



# Numerical characterisation of the convective heat transfer and fluid flow for inline woven wire meshes in solar volumetric receivers

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## ABSTRACT

In solar tower systems, metallic woven wire meshes following an inline arrangement are a promising geometry for commercial application in open volumetric receivers (OVRs). To assess their potential, two main parameters, with two different methodologies, need to be analysed in depth: the convective heat transfer coefficient (HTC) and the pressure drop per unit length ( $\Delta P/L$ ). In this study, three wire diameters, 0.7, 0.4 and 0.1 mm, with a fixed porosity of 80 %, have been selected to establish a baseline for the HTC and the  $\Delta P/L$  sensitivity studies. In the HTC methodology, six air inlet velocities between 0.5 and 5 m/s, and constant solid matrix temperatures of 700, 1100 and 1500 K, have been evaluated. For the  $\Delta P/L$  methodology, six air inlet velocities between 0.5 and 5 m/s and a constant solid matrix temperature of 300 K was used.

For each thermo-fluid-dynamic attribute, HTC and  $\Delta P/L$ , and each wire diameter (0.7, 0.4 and 0.1 mm) with 80 % porosity, an individual correlation is presented resulting in a total of six individual correlations. In addition, two generalised correlations, one of each thermo-fluid-dynamic attribute, for 80 % porosity and wire diameter ranging from 0.1 to 0.7 mm are shown, taking into account the specific geometrical parameters of the wire mesh lattices. The generalised correlations obtained in the study present deviations under 7 % from the CFD results in both thermo-fluid-dynamic attributes, showing results as good as the ones obtained with the individual correlations. As a result, it is demonstrated that the geometrical parameters must be included in the correlations used to predict the thermo-fluid-dynamic attributes in wire mesh lattices. All the correlations have been validated through porous models with good agreement and they would work for future optimisation studies.

## 1. Introduction

In recent years, climate change has increased the interest in renewable energy as a way of reducing pollution. Concentrated Solar Power (CSP) technology, also known as Solar Thermal Electricity technology, is one of the most promising renewable energies due to its capability to store and dispatch energy when it is required, acting as a base energy producer. However, CSP technology has a higher levelised cost of energy than other renewable energies such as photovoltaic or wind energy [1]. For that reason, improving dispatch capabilities and cutting-edge research and development are needed to make this technology attractive to industry.

Improvements in the heliostat field [2], irradiance measurements [3]

and different receiver technologies [4] have been identified as the primary areas for R&D in Central Receiver System (CRS) technology. In the area of receiver components, the technology implemented commercially is based on tubular receivers with molten salts, while the most favourable lines of research are tubular receivers with new fluids, particle receivers and gas receivers [5,6].

Tubular receivers are based on vertical tubes which are heated by reflected radiation from the heliostat field, and consequently, the fluid that flows along the tubes is heated, typically molten salts. The receivers used in the tubular technology are cavity and cylindrical receivers. Particle receivers operate by heating particles through reflected radiation from the heliostat field, employing various concepts such as free-falling, obstructed, rotating kiln/centrifugal, fluidised, and gravity-

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driven particle receivers. As a result, different types of concepts are analysed; however, the most commonly employed receiver design for the particles receivers is the cavity receiver. And, gas receiver technology employs a gas heated through porous materials, tubes and micro-channel configurations [7–10].

In this work, open volumetric receivers (OVRs) are prioritised, as their main advantages include a simple design and the use of an inexpensive working fluid, offering great potential. To deal with the high operational temperatures required, OVRs are manufactured with ceramic or metallic materials. Ceramic materials withstand operational temperatures up to 1500 °C, but show material degradation after a few thermal cycles [11]. Metals have a lower operational temperature, up to 1000 °C, but offer greater mechanical stability [12]. In addition, metallic materials remain of interest as an intermediate step between the current state-of-the-art molten salts and ceramic absorbers. Despite significant efforts to achieve thermal efficiencies above 90 % with air temperatures of 1000 °C in ceramic absorbers, this goal has not yet been reached.

However, materials are not the only factor that affects the thermal efficiency of the OVRs. The convective heat transfer coefficient (HTC) and the pressure drop are the most important attributes in OVRs, and they are both dependent on the geometrical characteristics of the absorber [11,12]. As a result, to understand the effect of the geometry and optimise the structure of OVRs, correlations from both thermo-fluid-dynamic attributes are necessary.

The first experimental woven screen and crossed-rod matrices were tested by Tong and London [13] in 1956, for different engineering applications, e.g., heat exchangers and the nuclear industry. In this study, heat transfer and flow friction were analysed in relation to geometrical characteristics with the aim of obtaining a Reynolds number range up to 100,000. The main geometrical characteristics of the six matrices in the study were a porosity range from 0.6 to 0.83, a rod diameter of 0.375 in. and a random stacking configuration. In that work, an experimental method was presented as well as global HTC and flow friction correlations for the woven screen and crossed-rod geometries. Due to the potential of the woven screen and crossed-rod matrices in engineering applications, in 1960, London et al. [14] continued the previous study adding one more sample with 0.5 porosity and the same 0.375 in. rod diameter. However, in this work, instead configurations were also implemented. In terms of HTC, the results showed no marked differences, whereas, in terms of flow friction, there were significant differences when porosity exceeded 0.7.

After these studies in the 50s and 60s, a research line on metal porous materials opened for distinct applications [15–20]. Some of the most relevant work of the current century on HTC and  $\Delta P/L$  in distinct fields are summarised next. Note that in the following, metal foams will be discussed first and then wire mesh matrices.

For engineering applications with metal foams, in 2005, Dukhan et al. [21] combined experimental with numerical simulations for engineering applications. Dukhan et al. implemented a one-dimensional HTC analysis in metal foams with a 0.918 porosity and a specific surface of 804 m<sup>2</sup>/m<sup>3</sup>. The main outcome of this work was experimental verification of the numerical simulations for metal foams with good agreement. In 2010, Kopanidis et al. [22] analysed metal foams through a novel 3D detailed numerical simulations of flow conjugated with HTC. The porosity of the metal foams was 0.97, the pore densities were 10 and 40 PPI, with 6.9 mm and 2.1 mm pore diameters, respectively. Later, in 2015, Liu et al. [23] experimentally analysed the HTC and pressure drop of supercritical CO<sub>2</sub> in tubes partially filled with metal foam. The porosities of the samples were 0.85, 0.9 and 0.95 with pore densities of 20, 40 and 60 PPI respectively. This work presented a forced convective HTC correlation and pressure drop correlation based on structural parameters. In 2017, Xia et al. [24] studied the convective HTC of metal foams experimentally. The metal foam samples had porosities between 0.87 and 0.97 with pore densities ranging from 10 to 40 PPI. The conclusions of the work presented the experimental data adjusted by the

Dietrich correlation [25] and a correlation based on the experimental data for a Reynolds number range from 20 to 1000. In a 2023 study, Jafarizadeh et al. [26] demonstrated a new approach for predicting pressure drop in metal foams through 3D micro-scaling numerical simulations and machine learning. A total of 217 foams were used with a porosity range from 65 to 95 % and a pore diameter ranging from 200 to 1000 µm. New correlations for the pressure drop were developed through artificial intelligence.

