{-4} \cdot T - 7.902 \cdot 10^{-8} \cdot T^2 + 4.117 \cdot 10^{-11} \cdot T^3 - 7.439 \cdot 10^{-15} \cdot T^4 \quad (17)$$

On the other hand, the effective thermal conductivity of the porous absorber expressed by the next equation:

$$k_{eff,s} = \frac{1 - \phi}{3} \cdot k_s \quad (18)$$

where k_s is the conductivity of the material. In this case, the wire meshes analysed are available commercially with 310 stainless steel as the material which conductivity is defined by Ref. [28]:

$$k_s = 4.0106 + 0.02750782 \cdot T - 6.173461 \cdot 10^{-6} \cdot T^2 \quad (19)$$

The value of the fluid specific heat, c_{pf} , and the value of the fluid viscosity, μ , are dependent on the temperature, and are defined by Ref. [28]:

$$c_{pf} = 1047.637 - 0.3726 \cdot T + 0.000945 \cdot T^2 - 6.024 \cdot 10^{-7} \cdot T^3 + 1.286 \cdot 10^{-10} \cdot T^4 \quad (20)$$

$$\mu = -8.383 \cdot 10^{-7} + 8.357 \cdot 10^{-8} \cdot T - 7.694 \cdot 10^{-11} \cdot T^2 + 4.644 \cdot 10^{-14} \cdot T^3 - 1.066 \cdot 10^{-17} \cdot T^4 \quad (21)$$

Table 9 shows the main geometrical properties and parameters that were used in the 2D HEM for each arrangement and wire diameter. The compactness factor assumed influenced the geometrical properties of the staggered arrangement.

3.3. Assumptions and boundary conditions

The assumptions implemented in the HEM with LTNE methodology to estimate the behaviour of a real volumetric absorber under high irradiation circumstances, as would occur in CST technology, are as follows:

- The fluid was air, which was modelled as an ideal gas and under steady state and laminar flow conditions. The laminar solver was implemented because the maximum Reynolds numbers in the simulations were 54 for the IL pattern and 19.7 for the ST pattern, and in previous studies, the laminar solver was demonstrated to be valid [23,28].
- The thermo-physical properties of the air and the porous material were dependent on the temperature as shown in section 3.2.
- The geometrical properties were calculated only by means of the geometrical parameters, d and w (Fig. 2), for each wire diameter and arrangement.
- The volumetric absorber had isotropic radiative properties.

Fig. 3 presents the boundary conditions used in this OVR model. The volumetric absorber dimensions are 60 mm × 60 mm, and the boundary conditions implemented in the analyses were:

- The frontal surface had various closure conditions: a) For the fluid, 1 m/s was the velocity at the inlet, which is a typical air inlet velocity in CST [55], the air temperature was 300 K. b) For the porous absorber, a heat flux was calculated for the frontal losses that an OVR has when it is struck by the solar irradiation. These frontal losses were defined by Ref. [56]:

Table 9

Main geometrical properties and parameters implemented in each arrangement and wire diameter for the 2D HEM.

General			Inline				Stagger			
d	M mm ⁻¹	ϕ SM	ϕ IL	d _{h-IL}	a _{v-IL}	β _{IL} m ⁻¹	ϕ ST	d _{h-ST}	a _{v-ST}	β _{ST} m ⁻¹
mm		%	%	mm	m ⁻¹		%	mm	m ⁻¹	
0.7	0.35	80	80	2.80	1,142	727	64.9	1.30	2,003	1275
0.4	0.61	80	80	1.60	2,004	1276	64.8	0.74	3,515	2238
0.1	2.44	80	80	0.40	7,984	5083	65.0	0.19	14,006	8917

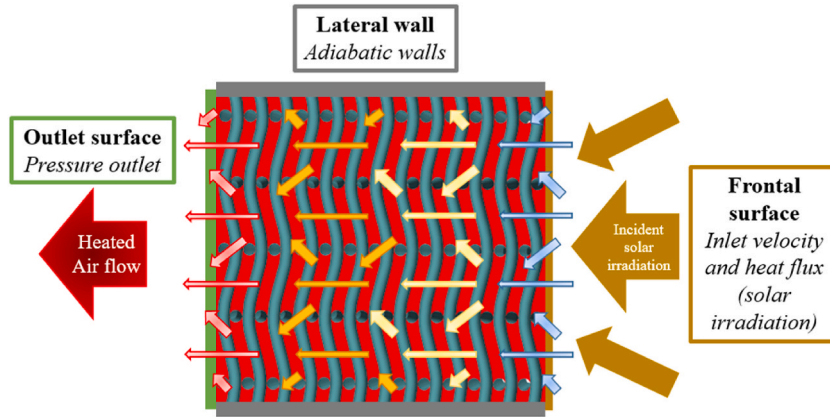


Fig. 3. Boundary conditions of the HEM with LTNE methodology for volumetric absorbers in CST technology.

$$TL_{inlet} = -\varepsilon \cdot \sigma \cdot (T^4 - T_{amb}^4) \quad (22)$$

In addition, the collimated power density that impacts the porous absorber was 600 kW/m^2 . Beer's Law governed the penetration of the radiation through the volumetric absorber as described by Eq. (12).

- The lateral walls of the volumetric absorber were considered adiabatic.
- A pressure outlet condition at the outlet surface of the volumetric absorber was assumed to be 0 Pa resulting in an atmospheric pressure at the outlet.

4. Results and discussion

The results part of this study analyses the different thermo-fluid-dynamic attributes obtained for the IL and ST arrangements in an OVR using a homogeneous equivalent method (HEM), also known as a porous medium method. Firstly, the numerical method presented in this section was validated, after that, the HEM was used to compare the values generated by the ICs and GCs to corroborate the results yielded by the latter. Secondly, the influence of the geometrical parameters, d and w , and the wire meshes with IL and ST arrangements on the performance of an OVR when employed in a 2D HEM with the LTNE state is analysed. Additionally, the results obtained in this study are compared with results in the literature for wire mesh absorbers with similar geometrical properties and parameters.

4.1. HEM results

This section validates the HEM with the LTNE state developed for this work with data from Avila-Marin study [28]. Then, the HEM-LTNE is used to analyse thoroughly: (i) the validity of the GCs with the ICs for two of the main thermo-fluid-dynamic attributes in OVRs (pressure drop and HTC), (ii) the impact of the wire mesh arrangements (IL and ST), and diameters (0.7, 0.4, 0.1 mm) in the temperature profiles, (iii) the effect of the wire mesh arrangements (IL and ST), and diameters (0.7, 0.4, 0.1 mm) on the thermal and hydrodynamic performance.

4.1.1. Validation of the 2D HEM-LTNE methodology

Before analysing the influence of the wire mesh arrangements and diameters on the thermal and fluid flow performance, a

Table 10

Validation of the methodology implemented in the 2D HEM-LTNE with Rosseland approximation (R) with respect to the P1 approximation from Avila-Marin's work [28].

Mesh type	P1 approximation literature (P1- [28])		Rosseland approximation (R)		Deviation between P1- [28] and R	
	Tf _{out} K	Ts _{in} K	Tf _{out} K	Ts _{in} K	Tf _{out} %	Ts _{in} %
A	716.5	1,008.0	716.7	1,015.6	0.0	-0.8
B	735.1	914.3	729.0	916.2	0.8	-0.2
C	740.2	811.1	734.3	812.9	0.8	-0.2
D	710.7	715.8	692.7	699.7	2.5	2.3
E	723.2	751.3	735.4	771.7	-1.7	-2.7
F	680.9	684.2	665.3	669.5	2.3	2.1

validation of the methodology applied in the 2D HEM with the LTNE state with the Rosseland approximation (R) as a radiative model was carried out.

The methodology used in this research was validated through a comparison with data available in Avila-Marin's work [28], which uses the P1 approximation as radiative models. The geometrical parameters and properties of the meshes used in Avila-Marin's work [28] were implemented in the methodology applied in the current study.

Table 10 presents the fluid temperature at the outlet of the OVR and the solid matrix temperature at the inlet of the OVR with the methodology implemented in the current study, following the Rosseland (R) approximation. These results are compared to the results from Avila-Marin's work that used the P1 approximation. The deviations between the R and the P1 radiative models are lower than 3 %. Hence, the methodology implemented in the 2D HEM-LTNE developed in the current study is considered valid.

4.1.2. Evaluation of the inline and staggered arrangements with the 2D HEM

The methodology implemented in the 2D HEM-LTNE applied in the current research is used to firstly, validate the GCs with the ICs in a porous model; secondly, to measure the impact of the wire mesh diameters and arrangements on the profile temperatures; and, finally, to analyse the effect of the wire diameters (0.7, 0.4, 0.1 mm) and mesh arrangements on the thermal and hydrodynamic performance of dense wire mesh OVRs. In all the cases, the porosity of the single mesh is kept constant at 80 %. Also, the physics, the effective properties, and the assumptions and boundaries presented in section 2.2.3 apply in all the cases. For the sake of clarity, a velocity inlet of 1 m/s and a collimated power density at the inlet of the OVR of 600 kW/m^2 , typical OVR operational conditions, were used in all the cases as well [7,21].

4.1.2.1. Corroboration of the GCs with the ICs in the HEM-LTNE. ICs and GCs are widely used to estimate the pressure drop and the convective HTC in porous structures [15,16,57–62]. Usually, ICs present better agreement with numerical detailed simulations and/or experimental results, while GCs offer greater versatility to perform parametric/sensitive analysis in optimisations because geometrical properties are considered. This section analyses the influence of both types of correlations, ICs and GCs, regarding the pressure drop, and the solid and the fluid temperatures for dense wire meshes with an 80 % single-mesh porosity.

Fig. 4 shows the effects of introducing the ICs and GCs in the HEM-LTNE on the pressure drop results for the IL and ST arrangement and the three wire diameters (0.7, 0.4, 0.1 mm).

The pressure drop of the IL pattern results in the left scatter plot of Fig. 4. All the deviations presented use the IC as the reference, for the 0.1 mm wire diameter, the pressure drop obtained with the IC is 2,062 Pa whereas with the GC is 2,094 Pa, a deviation of 1.6 %. For the 0.4 mm wire diameter, the pressure drop results are approximately 147 Pa for both the IC and the GC, which represents a 0.2 % deviation. Finally, for the 0.7 mm wire diameter, the pressure drop estimated with the IC is 24 Pa whereas with the GC is 51 Pa, showcasing a deviation of 110 %. Even though the deviation percentage is large for the 0.7 mm wire diameter, the real value difference is not that significant. Besides, in this case, the GC at low velocities is more accurate than the IC as presented in section 2.3.2.

The pressure drop of the ST pattern results are depicted in the right scatter plot of Fig. 4. All the deviations presented use the IC as the reference, thus the pressure drop obtained for the 0.1 mm wire diameter is 12,236 and 12,029 Pa for the IC and GC respectively, a 1.7 % deviation. For the 0.4 mm wire diameter, the IC and the GC present a deviation of 10.4 %, with pressure drop values of 940 and 1,038 Pa respectively. Finally, for the 0.7 mm wire diameter, the pressure drop results are 367 and 387 Pa for the IC and GC respectively, a 5.4 % deviation.

In general, the ICs and the GCs present pressure drop values in a similar range, which means that the use of GCs for obtaining the pressure drop in a HEM-LTNE simulation are as valid as the use of ICs, and, in some cases, even better. There are two key differences between the GCs and ICs used in this study. First, the GCs are correlations developed specifically for wire meshes, and they are based on the geometrical parameters, d and w . Second, the ICs are obtained from literature-based correlations for pressure drop in general, they are not specific of wire meshes, in spite of which the ICs have been applied to each wire diameter and arrangement.

Fig. 5 presents the solid temperature at the inlet of the OVR, $T_{s,in}$, and the fluid temperature at the outlet of the OVR, $T_{f,out}$, for different wire diameters (0.7, 0.4, 0.1 mm) and arrangements (IL, ST). The results obtained with the implementation of both

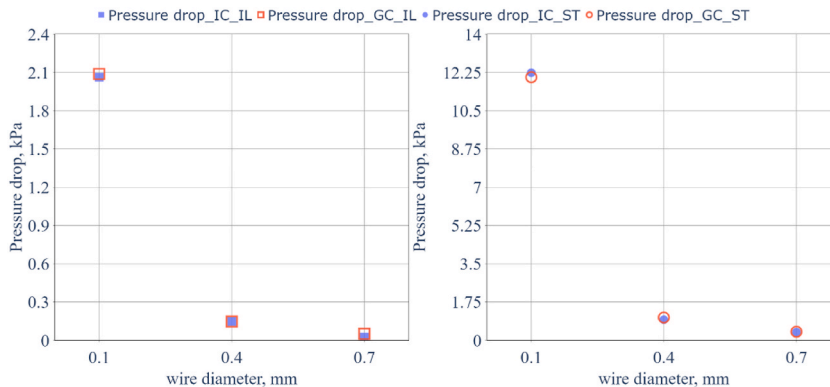


Fig. 4. Effect of introducing the ICs and GCs in the HEM-LTNE on the pressure drop results for the distinct arrangements and wire diameters.

correlations, ICs and GCs, in the HEM-LTNE show very good agreement in all the T_{S-in} and T_{f-out} of the OVRs with deviations of less than 1.4 %. The largest deviations between the ICs and the GCs in the T_{S-in} and the T_{f-out} of the OVRs are obtained for the 0.7 mm wire diameter for both the IL and the ST arrangements. For a wire diameter of 0.7 mm with a ST pattern, the T_{S-in} of the OVR calculated using the GC is 13.2 K higher than that obtained using the IC, while the T_{f-out} with the GC is 2 K lower compared to the IC. Similarly, for the IL pattern with a wire diameter of 0.7 mm, the T_{S-in} obtained using the GC is 14.1 K higher than that of the IC, whereas the T_{f-out} shows a temperature difference of 5.2 K between the GC and the IC.

As demonstrated previously with the thermo-fluid-dynamic results, the GC for both the IL and the ST patterns are in good agreement with the ICs across all wire diameters and patterns. Consequently, the GC for each arrangement will be used in the following analyses.

4.1.2.2. Impact of the wire diameters and arrangements on the temperatures profiles. This section discusses how the wire diameters and mesh arrangements analysed affect the T_{S-in} and T_{f-out} profiles emphasising the most important differences. Fig. 6a, based on a wire diameter of 0.7 mm, shows a difference of 321 K in the OVR's T_{S-in} between the IL, 1280 K, and the ST arrangement, 959 K. The reason of that difference in the T_{S-in} of the OVR is because, despite the single mesh having the same porosity (80 %), the analytical volumetric porosity for the IL pattern is 80 % whereas for the ST pattern is nearly 65 %, increasing the specific surface area (a_v) in the ST pattern compared to the IL pattern as presented in Table 9. An increment in the a_v produces a rise in the convective HTC, and for that reason a reduction in the T_{S-in} of the OVR [33]. Moreover, with this wire diameter (0.7 mm), the T_{f-out} of the OVR is 724 K for the ST pattern and 650 K for the IL pattern, indicating the ST configuration has better overall performance, having a lower T_{S-in} and a higher T_{f-out} , which means a better balance of the a_v and extinction coefficient (β) for the ST pattern. Finally, the thermal equilibrium is considered to be reached when the difference value between the T_{S-in} and T_{f-out} is 0.025 K. Taking that into account, the IL pattern needs nearly 4.7 cm to achieve the thermal equilibrium while the ST pattern needs less than half the distance (1.9 cm). This effect is directly related with the a_v and the β , which both are larger in the ST case than in the IL (Table 9), producing quicker extinction and convective heat transfer in the ST configuration.

Fig. 6b, based on a wire diameter of 0.4 mm, shows a difference of 286 K in the T_{S-in} between the IL, 1139 K, and the ST pattern, 853 K, which is less than in case (a) with a wire diameter of 0.7 mm. This difference is explained by a higher a_v because using a wire diameter of 0.4 mm enhances the convective HTC compared to case (a), consequently the distance required to extinguish the solar irradiation is reduced. Case (b) reduces the T_{S-in} consequently it increases the T_{f-out} of the OVR for IL and ST arrangement. Moreover, the OVR's T_{f-out} with the ST arrangement (730 K) and with the IL arrangement (691 K) has a difference of 39 K reduction compared to the 74 K difference obtained in case (a). Finally, case (b) requires 1.9 cm with the IL configuration to achieve the thermal equilibrium, while the ST arrangement needs 0.9 cm.

Note that the IL arrangement with the 0.4 mm wire diameter needs the same distance to reach the thermal equilibrium as the ST arrangement with the 0.7 mm wire diameter. Although both have the same a_v and β (Table 9), for 0.7 mm wire diameter and ST pattern, the T_{S-in} and T_{f-out} of the OVR are 959 K and 724 K respectively, and for 0.4 mm wire diameter and IL pattern are 1139 K and 691 K respectively. In the following section the reason of these results is extended in detail.