Then, for heat exchangers, in 2015, an important study in metal foams was carried out by Zafari et al. [27]. This work implemented a 3D detailed numerical simulation through tomography for fluids and HTC in metal foams with three porosities, 85, 90 and 95 %. Pressure drop correlations were presented for each porosity, and a generalised HTC correlation for all porosities was shown as well. And, in 2019, Jafarizadeh et al. [28] developed a 3D detailed numerical simulation of gas/solid HTC and pressure drop for metallic foams with a porosity range from 0.76 to 0.96 and pores diameters from 100 to 500 µm. The work presented a new generalised correlation of HTC for the porosity range from 0.76 to 0.96 for metal foams.

Metal foams are not the only metal porous media option. Woven wire meshes are another geometry which has been analysed. For heat dissipation applications, in 2016, Kurian et al. [29] evaluated the convective HTC and pressure drop in a vertical channel with wire meshes. The porosity of the wire meshes was from 0.77 to 0.85. The study presented individual correlations for the pressure drop for each wire mesh sample, and also, presented a HTC correlation which works for a Reynolds number range of 29 to 1400. Also in 2016, Ma et al. [30] analysed the flow and HTC for gradual woven wire meshes for heat dissipation. The wire meshes had porosities in the range of 25.6 to 55.1 %, divided into three samples, and an average pore size from 43.6 to 93.7 µm. They presented individual correlations for the pressure drop for each sample, and, for the HTC, corroborated the increase of the Nusselt number as porosity increased.

Recently, in 2021, Iwaniszyn et al. [31] evaluated the HTC and flow resistance of five woven wire gauzes for catalytic applications both experimentally and numerically. The pore diameter ranged from 1.5 to 3 mm and the porosity ranged from 0.73 to 0.78. The numerical simulations were based on 3D detailed simulations of a woven wire gauze. The authors presented new correlations for the HTC and flow resistance through woven metal wire gauzes. In heat exchanger applications, woven wire meshes were studied by Trilok G et al. [32] in 2022. Firstly, Trilok G et al. presented the analytic equations of the geometrical properties of wire meshes with distinct arrangements, two with the inline pattern and one with the staggered pattern, similarly to that of Zhao [33]. Then, they developed a 2D porous model to study the effect of the geometrical properties in the global HTC and pressure drop using correlations from the literature [34–36]. The work considered porosities from 0.8 to 0.95 and pore densities from 5 to 45 PPI to carry out the macro-scale simulations. The porous model presented a heat flux of 20 W in one of the walls leaving the opposing one adiabatic. The results focused on the global HTC of the wall and the pressure drop of the distinct meshes and velocities lower than 0.8 m/s. This work was validated with experimental data from Kurian et al. [29], but no HTC or pressure drop correlations were presented for the wire mesh arrangements.

Lastly, for woven wire meshes for CSP applications, in 2013 and 2014, Costa et al. [37,38] implemented 3D detailed numerical studies for the HTC and pressure drop for a wound, stacked woven wire matrix used as a regenerator in Stirling dishes. In the first study, 4 different woven wire meshes were used with a wire diameter ranging from 0.08 to 0.11 mm and a volumetric porosity range between 60 and 68 %. In the second study, 9 different wire meshes were analysed with a wire diameter ranging from 0.08 to 0.11 mm and a volumetric porosity range between 47 and 64 %. The main outcomes of these studies were, first, HTC correlations for the wound, stacked woven wire matrices and for a Reynolds number up to 400, and, second, pressure drop correlations

dependent on the wire diameter for a Reynolds number up to 400. For central receiver systems (CRS), in 2014, Avila-Marin et al. [39] experimentally evaluated three different wire meshes with 0.7, 0.63 and 0.5 mm wire diameter, and 61, 38 and 54 % porosities, respectively. In that work, the authors presented a new experimental setup as well as mixing the three different wire meshes in order to analyse gradual porosities. And finally, in 2017, Avila-Marin et al. [40] presented numerical simulations of convective HTC for IL and ST wire mesh screens. The wire diameter of the meshes was 1 mm with IL porosity and ST porosities of 83.9 and 70.1 %, respectively. In the work, the authors carried out 3D detailed simulations of the HTC for the wire mesh samples, obtaining individual correlations for the IL and ST arrangements. Moreover, a porous model under the LTNE state for wire meshes was presented with the corresponding experimental validation.

While there have been many studies of the behaviour of wire mesh for different applications over the years, to the best of the authors' knowledge, there has been no work specifically on the effect of geometrical characteristics on the HTC and pressure drop per unit length ( $\Delta P/L$ ) for IL woven wire meshes in volumetric CRS. Wire meshes continue to be of interest for CRS applications due to their efficiency, which, as Livshits indicates in her work [12], is comparable to that of ceramic absorbers. Additionally, wire meshes are more cost-effective, as they are commercially available on an industrial scale. They offer design flexibility, allowing for easy adjustment and customisation of geometrical parameters [6]. Moreover, the ability to achieve gradual porosity further enhances their versatility, which is considered their greatest advantage [41].

The IL pattern is a geometrical arrangement of interest because its attributes – high porosity, a low pressure drop and a low HTC – contrast with those of the ST arrangement. In addition, this work seeks to characterise how the IL geometry influences the convective HTC and the pressure difference independently of the material employed. Understanding the influence of geometry on the HTC is essential for designing OVRs because convection is a main heat transfer mechanism. Moreover, the pressure drop provides a direct measure of parasitic costs in CRS applications.

To sum up, this 3D micro-scale or detailed numerical study analyses the behaviour of the convective HTC and the  $\Delta P/L$  based on the geometrical characteristics of IL woven wire screens. The study is carried out for three wire meshes with 80 % fixed porosity. Individual correlations (ICs) of the convective HTC and  $\Delta P/L$  for three meshes are presented, as well as a generalised correlation (GC) based on the woven wire mesh geometry. Also, numerical validations of the convective HTC and  $\Delta P/L$  by means of the ICs and GCs are presented.

## 2. Wire mesh characterisation and cases studied

The characterisation of the IL stack woven wire meshes, together with the study cases adopted in this work are addressed in this section. Firstly, the geometrical parameters and properties of the IL wire meshes are defined. Then, secondly, in order to understand the behaviour of the convective HTC and the fluid flow through the IL lattice woven wire meshes, three cases with the same volumetric porosity are presented.

### 2.1. Characterisation of the lattice woven wire mesh

The main physical properties that characterise a lattice woven wire mesh are the wire diameter ( $d$ ), the wire spacing or pitch ( $w$ ), the mesh count ( $M = 1/(d + w)$ ) and the arrangement. Fig. 1 (a) represents the frontal and top view of a single woven wire mesh. Fig. 1 (b) shows the frontal view of the IL stack wire mesh which is evaluated in this work. An IL arrangement is defined by the alignment of the wire meshes within each other.