Fig. 6c, based on a wire diameter of 0.4 mm, presents a difference of 132 K between T_{S-in} with the IL pattern (790 K) compared to the ST pattern (658 K), being higher for the IL pattern as usual. However, the T_{f-out} behaves upside down, having a lower value for the ST pattern (647 K) than for the IL pattern (711 K), with a difference of -64 K. The behaviour of the T_{S-in} is explained, again, by a higher

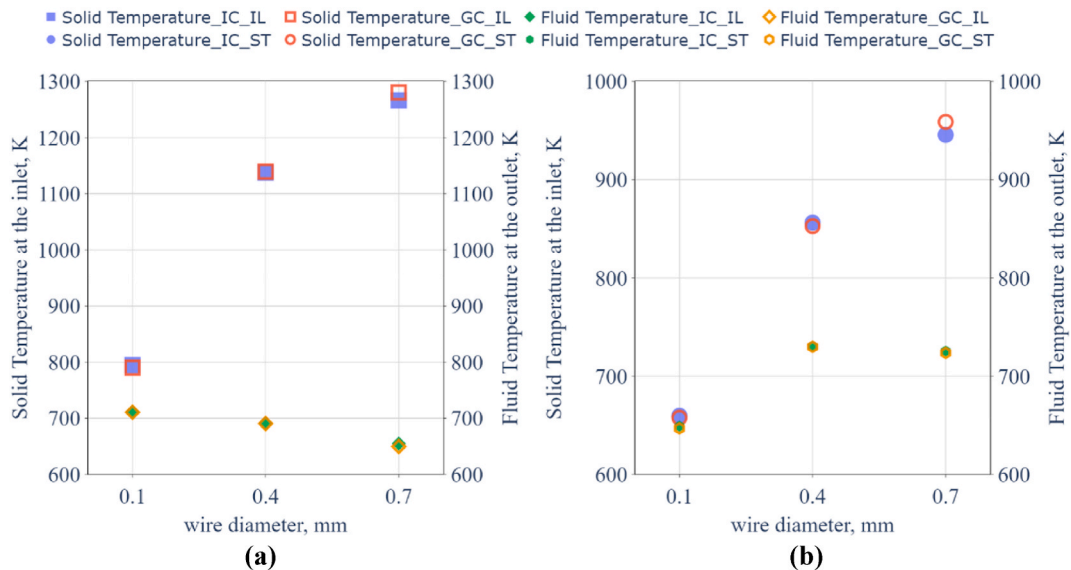


Fig. 5. Comparison between the IL and ST patterns with wire diameters 0.7, 0.4 and 0.1 mm.

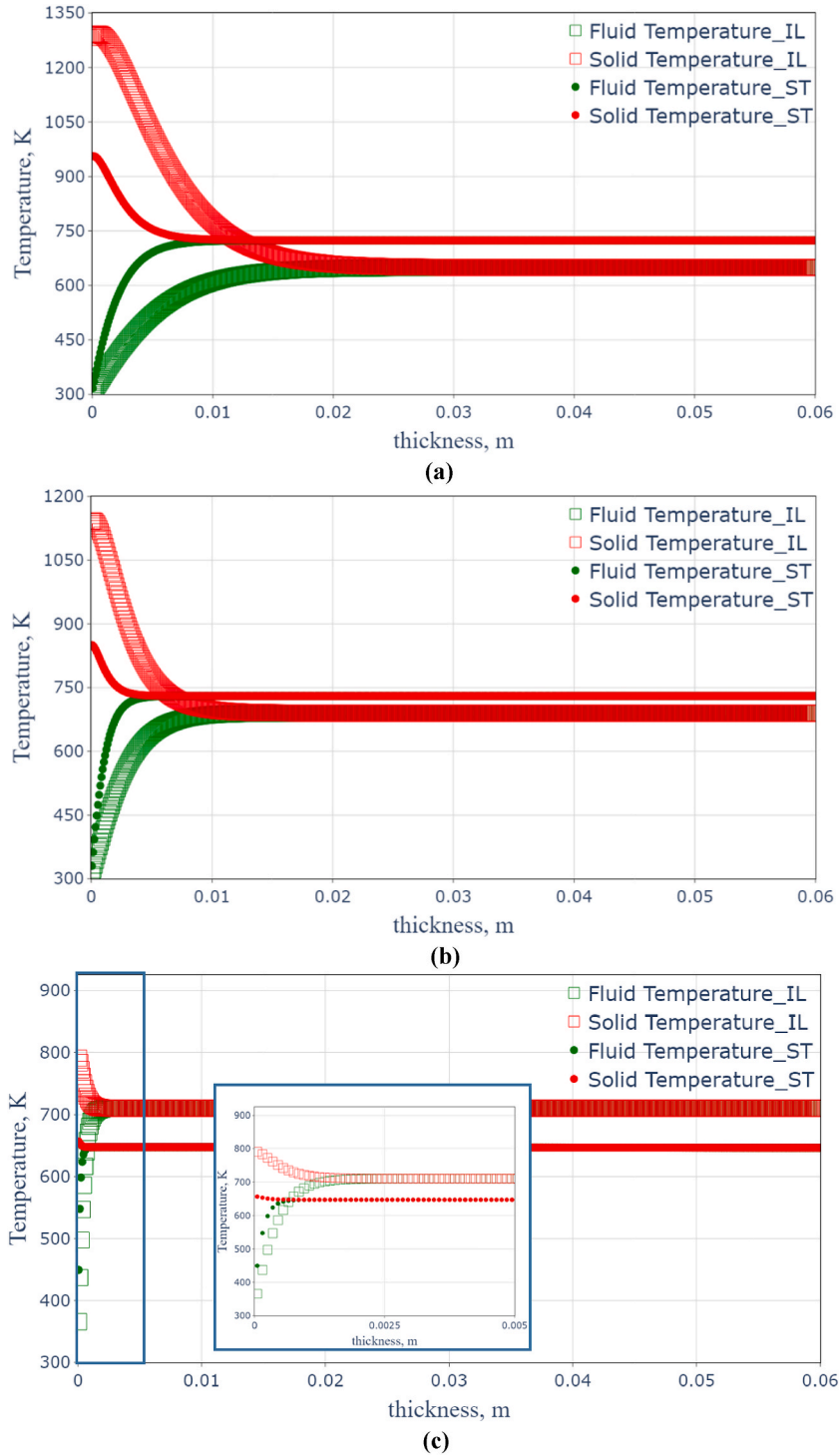


Fig. 6. Temperature profile of the fluid and the solid matrix over the length of the porous medium for a collimated flux density of 600 kW/m^2 : (a) 0.7 mm wire diameter, (b) 0.4 mm wire diameter and (c) 0.1 mm wire diameter.

convective HTC compared to case (a) and (b), while in terms of T_{f-out} , the balance of the β and convective HTC is better in the IL pattern showing a change in the tendency. The distance to reach the thermal equilibrium is 3.5 mm and 1.4 mm for the IL and the ST pattern respectively.

Overall, Fig. 6 shows that the ST pattern reduces the $\Delta T = T_{S-in} - T_{f-out}$ when reducing the wire diameter from 235 K at 0.7 mm, to

123 K at 0.4 mm, and 11 K at 0.7 mm. The reduction in temperature difference is produced by a very fast increase of the α_v and β , up to very high values, as can be seen in Table 9.

Similarly, the IL pattern follows the same tendency but at a faster rate with more moderate values of α_v and β (Table 9). The $\Delta T = T_{S-in} - T_{f-out}$ changes from 630 K for a wire diameter of 0.7 mm, to 448 K for a wire diameter of 0.4 mm, and 79 K for a wire diameter of 0.1 mm. Furthermore, the distance needed to achieve thermal equilibrium distance is reduced in parallel with the reduction of the wire diameter, but with greater distances compared to the similar cases with the ST pattern.

Finally, for case (c), for the first time, the IL configuration is able to achieve a higher T_{f-out} compared to the ST configuration, even though the T_{S-in} is higher. That means that the balance between convective and radiative mechanisms is better in the IL configuration. This analysis is extended in the next section, where other thermal and hydrodynamic operating variables are presented.

4.1.2.3. Effect of wire diameters and the arrangements on thermal and hydrodynamic performance. This section presents how the wire diameters and the arrangements in the dense wire mesh lattices influence their thermal and hydrodynamic performance. To do so, the radiative thermal losses, the convective HTC, the pressure drop and the thermal efficiency are analysed for each wire diameter and arrangement.