This IL arrangement contrasts with the ST arrangement presented in previous studies for being characterised by low pressure drops and convective HTC, together with high porosity [42,43].

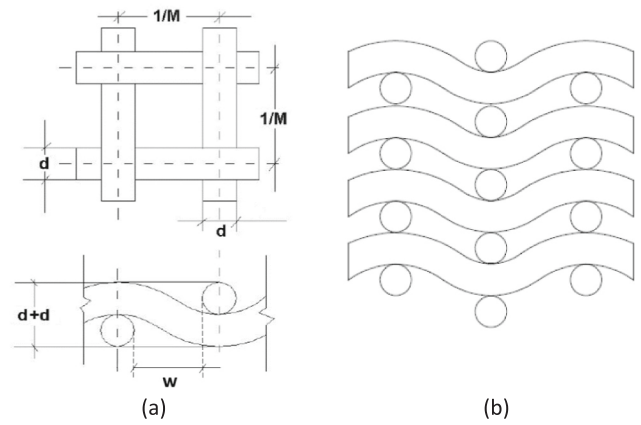


Fig. 1. Scheme of the IL arrangement: (a) top and front view of a single screen woven wire mesh and (b) front view of a woven wire mesh stack.

The convective HTC and fluid flow in the woven wire meshes are influenced by the specific surface area and the porosity respectively. Eq. (1) to Eq. (4) describe the volumetric porosity which is ratio of void spaces within a material between the material's total volume ( $\emptyset$ ), the specific surface area which is the ratio between the area that surrounds the wires and the total volume (air plus wires) considered ( $a_v$ ), the hydraulic diameter which relates the volumetric porosity with the specific surface area of non-circular shaped channels ( $d_h$ ) and parameter  $C$  which is a dimensionless constant depending on the geometrical parameters of the mesh,  $M$  and  $d$ , and extracted from Zhao et al. [33]. In summary, all the geometrical properties are a function of the physical properties of the wire mesh:  $d$ ,  $w$  and  $M$ . The compactness factor,  $c_f$ , takes a value of 1 for an uncompressed IL wire mesh pattern.

$$\emptyset = 1 - \frac{\pi \cdot M \cdot d}{4 \cdot c_f} \quad (1)$$

$$a_v \cdot d = \frac{\pi \cdot M \cdot d}{c_f} \quad (2)$$

$$d_h = \frac{4 \cdot \emptyset}{a_v} \quad (3)$$

$$C = \left( \frac{M \cdot d}{2} + \frac{1}{2 \cdot M \cdot d} \right) \arcsin \left( \frac{1}{\frac{M \cdot d}{2} + \frac{1}{2 \cdot M \cdot d}} \right) \quad (4)$$

### 2.2. Cases studied

Convective HTC and fluid flow are analysed in this work for three woven wire meshes with an IL arrangement like that shown in Fig. 1. The three meshes have wire diameters of 0.7, 0.4 and 0.1 mm, each with a fixed 80 % volumetric porosity. Table 1 summarises the geometrical parameters and properties of the cases studied.

The 80 % porosity is chosen because, according to a previous study, it appears to offer the most promising results [44]. Regarding the wire diameters, values larger than 0.7 mm tend to exhibit lower convective heat transfer coefficients, which is not desirable for our purposes [41].

Table 1

Geometrical properties and parameters of the IL lattice woven wire mesh stack cases.

| Case | d, mm | M, mm <sup>-1</sup> | $\emptyset_{SM}$ , % | $\emptyset_{IL}$ , % | $a_v$ , m <sup>-1</sup> | $d_h$ , mm |
|------|-------|---------------------|----------------------|----------------------|-------------------------|------------|
| # 1  | 0.7   | 0.35                | 80.0                 | 80.0                 | 1,138                   | 2.81       |
| # 2  | 0.4   | 0.61                | 80.0                 | 80.0                 | 1,998                   | 1.60       |
| # 3  | 0.1   | 2.44                | 80.0                 | 80.0                 | 8,081                   | 0.39       |

Conversely, wire diameters smaller than 0.1 mm result in significant pressure drops, which, in turn, negatively affect hydrodynamic performance [45]. That being said, the authors decided to select the wire diameter range presented in this work, from 0.1 to 0.7 mm.

### 3. Numerical methodologies

This section presents the two numerical studies used to determine the thermal and hydrodynamic behaviour of the cases shown in Table 1. The computational fluid dynamics (CFD) software STAR-CCM+ was used with both CFD methodologies, first, the convective HTC and then, the  $\Delta P/L$ . Both CFD methods have been validated previously in the literature [42,45]. A supercomputer with 5 nodes of 36 processors each, clock speeds of 3.1 GHz and 192 GB of memory per node was used to perform the 3D detailed simulations.

#### 3.1. Convective heat transfer coefficient method

A 3D micro-scale simulation was chosen to determine the convective HTC under different operational conditions (air inlet velocity and wire temperature) at thermal equilibrium. Fig. 2 shows the 3D geometry of a representative section of a real volumetric absorber. A section is used because even current supercomputer resources will not allow the 3D simulation of an entire volumetric absorber. The methodology of this study has been validated previously in the literature for both ceramic and metallic lattices [42,43,46].

##### 3.1.1. Governing equations and grid analysis

The wire mesh temperature and the air inlet velocity are the main variables in the convective HTC method. In this study both inputs were used to perform a sensitivity study in which different air inlet velocities between 0.5 and 5 m/s were modelled for constant temperatures of 700, 1100 and 1500 K. Note that radiative phenomenon was not considered in this study due to a wide range of constant and uniform temperatures in the wires was used to cover the OVR operating temperatures as indicated in the literature [40,46]. Additionally, numerical simulations are essential for isolating the convective heat transfer from the radiation mechanism.

The aim of the sensitivity study was to predict the HTC behaviour and to obtain both ICs of the convective HTC with the geometries of the three IL cases studied (Table 1) and a GC for an IL arrangement with 80 % porosity. In order to obtain the results for the HTC, the numerical simulations were governed by the following equations [47–49]:

Continuity equation for the fluid phase in steady state conditions:

$$\nabla \cdot (\rho \cdot \vec{v}) = 0 \quad (5)$$

Momentum equation for the fluid phase in steady state conditions:

$$\nabla \cdot (\rho \cdot \vec{v} \otimes \vec{x}) = \nabla \sigma + f_b \quad (6)$$

Energy equation for the fluid phase in steady state conditions:

$$\nabla \cdot (\rho \cdot c_p \cdot \vec{v} \cdot T) = \nabla \cdot (k \cdot \nabla T) + S \quad (7)$$

$S$  is defined as the volumetric rate of heat generation.

A fine grid is necessary to achieve accurate results. However, the fine grid needed for accuracy must be balanced with the computational time required to solve the governing equations. Nonetheless, in this study, polyhedral cells were used to obtain the most accurate results despite the computational time required. In addition, two prism layer cells were added around the wires because they allow the convective HTC to be obtained more accurately [49]. Fig. 3 shows the grid analysis for case #2 based on the base size in the mesh.