Fig. 7 shows the variation of the T_{S-in} , the thermal losses and the convective HTC at the OVR inlet for IL and ST patterns with the wire diameters studied (0.7, 0.4, 0.1 mm). The T_{S-in} and the radiative losses at the inlet are depicted for both patterns, top-left for the IL pattern and top-right for the ST pattern. In both scatter plots, the increase in T_{S-in} with the wire diameter causes an increase in the radiative losses. The IL pattern has higher thermal losses at the inlet than the ST pattern due to the higher T_{S-in} . Having a higher or lower T_{S-in} depends on the convective HTC, which in turn is a function of the geometrical parameters (d and w) of the OVR. Thus, the T_{S-in} is shown together with the convective HTC at the inlet for IL pattern (bottom-left) and ST pattern (bottom-right), and each wire diameter. For both patterns, the scatter plots show a reduction in the convective HTC at the inlet when the wire diameter increases. In addition, the convective HTC at the inlet for the ST pattern is higher compared to the IL arrangement, similar to the trend observed in the α_v . Consequently, due to the lower convective HTC at the inlet, the IL pattern experiences both a higher T_{S-in} and greater radiative losses compared to the ST pattern.

Examining the effects of the wire diameter more closely, an increase in the wire diameter leads to an augmentation of the radiative losses at the inlet and a reduction in the convective HTC there regardless of the pattern, such that the T_{S-in} is the highest for the largest wire diameter.

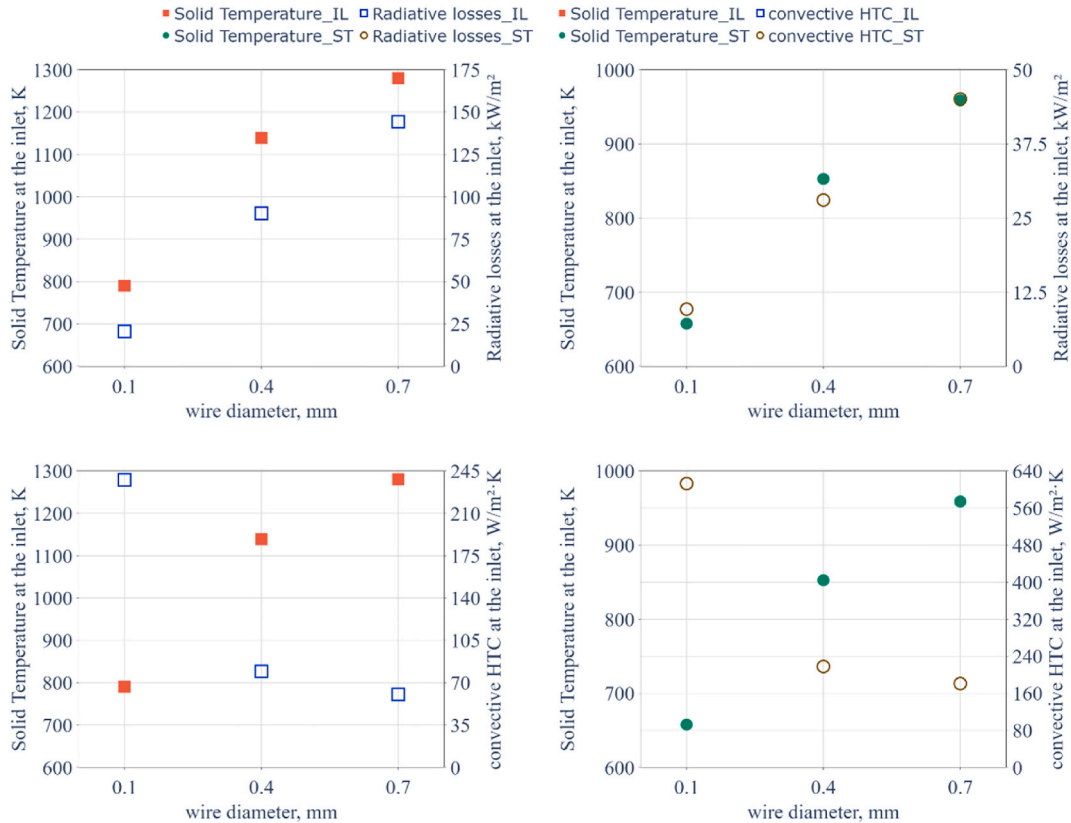


Fig. 7. Comparison between the T_{S-in} with the radiation losses and the convective HTC at the inlet for each wire diameter studied with the IL pattern (left) and the ST pattern (right).

Fig. 8 shows the impact of the wire diameters and patterns on the T_{f-out} and the pressure drop of the fluid over the length of the absorber. For IL and ST patterns, the pressure drop decreases as the wire diameter increases. However, the pressure drop for the ST pattern is higher compared to the IL pattern. The smaller the wire diameter, the greater the difference in the pressure drop between the IL and the ST pattern as shown in Table 11. The difference in pressure drop between the two patterns for the 0.1 mm wire is 6 times higher for the ST pattern, but 7.6 times higher for the 0.7 mm wire.

The trend of the T_{f-out} varies with the patterns. While the IL pattern, Fig. 8a, has a T_{f-out} that decreases as the wire diameter increases, the ST arrangement, Fig. 8b, has its maximum T_{f-out} for a wire diameter of 0.4 mm.

Finally, Fig. 9 presents the variation of the T_{f-out} and the thermal efficiency of the OVR for IL and ST patterns, and the three wire diameters. The thermal efficiency results closely mirror those for T_{f-out} with the ST pattern having higher thermal efficiency than the IL pattern, except for the smallest wire diameter (0.1 mm).

The ST pattern with 0.4 mm wire diameter shows the highest thermal efficiency taking a value of 88, with a T_{f-out} of 730 K and a T_{S-in} of 853 K. The 0.7 mm, shows a small reduction in thermal efficiency (86.1 %), and 0.1 mm wire has the lowest thermal efficiency of the ST OVR at 78 %.

By contrast, the IL configuration has its highest thermal efficiency of 84.8 % with the 0.1 mm wire diameter, with a T_{f-out} of 711 K and a T_{S-in} of 790 K. The thermal efficiency of the 0.4 and 0.7 mm wire diameters is lower at 79 and 70.4 %, respectively.

Overall, with the exception of the 0.1 mm wire diameter, the ST pattern has better thermal results despite its higher hydrodynamic resistance. However, in some applications it might be an advantage to consider an IL configuration because its thermal performance is close to or even better than that of the ST configuration, and its hydrodynamic performance is better in all cases compared to the ST configuration.

Lastly, given that α_v and β (Table 9) are the same for both the IL arrangement with 0.4 mm wire diameter and the ST arrangement with 0.7 mm wire diameter, but their hydrodynamic and thermal performance differ substantially, a more detailed comparison is warranted. The pressure drop of the IL-0.4 mm is 147 Pa whereas the ST-0.7 mm is 387 Pa, an increase of 163 %. In terms of convective HTC at the OVR inlet, the IL-0.4 mm results in 79 W/m²·K whereas the ST-0.7 mm results in 181 W/m²·K, an increase of 128 %. In addition, comparing in detail the operating variables, the T_{S-in} in the OVRs are 1139 K and 959 K for the IL-0.4 mm and the ST-0.7 mm respectively. And, the T_{f-out} in the OVRs are 690 K and 724 K for the IL-0.4 mm and the ST-0.7 mm respectively. Finally, regarding the thermal efficiency, the IL-0.4 mm has 79 % whereas the ST-0.7 mm has 86.1 %, a difference of 7.1 %. Concluding that the α_v and β are parameters that could be used to compare OVR performance with the same arrangement, but not with different arrangements.

Conversely, having different α_v and β (Table 9), the IL-0.1 mm and the ST-0.4 mm have the best thermal efficiencies for the IL and the ST arrangements. The IL-0.1 mm achieves similar convective HTC values than the ST-0.4 mm, 238 W/m²·K and 218 W/m²·K respectively, resulting in similar thermal efficiencies, 84.8 % for the IL-0.1 mm and 88 % for the ST-0.4 mm. Consequently, the difference between the T_{S-in} and T_{f-out} is minimum; for the IL-0.1 mm, the T_{S-in} and T_{f-out} are 790 K and 711 K respectively, and for the ST-0.4 mm, 853 K and 730 K respectively. All in all, the importance of emphasising the effect of the arrangement, the porosity and the wire diameter on the operating variables becomes clear.

4.2. Comparison with literature

The consistency of this work is demonstrated by a last comparison with results from the literature. This comparison consists on using the results obtained in Avila-Marin's work [28] and observe similarities with the results obtained in this study. In both studies, the incident radiation is 600 kW/m², however, Avila-Marin implemented a P1 radiative approximation while in this study the Roseland radiative approximation is applied. In Avila-Marin's work, the wire mesh arrangement studied was the ST, for that reason, the comparison only shows ST wire mesh absorbers. Two of the six mesh absorbers studied by Avila-Marin, Mesh B and C, have similar geometrical parameters and properties to two of the mesh absorbers studied in this work, ST-0.7 mm (S1) and ST-0.4 mm (S2) as shown in Table 12.

Table 13 shows the comparison between the two wire mesh absorbers of Avila-Marin's work [28] and the two wire mesh absorbers of this study with similar geometrical parameters and properties. As aforementioned, Mesh B and S1 are two wire mesh absorbers with similar geometrical parameters and properties and because of that, they are compared. The same occurs with Mesh C and S2.

Both pair of wire meshes, comparison 1 (Mesh B – S1) and comparison 2 (Mesh C – S2), have similar T_{S-in} and T_{f-out} trends. In both comparisons, the T_{S-in} is higher than the T_{f-out} which validates the results obtained in this study. In addition, the deviations of the comparison 1 are lower than 5 % for the T_{S-in} and 1.5 % for the T_{f-out} , resulting in 45 K higher and 11 K lower for the S1 than for the Mesh B. The deviations of the comparison 2 are 5.2 % for the T_{S-in} and 1.4 % for the T_{f-out} , resulting in 42 K higher and 10 K lower for the S2 than for the Mesh C. These results are significant because even if the wire meshes have different porosity and geometrical parameters and properties and the numerical simulations are using different radiative approximations, the resulting outcomes remain consistent.

5. Conclusions

This study highlights the significance of comparing two different wire mesh arrangements, inline (IL) and staggered (ST), in terms of thermo-fluid-dynamic attributes. Both arrangements have a fixed single-mesh porosity of 80 % and are evaluated for three wire diameters: 0.7, 0.4 and 0.1 mm.

The comparison began with an examination of the thermo-fluid-dynamic attributes, including the pressure drop and the convective heat transfer coefficient (HTC), based on data from the literature using detailed 3D simulations. This examination validated the

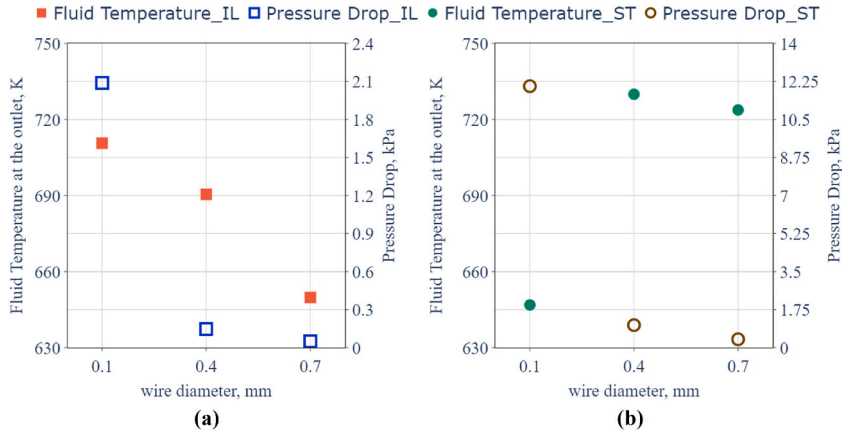


Fig. 8. Comparison between the T_{f-out} and the ΔP of the fluid along both patterns, IL pattern (a) and ST pattern (b).

Table 11

Pressure drop for IL and ST arrangement and 0.1 and 0.7 mm wire diameters.

Wire diameter	Pressure drop for 0.1 mm kPa	
Arrangement	Pa	
Inline (IL)	2	51
Stagger (ST)	12	387

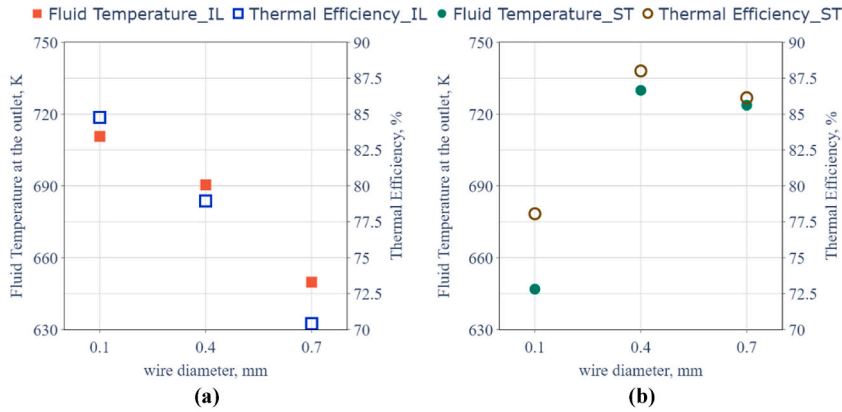


Fig. 9. Comparison between the T_{f-out} and the thermal efficiency for both patterns, IL (a) and ST (b).

Table 12

Geometrical parameters and properties of two wire mesh absorbers of the Avila-Marin's work [28] and two wire mesh absorbers of this study.

Mesh	Mesh B	Mesh C	S1 (ST-0.7 mm)	S2 (ST-0.4 mm)
$\phi_{ST}, \%$	68	62	65	65
d, mm	0.7	0.5	0.7	0.4
a_v, m^{-1}	1849	3044	2003	3515
β, m^{-1}	1177	1938	1275	2238

Individual Correlations (ICs) and Generalised Correlations (GCs) with the CFD results for both OVR arrangements (IL and ST) and the three wire diameters. In the results section, the comparison followed three phases:

1. Validation of the GCs with the ICs using a 2D Homogeneous Equivalent Model (HEM) with local thermal non-equilibrium (LTNE) state.
2. Analysis of the wire diameters and patterns on the thermal and fluid flow performance of OVRS.

Table 13

Comparison between Avila-Marin's wire mesh absorbers [28] and the wire mesh absorbers of this study.

Mesh name	Comparison 1		Comparison 2		Deviations, %	
	Mesh B	S1 (ST-0.7 mm)	Mesh C	S2 (ST-0.4 mm)	Comparison 1	Comparison 2
T_{S-in} , K	914	959	811	853	−4.9	−5.2
T_{f-out} , K	735	724	740	730	1.5	1.4

3. Corroboration of results with existing literature to confirm the model's reliability.

The HEM with LTNE state demonstrated that GCs results are consistent with those obtained from ICs, thereby affirming that GCs can be employed in OVR optimisation studies with the same confidence as ICs, regardless of wire diameters or arrangement.

The research revealed that the wire diameter and arrangement significantly influence the thermal and hydrodynamic performance. A larger a_v enhances the convective HTC but also increases the pressure drop, which may impact negatively the absorber performance, as observed with ST-0.1 mm wire diameter. Higher convective HTC causes lower radiative losses and lower T_{S-in} . For the IL pattern, the T_{f-out} decreases with the wire diameter, whereas for the ST pattern, the T_{f-out} reaches a maximum at a wire diameter of 0.4 mm.

The thermal efficiency which is directly related to the T_{f-out} reached an 88 % for the ST pattern with the 0.4 mm wire diameter, and an 84.8 % for the IL pattern with the 0.1 mm wire diameter. These findings emphasise the trade-offs between the thermal and hydrodynamic performance. With regard to thermal efficiency, the ST pattern generally outperforms the IL pattern for the given 80 % porosity, whereas the IL pattern exhibits superior hydrodynamic performance. Achieving an optimal dense wire mesh OVR requires balancing the convective HTC and the pressure drop.

Finally, the consistency of the results with existing literature validates the numerical model and its outcomes.

For future work, the optimal design with 80 % porosity will be determined together with the optimisations of 75 % and 85 % for the wire diameters ranging from 0.1 to 0.7 mm.

CRediT authorship contribution statement

Daniel Sanchez-Señoran: Writing – review & editing, Writing – original draft, Visualization, Validation, Software, Resources, Methodology, Investigation, Data curation. **Miguel A. Reyes-Belmonte:** Writing – review & editing, Resources, Investigation, Formal analysis. **Michael P. Kinzel:** Writing – review & editing, Resources, Investigation, Formal analysis. **Marina Casanova:** Writing – review & editing, Resources, Investigation, Formal analysis. **Antonio L. Avila-Marin:** Writing – review & editing, Writing – original draft, Visualization, Supervision, Software, Resources, Project administration, Methodology, Investigation, Funding acquisition, Formal analysis.

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.