The base size of the cell is determinant for the performance of the mesh due to it being the parameter with the greatest impact for this type of simulations as seen in the literature [42,43]. The results of the grid sensitivity study for case #2 are presented in Table 2. The variation in the base size of the cell influences the number of cells and the value of the convective HTC at thermal equilibrium. Table 2 shows that the deviation becomes smaller when the number of cells is increased. A base size of 0.4 mm, which deviated by only 0.1 % from the best grid analysed for the convective HTC at thermal equilibrium zone was chosen for case #2 for providing great accuracy while saving some computing time.

The same procedure was followed in the rest of the cases, the details of which can be seen in Appendix 1. In all the cases the deviations were

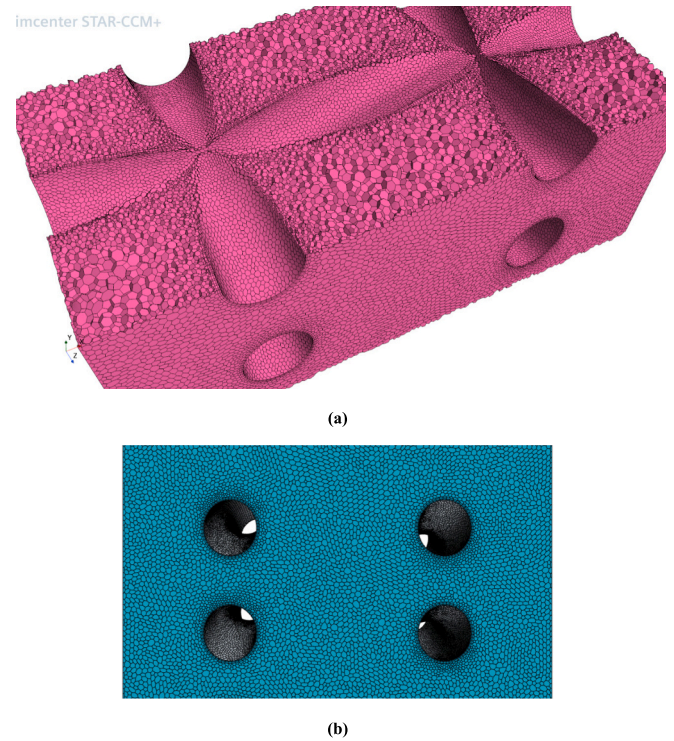


Fig. 3. Grid analysis for case #2 with base size of 0.4 mm: (a) core grid and (b) surface grid.

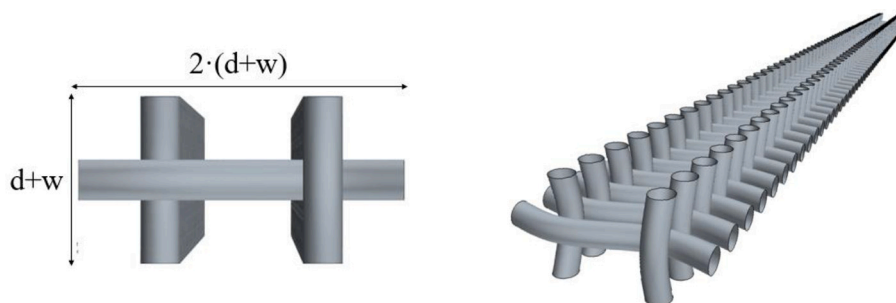


Fig. 2. Frontal (left) and isometric (right) view of the geometry studied in the convective HTC numerical analysis.

**Table 2**

Grid analysis for case #2. Convective HTC at 3 m/s with the temperature of the wires at 700 K and deviations for each base size.

| Base Size<br>mm | Number of cells | $h_{iv}$<br>kW/m <sup>3</sup> ·K | D<br>% |
|-----------------|-----------------|----------------------------------|--------|
| 1               | 1,884,321       | 289                              | 12.0   |
| 0.8             | 3,014,913       | 282                              | 9.3    |
| 0.4             | 13,663,695      | 258                              | 0.1    |
| 0.25            | 20,052,654      | 258                              | –      |

not significant, with values lower than 3 %. Table 3 presents the number of cells employed in each case to obtain the convective HTC.

The length of the stacks is determined by when the local thermal equilibrium can be reached as in previous literature [42]. This length is also critical in determining the number of cells needed, such that, as shown, the larger the wire diameter, the longer the stack length, and the greater the number of cells.

The dimensions of the domain are different for each case, the main ones are summarised in Table 1 and the frontal surface area is calculated with the formula shown in Fig. 2. In addition, Table 3 summarises the length of the wire mesh domain for each case, while the length of the rear air domain depends on the development of the flow, aiming to prevent reverse flow. The air domain at the frontal surface extends to a length equivalent to seven times the wire diameter.

### 3.1.2. Assumptions and boundary conditions

The governing equations in the convective HTC method require the establishment of assumptions and boundary conditions. The assumptions adopted in these simulations were:

- A steady state condition was assumed given that the local thermal equilibrium did not depend on time. Also, a laminar solver was adopted because 580 is the highest Reynold number in these simulations. Previous works demonstrated that this solver is valid [42,43], additionally in Appendix 2 can be found a comparison between the turbulence and laminar flow solvers for the studied cases at an air inlet velocity of 1 m/s and a wire temperature of 700 K because 1 m/s air inlet velocity is a typical air velocity in CRS applications [50]. The Reynold number was inferred by means of the hydraulic diameter.
- The fluid used in the numerical studies was air, which was assumed to be an ideal gas whose state equations could be solved through the segregated flow solver. In the STAR-CCM+ software, the properties of air are selected by default and dependent on the temperature.
- The segregated fluid temperature was used as a second order solver.
- Radiative effects were not considered. The reason is that the temperature of the solid matrix remained fixed to study different operational conditions at equilibrium, consequently the radiative effects approached zero. In addition, the air was regarded as a non-participating medium as presented in the literature [46].

The boundary conditions implemented in the simulations for the IL stack are depicted in Fig. 4 and summarised as follows:

- The inlet surface boundary was based on a “velocity inlet” condition with a uniform value defined at the entrance of the channel, and with a static temperature fixed at 300 K. Six inlet velocities between 0.5 and 5 m/s were studied in the sensitivity analysis.