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Appendix 1

With respect to I1 and S1 cases, the GCs predict the convective HTC at thermal equilibrium better with 5.6 % and 7.7 % average deviations, respectively, compared to the ICs with 9.3 % and 12.9 % average deviations, respectively. However, the ICs for cases I3 and S3 predict the convective HTC at thermal equilibrium better with 0.4 % and 1.3 % average deviations, respectively, compared to the GCs for case I3 with a 3.6 % average deviation and for case S3 with a 4.7 % average deviation.

Table A.1.1

Convective HTC at thermal equilibrium obtained with the CFD simulation, the IC and the GC for I1 case (IL-0.7 mm).

v m/s	CFD	IC	GC
	h_{lv} , kW/m ³ .k	h_{lv} , kW/m ³ .k	h_{lv} , kW/m ³ .k

(continued on next page)

Table A.1.1 (continued)

v m/s	CFD			IC			GC		
	h_{lv} , kW/m ³ ·k			h_{lv} , kW/m ³ ·k			h_{lv} , kW/m ³ ·k		
	700 K	1100 K	1500 K	700 K	1100 K	1500 K	700 K	1100 K	1500 K
	700 K	1100 K	1500 K	700 K	1100 K	1500 K	700 K	1100 K	1500 K
1	77	105	134	87	104	118	81	105	132
3	101	125	149	116	139	157	112	121	142
5	165	165	180	132	157	179	154	142	154
Average deviation, %				9.3			5.6		

Table A.1.2

Convective HTC at thermal equilibrium obtained with the CFD simulation, the IC and the GC for I3 case (IL-0.1 mm).

v m/s	CFD			IC			GC		
	h_{lv} , MW/m ³ ·k			h_{lv} , MW/m ³ ·k			h_{lv} , MW/m ³ ·k		
	700 K	1100 K	1500 K	700 K	1100 K	1500 K	700 K	1100 K	1500 K
1	3.6	5.0	6.5	3.6	5.0	6.5	3.7	5.1	6.6
3	3.6	5.0	6.4	3.6	5.0	6.4	3.8	5.2	6.6
5	3.6	5.0	6.3	3.6	4.9	6.3	3.9	5.2	6.6
Average deviation, %				0.4			3.6		

Table A.1.3

Convective HTC at thermal equilibrium obtained with the CFD simulation, the IC and the GC for S1 case (ST-0.7 mm).

v m/s	CFD			IC			GC		
	h_{lv} , kW/m ³ ·k			h_{lv} , kW/m ³ ·k			h_{lv} , kW/m ³ ·k		
	700 K	1100 K	1500 K	700 K	1100 K	1500 K	700 K	1100 K	1500 K
1	344	467	594	393	452	482	374	481	606
3	509	593	692	601	687	739	572	567	646
5	894	903	938	731	840	901	866	702	727
Average deviation, %				12.9			7.7		

Table A.1.4

Convective HTC at thermal equilibrium obtained with the CFD simulation, the IC and the GC for S3 case (ST-0.1 mm).

v m/s	CFD			IC			GC		
	h_{lv} , MW/m ³ ·k			h_{lv} , MW/m ³ ·k			h_{lv} , MW/m ³ ·k		
	700 K	1100 K	1500 K	700 K	1100 K	1500 K	700 K	1100 K	1500 K
1	16	23	30	17	23	30	16	22	29
3	16	22	28	16	22	28	16	22	29
5	15	21	27	15	21	27	17	22	28
Average deviation, %				1.3			4.7		

Appendix 2

Table A.2.1Numerical $\Delta P/L$ results obtained with the CFD simulation, the IC, and the GC for I1 and S1 cases (IL and ST pattern with 0.7 mm wire diameter).

v m/s	I1			S1		
	CFD	IC	GC	CFD	IC	GC
	$\Delta P/L$ Pa/m	$\Delta P/L$ Pa/m	$\Delta P/L$ Pa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m
1	279	176	289	1.9	1.8	1.9
3	1,547	1,578	1,675	12	12	11
5	4,521	4,381	4,140	30	30	28
Average deviation, %	–	20.5	6.2	–	1.0	2.5

Table A.2.2Numerical $\Delta P/L$ results obtained with the CFD simulation, the IC, and the GC for I3 and S3 cases (IL and ST pattern with 0.1 mm wire diameter).

v m/s	I3	S3
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Table A.2.2 (continued)

v m/s	I3			S3		
	CFD	IC	GC	CFD	IC	GC
	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m
	CFD	IC	GC	CFD	IC	GC
	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m	$\Delta P/L$ kPa/m
1	8.2	8.3	8.4	53.6	54.1	53.3
3	31	31	31	201	200	200
5	61	61	61	397	397	398
Average deviation, %	–	0.8	1.3	–	0.5	0.4

Nomenclature

M	Mesh count, m^{-1}
d	Wire diameter, mm
w	Wire spacing, m
c_f	Compactness factor
$\varnothing$	Volumetric porosity, %
$\varnothing_{SM}$	Single mesh porosity, %
$\varnothing_{IL}$	Inline mesh porosity, %
$\varnothing_{ST}$	Staggered mesh porosity, %
a_v	Specific surface, m^{-1}
a_{v-IL}	Specific Surface of the Inline arrangement, m^{-1}
a_{v-ST}	Specific Surface of the Stagger arrangement, m^{-1}
d_h	Hydraulic diameter, m
d_{h-IL}	Hydraulic diameter of the Inline arrangement, m
d_{h-ST}	Hydraulic diameter of the Stagger arrangement, m
F	Geometrical factor, -
ρ	Density, $kg\ m^{-3}$
v	Superficial velocity at the inlet, $m\ s^{-1}$
μ	Dynamic viscosity, $kg\ m^{-1}\ s^{-1}$
K_1	Viscosity coefficient, m^2
K_2	Inertial coefficient, m
Re	Reynolds number, -
Pr	Prandtl number, -
Nu_{lv}	Nusselt number, -
h_v	Volumetric convective heat transfer coefficient, $W\ m^{-3}\ K^{-1}$
P_f	Differential pressure, $Pa\ m^{-1}$
c_{pf}	Specific heat of the fluid, $J\ kg^{-1}\ K^{-1}$
T_f	Temperature of the fluid, K
T_s	Temperature of the solid, K
$k_{eff,f}$	Effective conductivity of the fluid, $W\ m^{-1}\ K^{-1}$
$k_{eff,s}$	Effective conductivity of the solid, $W\ m^{-1}\ K^{-1}$
S_{rad}	Radiative source term, $W\ m^{-3}$
ε	Emissivity, -
a	Absortance, -
β	Extinction, m^{-1}
I	Incident solar radiation, $W\ m^{-2}$
σ	Stefan Boltzmann constant, $W\ m^{-2}\ K^{-4}$
q_{rad}	Thermal radiation, $W\ m^{-3}$
k_{rad}	Radiative conductivity, $W\ m^{-1}\ K$
k_f	Conductivity of the fluid, $W\ m^{-1}\ K$
k_s	Conductivity of the solid, $W\ m^{-1}\ K$
TL_{inlet}	Frontal thermal losses, $W\ m^{-2}$
T	Temperature, K
T_{amb}	Room temperature, K
$\Delta P/L$	Pressure drop per unit length, $Pa\ m^{-1}$
CFD	Computational Fluid Dynamics
OVRs	Open volumetric receivers
HEM	Homogenous equivalent model
LTNE	Local thermal non equilibrium
HTC	Heat transfer coefficient
CST	Concentrated Solar Tower
SiC	Silicon carbide
ST	Stagger
IL	Inline
IC	Individual Correlation
GC	Generalised Correlation
$\langle \rangle$	Volumetric average

Data availability

Data will be made available on request.