**Table 3**

Number of cells needed for each HTC numerical case.

| Case                      | #1   | #2   | #3   |
|---------------------------|------|------|------|
| Wire diameter, mm         | 0.7  | 0.4  | 0.1  |
| Length of the stack, mm   | 89.6 | 68.8 | 12.8 |
| Number of cells, millions | 42.9 | 13.7 | 5.5  |

- The outlet surface boundary was analysed under a “pressure outlet” boundary condition with a zero-gauge pressure value.
- The interior wires boundary was based on the definition of all the wires as “non-slip walls” with temperature fixed and uniform along the domain at 700, 1100 and 1500 K. The objective of a wide temperature range study is to predict the behaviour of the convective HTC between the air and the IL arrangement at different typical operating temperatures neglecting the radiative effect [6,43]. The behaviour of the convective HTC was only analysed as a function of the geometrical properties, because the material is not considered in convective HTC studies [24,27,42,46,51]. As a consequence, the findings based on the wire geometry could be applied to both metallic and ceramic materials, both of which are typically found in CRS [6].
- The lateral air boundaries were analysed as a “symmetry plane” boundary condition for the four sides of the 3D geometry, based on the literature [46,51]. A “symmetry plane” boundary condition is an imaginary plane of symmetry mirroring the geometry and conditions from one side to the other. The resulting reduction in the computational domain does not affect the solution when the “symmetry plane” boundary condition is applied [49].

### 3.2. Pressure drop method

The determination of the  $\Delta P/L$  requires a representative 3D micro-scale simulation in order to obtain accurate results. Fig. 5 depicts the 3D detailed geometry implemented in this work. The  $\Delta P/L$  methodology adopted in this research has been validated previously [45,52,53].

In a previous study [45], it was concluded that the shape of the 3D geometries influences the differential pressure per unit length ( $\Delta P/L$ ). The boundary condition set to the lateral wall of the domain was also analysed, proving that the “symmetry plane” boundary resulted in the best numerical approximation to the real samples results. Because of these analyses, the representative 3D geometry simulated in this section has a larger surface at the inlet with a different shape, same boundary condition and fewer meshes (Fig. 5) than for the convective HTC geometry (Fig. 2). The physics for the pressure drop analysis is not as demanding as the convective HTC. In addition, in this study, the pressure drop is obtained per unit length meaning that the length of the absorber does not need to be as great as for the convective HTC analysis. These considerations allow the representative 3D geometry can be smaller and less computationally demanding.

#### 3.2.1. Governing equations and grid analysis

The simulations of the  $\Delta P/L$  method are influenced by the air inlet velocity and the air temperature. However, in the pressure drop study the air temperature was held constant at 300 K, as is usual in this type of simulations [26,52,54]. As a result, the parametric study implemented to analyse the  $\Delta P/L$  used only the air inlet velocity as the main variable. The aim of the parametric study was to predict the fluid flow behaviour to obtain ICs of the  $\Delta P/L$  for the three IL cases (Table 1) and a GC for 80 % porosity and the IL pattern. The numerical simulations were governed by Eq. (5) and (6).

The grid adopted in the three IL cases for the  $\Delta P/L$  simulations was based on the same grid settings presented in section 3.1.1. More details about the grid analysis can be seen in Appendix 3. As indicated, the grid adopted polyhedral cells with prism layer cells around the contours of the wires in order to obtain more accurate results. The number of cells varied depending on the wire diameter because the thickness of the volumetric absorber is based on 10 meshes in all cases. As a result, more cells were needed for meshes with larger diameter wires. Table 4 summarises the number of cells needed in each case studied.

The domain dimensions are different for each case. The main wire mesh geometrical parameters are summarised in Table 1 and the frontal surface is calculated with the formula shown in Fig. 5. In addition, Table 4 summarises the length of the wire mesh domain for each case,

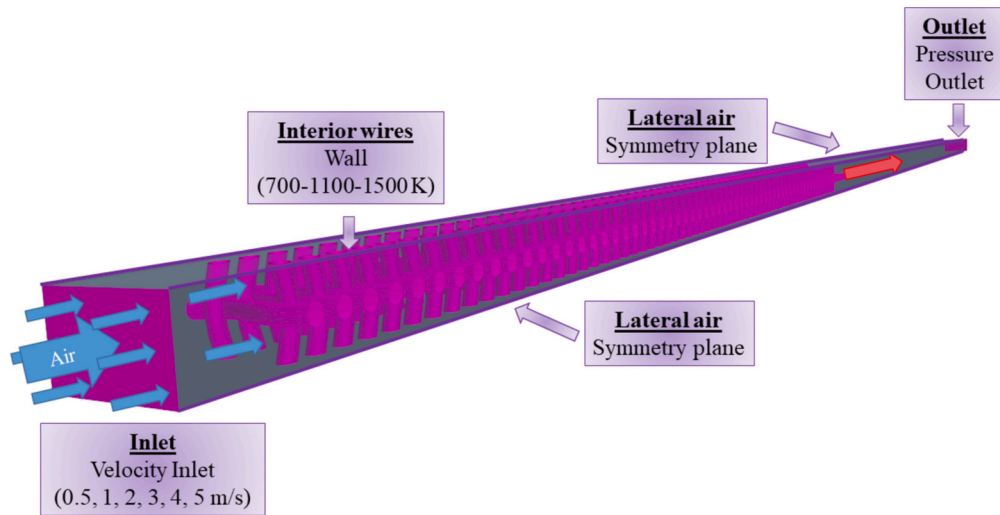


Fig. 4. Boundary conditions of the geometry used in the convective HTC numerical analysis.

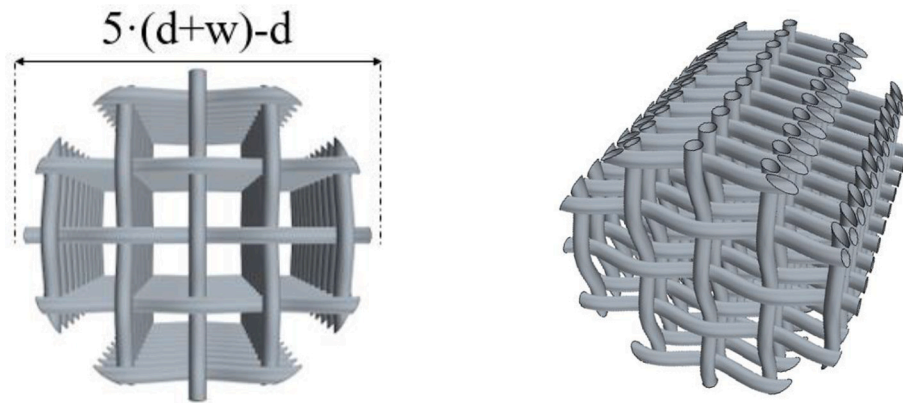


Fig. 5. Frontal (left) and isometric (right) view of the circular geometry used in the  $\Delta P/L$  numerical analysis.

Table 4

Number of cells needed in each pressure drop numerical case.

| Case                               | #1   | #2   | #3  |
|------------------------------------|------|------|-----|
| Wire diameter, mm                  | 0.7  | 0.4  | 0.1 |
| Length of the stack, mm            | 14   | 8    | 2   |
| Number of cells, millions of cells | 35.6 | 17.4 | 8.7 |

while the length of the rear air domain depends on the development of the flow, aiming to prevent reverse flow. The air domain at the frontal surface extends to a length equivalent to seven times the wire diameter. The magnitude used to analyse the pressure drop is the differential pressure divided by the length of the wire mesh absorber, for that reason less meshes are needed. Aside, the number of meshes needed was analysed in a previous work [45].

### 3.2.2. Assumptions and boundary conditions

Assumptions and boundary conditions must be established to apply the pressure drop method in the governing equations. The assumptions used in the  $\Delta P/L$  analysis have been validated in the literature and were as follows [45]:

- The steady state condition and laminar flow regime were applied in accordance with the HTC assumptions.
- The fluid used in the numerical studies was air.

- The fluid was assumed to be an ideal gas with constant density. The properties of air are based on a temperature of 300 K.
- The segregated flow solver was applied.

Fig. 6 depicts the boundary conditions implemented in the pressure drop methodology [45] and are summarised as follows:

- The inlet surface boundary was based on a “velocity inlet” condition with a uniform value defined at the entrance of the representative 3D geometry. The air inlet velocity tested were between 0.5 and 5 m/s.
- The outlet surface boundary was analysed under a “pressure outlet” boundary condition with a zero-gauge pressure value.
- The lateral air boundary was analysed as a “symmetry plane” boundary condition for the cylindrical side of the 3D geometry. To achieve a solution valid for a real geometry, the representative 3D geometry must apply the “symmetry plane” boundary condition as documented in the literature [45,49].
- Interior wire boundaries were based on the definition of all wires as “non-slip walls” and “adiabatic” conditions.

## 4. Results and discussion

The results of this work analysed stacked woven wire meshes with an IL arrangement, 80 % porosity and three different wire diameters (0.7, 0.4 and 0.1 mm) and are divided into three sections. The first section covers the analysis of the convective HTC providing ICs for each case

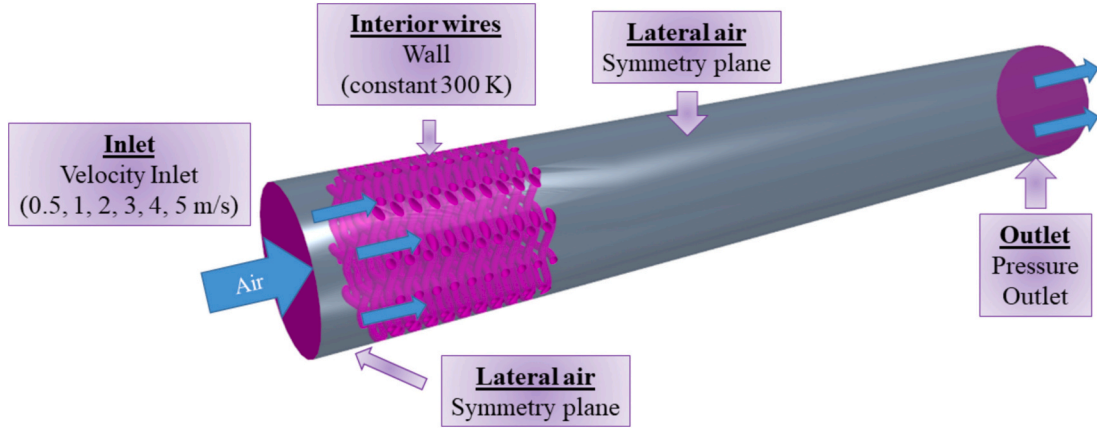


Fig. 6. Boundary conditions of the geometry used in the  $\Delta P/L$  numerical analysis.

presented in Table 1 and, subsequently, it presents a GC for the entire range of study. The second section mirrors the structure of the first section, focusing on the study of  $\Delta P/L$  instead. Lastly, the third section presents a numerical validation of the HTC and the  $\Delta P/L$  results.

#### 4.1. Convective heat transfer results

The local volumetric HTC,  $h_{lv}$ , is a key parameter in OVRs used in solar towers with concentrated solar energy because it couples the solid matrix and the circulating fluid energy equations. It is difficult to isolate the  $h_{lv}$  parameter experimentally, especially at high temperatures, for that reason, it has generally been evaluated numerically [43,52]. The local volumetric HTC is defined as follows:

$$h_{lv} = \frac{Q_{lv}}{(T_{ls} - T_{mlf})} \quad (8)$$

where  $Q_{lv}$  is the heat flux exchange within a certain volume, the  $T_{ls}$  is the local temperature of the solid matrix that is fixed in the sensitivity study (700, 1100, 1500 K), and,  $T_{mlf}$  is the mass-weighted average temperature of the fluid [42] which is calculated as follows:

$$T_{mlf} = \frac{1}{\int_S \dot{m} \cdot c_p \cdot dS} \int_S \dot{m} \cdot c_p \cdot T_{lf} \cdot dS \quad (9)$$

Where  $m$  (kg/s) is the mass flow rate,  $c_p$  (J/kg·K) is the specific heat and  $T_{lf}$  (K) is the mean fluid temperature. The mass-weighted average temperature is evaluated across all the thresholds defining the studied domain, as it is used to calculate the local volumetric HTC.

##### 4.1.1. Thermal attributes

This section points out the importance of the mass-weighted average temperature of the fluid ( $T_{mlf}$ ) and the numerical prediction of the HTC ( $h_{lv}$ ) in the thermal equilibrium zone. Fig. 7 depicts the profile of the  $T_{mlf}$  for the three geometries presented in Table 1 at an air inlet velocity of 1 m/s, as well as showing that the equilibrium between the wire matrix ( $T_{ls}$ ) and the fluid ( $T_{mlf}$ ) is reached at different relative depths in accordance with the geometrical properties of each case (Table 1) despite them having the same porosity, 80 %.

Comparing the cases with the greatest difference in wire diameters, case #1 with 0.7 mm and case #3 with 0.1 mm, the temperature was evaluated at three different points  $\left(\frac{z_l}{z_{maxi}}\right)_{i=1:3}$ : 0.2, 0.5 and 0.8. For a

$T_{ls}$  of 700 K, the  $T_{mlf}$  results for the case #1 are 572, 672 and 694.5 K respectively whereas for case #3, the  $T_{mlf}$  results are 699.6, 699.99 and 700 K. Repeating the evaluation process for a  $T_{ls}$  of 1500 K, the  $T_{mlf}$  results for case #1 are 1244, 1479 and 1498.7 K respectively, while for case #3 they are 1499.94, 1499.97 and 1500 K. These cases indicate

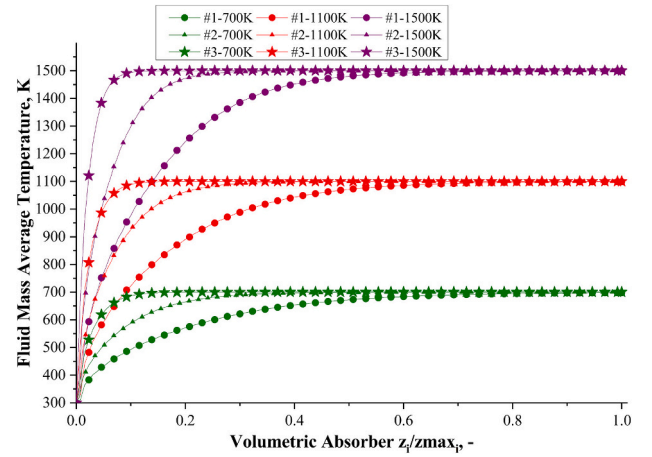


Fig. 7. Mass-weighted average temperature of the fluid profiles for the three IL woven wire mesh cases at an air inlet velocity of 1 m/s.

that, taking into account only the convective HTC, the temperature reaches equilibrium more quickly with smaller wire diameters.