References

- [1] IRENA, Renewable power generation costs in 2022. <https://www.irena.org/Publications/2023/Aug/Renewable-Power-Generation-Costs-in-2022>, 2022. (Accessed 27 September 2023).
- [2] R.P. Merchán, M.J. Santos, A. Medina, A. Calvo Hernández, High temperature central tower plants for concentrated solar power: 2021 overview, *Renew. Sustain. Energy Rev.* 155 (2022) 111828, <https://doi.org/10.1016/j.rser.2021.111828>.
- [3] C.K. Ho, Advances in central receivers for concentrating solar applications, *Sol. Energy* 152 (2017) 38–56, <https://doi.org/10.1016/j.solener.2017.03.048>.
- [4] A.L. Ávila-Marín, Volumetric receivers in solar thermal power plants with central receiver system technology: a review, *Sol. Energy* 85 (2011) 891–910, <https://doi.org/10.1016/j.solener.2011.02.002>.
- [5] R.D. Skocypiec, R.F. Boehm, J.M. Chavez, Heat transfer modeling of the IEA/SSPS volumetric receiver, *J. Sol. Energy Eng.* 111 (1989) 138–143, <https://doi.org/10.1115/1.3268299>.
- [6] T.E. Hellmuth, L.K. Matthews, Modeling and optimum design of a wire mesh solar volumetric air receiver, *J. Sol. Energy Eng.* 119 (1997) 208–213, <https://doi.org/10.1115/1.2888020>.
- [7] R. Pitz-Paal, B. Hoffschmidt, M. Böhmer, M. Becker, Experimental and numerical evaluation of the performance and flow stability of different types of open volumetric absorbers under non-homogeneous irradiation, *Sol. Energy* 60 (1997) 135–150, [https://doi.org/10.1016/S0038-092X\(97\)00007-8](https://doi.org/10.1016/S0038-092X(97)00007-8).
- [8] M. Böhmer, C. Chaza, The ceramic foil volumetric receiver, *Sol. Energy Mater.* 24 (1991) 182–191, [https://doi.org/10.1016/0165-1633\(91\)90058-S](https://doi.org/10.1016/0165-1633(91)90058-S).
- [9] J. Karni, A. Kribus, R. Rubin, P. Doron, The “porcupine”: a novel high-flux absorber for volumetric solar receivers, *J. Sol. Energy Eng.* 120 (1998) 85–95, <https://doi.org/10.1115/1.2888060>.
- [10] B. Hoffschmidt, R. Pitz-Paal, M. Böhmer, T. Fend, P. Rietbrock, 200 kWth open volumetric air Receiver (HiTRec) of DLR reached 1000 °C average outlet temperature at PSA, *J. Phys. IV* (1999) Pr3–556, <https://doi.org/10.1051/jp4:1999387>. France 09.
- [11] B. Hoffschmidt, F.M. Téllez, A. Valverde, J. Fernández, V. Fernández, Performance evaluation of the 200-kWth HiTRec-II open volumetric air receiver, *J. Sol. Energy Eng.* 125 (2003) 87–94, <https://doi.org/10.1115/1.1530627>.
- [12] T. Fend, R. Pitz-Paal, O. Reutter, J. Bauer, B. Hoffschmidt, Two novel high-porosity materials as volumetric receivers for concentrated solar radiation, *Sol. Energy Mater. Sol. Cell.* 84 (2004) 291–304, <https://doi.org/10.1016/j.solmat.2004.01.039>.
- [13] T. Fend, B. Hoffschmidt, R. Pitz-Paal, O. Reutter, P. Rietbrock, Porous materials as open volumetric solar receivers: experimental determination of thermophysical and heat transfer properties, *Energy* 29 (2004) 823–833, [https://doi.org/10.1016/S0360-5442\(03\)00188-9](https://doi.org/10.1016/S0360-5442(03)00188-9).
- [14] Th Fend, P. Schwarzbözl, O. Smirnova, D. Schöllgen, C. Jakob, Numerical investigation of flow and heat transfer in a volumetric solar receiver, *Renew. Energy* 60 (2013) 655–661, <https://doi.org/10.1016/j.renene.2013.06.001>.
- [15] Z. Wu, Z. Wang, Fully coupled transient modeling of ceramic foam volumetric solar air receiver, *Sol. Energy* 89 (2013) 122–133, <https://doi.org/10.1016/j.solener.2012.12.016>.
- [16] S. Mey, C. Caliot, G. Flamant, A. Kribus, Y. Gray, Optimization of high temperature SiC volumetric solar absorber, *Energy Proc.* 49 (2014) 478–487, <https://doi.org/10.1016/j.egypro.2014.03.051>.
- [17] M. Offergeld, K. Busch, P. Schwarzbözl, R. Uhlig, W. Schneider, A. Rosselló, B. Müller, VoCoRec - a novel two-stage volumetric conical receiver, *AIP Conf. Proc.* 2815 (2023) 100006, <https://doi.org/10.1063/5.0148830>.
- [18] J.E.P. Navalho, L.G.G. Silva, J.C.F. Pereira, An investigation on the thermal performance of volumetric solar absorbers with lateral wall power losses, *Sol. Energy* 282 (2024) 112937, <https://doi.org/10.1016/j.solener.2024.112937>.
- [19] S. Du, Y. He, D. Li, Z. Liu, M. Li, Comparison of direct pore-scale and volume-averaging methods for the performance evaluation of porous volumetric solar receiver, *J. Therm. Sci.* 33 (2024) 1607–1617, <https://doi.org/10.1007/s11630-024-2019-8>.
- [20] A.L. Avila-Marín, J. Fernandez-Reche, J.A. Carballo, M.E. Carra, S. Gianella, L. Ferrari, D. Sanchez-Señoran, CFD analysis of the performance impact of geometrical shape on volumetric absorbers in a standard cup, *Renew. Energy* 201 (2022) 256–272, <https://doi.org/10.1016/j.renene.2022.10.032>.
- [21] C. Pabst, G. Feckler, S. Schmitz, O. Smirnova, R. Capuano, P. Hirth, T. Fend, Experimental performance of an advanced metal volumetric air receiver for Solar Towers, *Renew. Energy* 106 (2017) 91–98, <https://doi.org/10.1016/j.renene.2017.01.016>.
- [22] S. Luque, G. Menéndez, M. Roccabruna, J. González-Aguilar, L. Crema, M. Romero, Exploiting volumetric effects in novel additively manufactured open solar receivers, *Sol. Energy* 174 (2018) 342–351, <https://doi.org/10.1016/j.solener.2018.09.030>.
- [23] A.L. Avila-Marín, C. Caliot, M. Alvarez de Lara, J. Fernandez-Reche, M.J. Montes, A. Martinez-Tarifa, Homogeneous equivalent model coupled with P1-approximation for dense wire meshes volumetric air receivers, *Renew. Energy* 135 (2019) 908–919, <https://doi.org/10.1016/j.renene.2018.12.061>.
- [24] R.T. Broeske, P. Schwarzbözl, L. Birkigt, S. Vasic, S. Dung, T. Doerbeck, B. Hoffschmidt, Experimentally assessed efficiency improvement of innovative 3D-shaped structures as volumetric absorbers, *Renew. Energy* 218 (2023) 119220, <https://doi.org/10.1016/j.renene.2023.119220>.
- [25] R. Capuano, T. Fend, H. Stadler, B. Hoffschmidt, R. Pitz-Paal, Optimized volumetric solar receiver: thermal performance prediction and experimental validation, *Renew. Energy* 114 (2017) 556–566, <https://doi.org/10.1016/j.renene.2017.07.071>.
- [26] A.L. Avila-Marín, M. Alvarez de Lara, J. Fernandez-Reche, Experimental results of gradual porosity volumetric air receivers with wire meshes, *Renew. Energy* 122 (2018) 339–353, <https://doi.org/10.1016/j.renene.2018.01.073>.
- [27] A.L. Avila-Marín, M. Alvarez-Lara, J. Fernandez-Reche, Experimental results of gradual porosity wire mesh absorber for volumetric receivers, *Energy Proc.* 49 (2014) 275–283, <https://doi.org/10.1016/j.egypro.2014.03.030>.
- [28] A.L. Avila-Marín, CFD parametric analysis of wire meshes open volumetric receivers with axial-varied porosity and comparison with small-scale solar receiver tests, *Renew. Energy* 193 (2022) 1094–1105, <https://doi.org/10.1016/j.renene.2022.05.060>.
- [29] R.T. Broeske, P. Schwarzbözl, B. Hoffschmidt, A new partitioned 1D LTNE continuum model for the simulation of 3D-shaped honeycomb absorbers, *Sol. Energy* 236 (2022) 533–547, <https://doi.org/10.1016/j.solener.2022.02.024>.
- [30] D. Sanchez-Señoran, M.A. Reyes-Belmonte, M. Farchado, A.L. Avila-Marín, Micro-scaling numerical model coupled with experimental validation of the pressure drop for dense wire mesh, *Int. J. Heat Mass Tran.* 223 (2024) 125263, <https://doi.org/10.1016/j.ijheatmasstransfer.2024.125263>.
- [31] D. Sanchez-Señoran, M.A. Reyes-Belmonte, J. Fernandez-Reche, A.L. Avila-Marín, Numerical simulation of convective heat transfer coefficient in wire mesh absorbers with fixed porosity, *Results in Engineering* 17 (2023) 100830, <https://doi.org/10.1016/j.rineng.2022.100830>.
- [32] D. Sanchez-Señoran, M.A. Reyes-Belmonte, M. Farchado, M. Casanova, A.L. Avila-Marín, Numerical characterisation of the convective heat transfer and fluid flow for inline woven wire meshes in solar volumetric receivers, *Int. Commun. Heat Mass Tran.* 159 (2024) 108269, <https://doi.org/10.1016/j.icheatmasstransfer.2024.108269>.
- [33] A.L. Avila-Marín, J. Fernandez-Reche, M. Casanova, C. Caliot, G. Flamant, Numerical simulation of convective heat transfer for inline and stagger stacked plain-weave wire mesh screens and comparison with a local thermal non-equilibrium model, *AIP Conf. Proc.* 1850 (2017) 030003, <https://doi.org/10.1063/1.4984346>.
- [34] J.-W. Park, D. Ruch, R. Wirtz, Thermal/fluid characteristics of isotropic plain-weave screen laminates as heat exchange surfaces, in: 40th AIAA Aerospace Sciences Meeting & Exhibit, American Institute of Aeronautics and Astronautics, 2002, <https://doi.org/10.2514/6.2002-208>.
- [35] Z. Zhao, Y. Peles, M.K. Jensen, Properties of plain weave metallic wire mesh screens, *Int. J. Heat Mass Tran.* 57 (2013) 690–697, <https://doi.org/10.1016/j.ijheatmasstransfer.2012.10.055>.
- [36] J. Xu, R.A. Wirtz, In-plane effective thermal conductivity of plain-weave screen laminates, *IEEE Trans. Compon. Packag. Technol.* 25 (2002) 615–620, <https://doi.org/10.1109/TCAPT.2002.807993>.