After post-processing of the thermal attributes results, similar to that presented in Sanchez-Señoran [42], the convective HTC of the three IL woven wire mesh cases are compared in Fig. 8 showing when thermal equilibrium was reached. Fig. 8 depicts three distinct zones:

- Fluid Inlet Zone (FIZ): in this zone the fluid faces the first woven wire mesh of the absorber causing an absolute maximum followed by a relative minimum in the convective HTC.
- Fluid Heating Zone (FHZ): the progressive heating of the fluid as it flows through the absorber is indicated by the increasing slope in the convective HTC.
- Thermal Equilibrium Zone (TEZ): the fluid and the solid matrix are on the verge of achieving a stable and balanced thermodynamic state. This state is associated with a constant convective HTC values.

Given that thermal equilibrium zone in the absorber is influenced by both the wire temperature and the air inlet velocity, these parameters with the geometrical properties of the porous media also affect when the thermal equilibrium is reached.

Importantly, Fig. 8 illustrates that higher equilibrium HTCs are achieved by meshes with smaller wire diameters. Comparing the two cases with the greatest differences, case #1 (0.7 mm) and case #3 (0.1 mm), at a solid matrix temperature of 700 K, the respective HTC at thermal equilibrium are 76.6 and 3,586.9 kW/(m<sup>3</sup>·K). In other words,

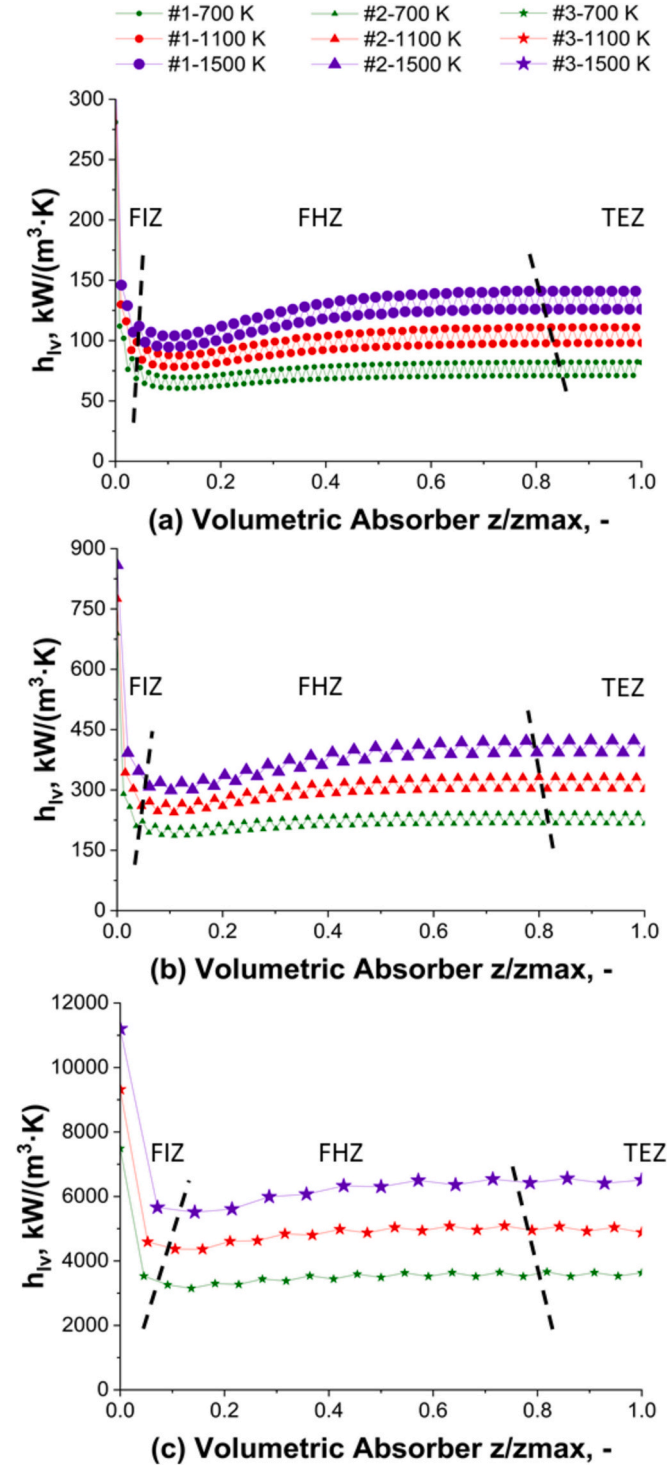


Fig. 8. Convective HTC of IL woven wire mesh absorbers with an air inlet velocity of 1 m/s for: (a) case #1, (b) case #2 and (c) case #3.

Table 5

HTC at thermal equilibrium zone for the three cases (#1, #2 and #3) at an air inlet velocity of 1 m/s and the  $T_{is}$  simulated.

| Case                          | #1  |      |      | #2  |      |      | #3    |       |       |
|-------------------------------|-----|------|------|-----|------|------|-------|-------|-------|
| $T_{is}$ , K                  | 700 | 1100 | 1500 | 700 | 1100 | 1500 | 700   | 1100  | 1500  |
| TEZ HTC, kW/m <sup>2</sup> ·K | 77  | 105  | 134  | 229 | 317  | 408  | 3,587 | 5,006 | 6,465 |

the HTC at thermal equilibrium in case #3 is 46 times that of case #1.

Table 5 presents the HTC at thermal equilibrium for the three  $T_{is}$  and the three cases of study. Comparing the HTC at thermal equilibrium for the maximum and minimum  $T_{is}$  and at an air inlet velocity of 1 m/s, in case #1, the HTC at thermal equilibrium increases by 74 %, with the increase in temperature, considering the HTC as the only heat transfer possibility. For case #2, the HTC at thermal equilibrium increases 78 % from the minimum to the maximum  $T_{is}$ , whereas for case #3, the HTC at thermal equilibrium zone increases 80 % under an air inlet velocity of 1 m/s and the same  $T_{is}$  (700 and 1500 K). Also, the increase in wire diameter augments the percentage difference between the maximum and the minimum HTC at thermal equilibrium under the same temperature and air inlet velocity conditions.

In summary, higher convective HTC are achieved with smaller wire diameters, while larger wire diameters result in lower HTC values. In addition, overall, the higher the wire temperature, the higher the HTC at thermal equilibrium.