- [37] X. Jia, J. Yang, M. Lin, Q. Wang, Pore scale analysis for flow and thermal characteristics in metallic woven mesh, *Int. J. Heat Mass Tran.* 218 (2024) 124740, <https://doi.org/10.1016/j.ijheatmasstransfer.2023.124740>.
- [38] B.T. York, B.D. MacDonald, Influence of misalignment and spacing on the pressure drop through wire mesh Stirling engine regenerators, *Energy Convers. Manag.* 245 (2021) 114588, <https://doi.org/10.1016/j.enconman.2021.114588>.
- [39] W.S. Chang, Porosity and effective thermal conductivity of wire screens, *J. Heat Tran.* 112 (1990) 5–9, <https://doi.org/10.1115/1.2910365>.
- [40] J. Xu, J. Tian, T.J. Lu, H.P. Hodson, On the thermal performance of wire-screen meshes as heat exchanger material, *Int. J. Heat Mass Tran.* 50 (2007) 1141–1154, <https://doi.org/10.1016/j.ijheatmasstransfer.2006.05.044>.
- [41] C. Li, G.P. Peterson, The effective thermal conductivity of wire screen, *Int. J. Heat Mass Tran.* 49 (2006) 4095–4105, <https://doi.org/10.1016/j.ijheatmasstransfer.2006.03.031>.
- [42] Frank Kreith, Raj M. Manglik, Mark S. Bohn, *Principles of Heat Transfer*, seventh ed., Cengage Learning, Inc, 2011.
- [43] K. Vafai, C.L. Tien, Boundary and inertia effects on flow and heat transfer in porous media, *Int. J. Heat Mass Tran.* 24 (1981) 195–203, [https://doi.org/10.1016/0017-9310\(81\)90027-2](https://doi.org/10.1016/0017-9310(81)90027-2).
- [44] Daniel Sanchez-Señoran, Miguel A. Reyes-Belmonte, Jesus Fernandez-Reche, L. Antonio, Avila-marin, numerical simulation of pressure drop in wire mesh absorbers with fixed porosity. Poster SolarPaces Conference 2022, 2022.
- [45] M.E. Nimvari, N.F. Jouybari, Q. Esmaili, A new approach to mitigate intense temperature gradients in ceramic foam solar receivers, *Renew. Energy* 122 (2018) 206–215, <https://doi.org/10.1016/j.renene.2018.01.117>.
- [46] F. Wang, J. Tan, S. Yong, H. Tan, S. Chu, Thermal performance analyses of porous media solar receiver with different irradiative transfer models, *Int. J. Heat Mass Tran.* 78 (2014) 7–16, <https://doi.org/10.1016/j.ijheatmasstransfer.2014.06.035>.
- [47] Siemens Digital Industries Software, User Guide - Simcenter STAR-CCM+ Documentation, SIEMENS, 2020.
- [48] H.K. Versteeg, W. Malalasekera, An introduction to computational fluid dynamics. The Finite Volume Method, Longman Scientific and Technical, 1995.
- [49] Suhas V. Patankar, *Numerical Heat Transfer and Fluid Flow*, McGraw-Hill, 1980.
- [50] S. Du, M.-J. Li, Y.-L. He, S. Shen, Conceptual design of porous volumetric solar receiver using molten salt as heat transfer fluid, *Appl. Energy* 301 (2021) 117400, <https://doi.org/10.1016/j.apenergy.2021.117400>.
- [51] Q. Li, F. Bai, B. Yang, Z. Wang, B. El Hefni, S. Liu, S. Kubo, H. Kiriki, M. Han, Dynamic simulation and experimental validation of an open air receiver and a thermal energy storage system for solar thermal power plant, *Appl. Energy* 178 (2016) 281–293, <https://doi.org/10.1016/j.apenergy.2016.06.056>.
- [52] F. Gomez-Garcia, J. González-Aguilar, G. Olalde, M. Romero, Thermal and hydrodynamic behavior of ceramic volumetric absorbers for central receiver solar power plants: a review, *Renew. Sustain. Energy Rev.* 57 (2016) 648–658, <https://doi.org/10.1016/j.rser.2015.12.106>.
- [53] S. Du, T. Xia, Y.-L. He, Z.-Y. Li, D. Li, X.-Q. Xie, Experiment and optimization study on the radial graded porous volumetric solar receiver matching non-uniform solar flux distribution, *Appl. Energy* 275 (2020) 115343, <https://doi.org/10.1016/j.apenergy.2020.115343>.
- [54] M.A. Schuetz, L.R. Glicksman, A basic study of heat transfer through foam insulation, *J. Cell. Plast.* 20 (1984) 114–121, <https://doi.org/10.1177/0021955X8402000203>.
- [55] A.L. Avila-Marin, J. Fernandez-Reche, A. Martinez-Tarifa, Modelling strategies for porous structures as solar receivers in central receiver systems: a review, *Renew. Sustain. Energy Rev.* 111 (2019) 15–33, <https://doi.org/10.1016/j.rser.2019.03.059>.
- [56] X. Chen, X.-L. Xia, X.-L. Meng, X.-H. Dong, Thermal performance analysis on a volumetric solar receiver with double-layer ceramic foam, *Energy Convers. Manag.* 97 (2015) 282–289, <https://doi.org/10.1016/j.enconman.2015.03.066>.
- [57] Z. Wu, C. Caliot, G. Flamant, Z. Wang, Coupled radiation and flow modeling in ceramic foam volumetric solar air receivers, *Sol. Energy* 85 (2011) 2374–2385, <https://doi.org/10.1016/j.solener.2011.06.030>.
- [58] M.A. Gómez, D. Patiño, R. Comesaña, J. Porteiro, M.A. Álvarez Feijoo, J.L. Míguez, CFD simulation of a solar radiation absorber, *Int. J. Heat Mass Tran.* 57 (2013) 231–240, <https://doi.org/10.1016/j.ijheatmasstransfer.2012.09.061>.
- [59] M. Faizan, A. Almerbati, B.S. Yilbas, A novel approach for volumetric solar receiver performance assessments, *Appl. Therm. Eng.* 211 (2022) 118487, <https://doi.org/10.1016/j.applthermaleng.2022.118487>.
- [60] J.E.P. Navalho, J.C.F. Pereira, Investigating the impact of modeling assumptions and closure models on the steady-state prediction of solar-driven methane steam reforming in a porous reactor, *Energy Convers. Manag.* 295 (2023) 117577, <https://doi.org/10.1016/j.enconman.2023.117577>.
- [61] S. Sharma, P. Talukdar, Thermo-mechanical performance enhancement of volumetric solar receivers using graded porous absorbers, *Energy* 304 (2024) 132070, <https://doi.org/10.1016/j.energy.2024.132070>.
- [62] J. Zhao, M. Sun, L. Zhang, C. Hu, D. Tang, L. Yang, Y. Song, Forced convection heat transfer in porous structure: effect of morphology on pressure drop and heat transfer coefficient, *J. Therm. Sci.* 30 (2021) 363–393, <https://doi.org/10.1007/s11630-021-1403-x>.