#### 4.1.2. Individual correlations (ICs)

Nusselt, Reynolds and Prandtl numbers are considered to be key dimensionless parameters in the behaviour of the convective HTC [25,40,46,55,56]. Eq. (10) to Eq. (12) are used to calculate the Nusselt number, the Reynolds number and the Prandtl number at the thermal equilibrium.

$$Nu_{lv} = \frac{h_{lv} \cdot d_h^2}{k} \quad (10)$$

$$Re = \frac{\rho \cdot d_h \cdot v}{\mu} \quad (11)$$

$$Pr = \frac{\mu \cdot c_p}{k} \quad (12)$$

In these equations  $\rho$  is the density of the air,  $\mu$  is the dynamic viscosity,  $d_h$  is the hydraulic diameter,  $v$  is the superficial velocity at the inlet,  $k$  is the thermal conductivity and  $c_p$  is the specific heat.

Once the dimensionless numbers were obtained in post-process work for all sensitivity studies conducted and according to the literature [19,24,25,31,42,55,56], ICs of the form presented in Eq. (13), were fitted to predict the convective HTC in porous geometries.

$$Nu_{lv} = a \cdot Re^b \cdot Pr^c \quad (13)$$

where “a”, “b”, and “c” are dimensionless constants for each case of study.

Table 6 shows the dimensionless constants (“a”, “b”, “c”), the mean absolute error (MeAE) and the range of validity for each IC based on each case of study presented in Table 1. The average deviation varies from a minimum value of 0.5 % for case #3 to a maximum value of 9.3 % for case #1.

Table 6

Dimensionless constants, mean absolute error and range of validity of each HTC-IC at thermal equilibrium zone.

| Case | “a”   | “b”      | “c”    | MeAE, % | Re range | Pr range  |
|------|-------|----------|--------|---------|----------|-----------|
| #1   | 8.284 | 0.2665   | 1.423  | 9.3     | 6–222    |           |
| #2   | 10.01 | 0.1031   | 0.3852 | 4.4     | 4–133    | 0.69–0.71 |
| #3   | 9.525 | −0.00044 | −0.326 | 0.5     | 1–31     |           |

#### 4.1.3. Generalised correlation (GC)

Generalised correlations (GCs) are used in OVR optimisation studies for CRS [57,58] as they offer relevant thermo-fluid-dynamic information for proper OVR design. Additionally, the use of GCs is due to its easier implementation and saving both computational time and storage when a range of geometrical properties is evaluated. This section presents a GC of the Nusselt number with the Reynolds number and a geometrical factor (F) assuming a constant Prandtl number for woven wire mesh geometry used in OVRs.

The most common general formula to correlate the Nusselt number with the Reynolds and Prandtl numbers is shown in Eq. (13). However, Eq. (13) does not consider the influence of the geometrical parameters in predicting the convective HTC. Many authors have obtained GCs including geometrical factors [59–61]. In this work, Eq. (14), similar to that of Wakao and Kaguei [61], presents a GC which relates the Nusselt with the Reynolds number and a geometrical factor (F) for woven wire meshes.

$$Nu = (m_1) \cdot Re^{m_2} + F^{m_3} \quad (14)$$

Here F is a dimensionless geometrical factor that is equal to " $a_v d_h$ ", the Prandtl number is assumed to be constant with a value of 0.7 integrated in the coefficient  $m_1$ , and the Reynolds number is a function of the superficial velocity at the inlet as defined by Eq. (11). Then,  $m_1$ ,  $m_2$  and  $m_3$  are dimensionless constants for the IL geometry with 80 % porosity and wire diameter varying from 0.1 to 0.7 mm, which are obtained through MATLAB fitting tool.

Table 7 shows the dimensionless coefficients that correlate the HTC at thermal equilibrium with the Nusselt and Reynolds number for wire mesh lattices with 80 % porosity and IL pattern. The average deviation varies from a minimum value of 2.9 % for case #3 to a maximum value of 5.8 % for case #1.

In addition, comparing the GC (Table 7) and the ICs (Table 6), the GC fits cases #1 and #2 better with GC MeAEs of 5.8 and 3.5 % respectively compared to ICs MeAEs of 9.3 and 4.4 %. The situation is reversed for case #3 where the IC fits better with a MeAE of 0.5 % compared to the GC's deviation of 2.9 %.

As a result, the GC is more stable, as the differences in its MeAEs across the three cases are smaller than those of the ICs.

#### 4.2. Pressure drop results

$\Delta P/L$  is an important parameter for the hydrodynamic analysis of volumetric absorbers. A high  $\Delta P/L$  translates to significant energy losses due to increased pumping power, impacting on the operating costs. In order to calculate the  $\Delta P/L$  in a volumetric absorber, a parametric study was carried out for six air inlet velocities between 0.5 and 5 m/s, keeping the fluid temperature constant at 300 K. Both conditions are typical for pressure drop analyses in volumetric absorbers [45,52,62,63]. In real working conditions, the prediction of the pressure drop at temperatures higher than 300 K, the density and the dynamic viscosity of the fluid must be inferred.

##### 4.2.1. Fluid flow performance

This section presents the CFD results for the  $\Delta P/L$  in the three study cases (0.7, 0.4 and 0.1 mm wire diameters) with a methodology validated in the literature [45,53]. Fig. 9 depicting the  $\Delta P/L$  of the absorber for the three cases studied shows that the  $\Delta P/L$  increases as velocity increases and wire diameter decreases. For case #1 the 1, 3 and 5 m/s air

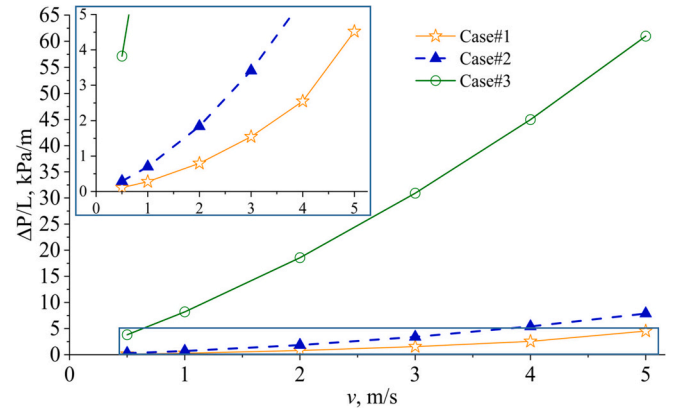


Fig. 9.  $\Delta P/L$  of the absorber for case #1, case #2 and case #3 evaluated in six inlet velocities between 0.5 and 5 m/s.

inlet velocities are associated with 0.3, 1.5, and 4.5 kPa/m  $\Delta P/L$ . In addition, case #3 shows relationship of an air inlet velocity of 1 m/s with a  $\Delta P/L$  of 8.2 kPa/m, of 3 m/s with 30.9 kPa/m, and of 5 m/s with 61 kPa/m.

##### 4.2.2. Individual Correlations (ICs)

Using the parametric analysis of each IL woven wire mesh geometry studied and presented in Table 1, ICs were obtained. Darcy's law with the Forchheimer extension was used to obtain the IC for each case [29,45,52] based on Eq. (15):

$$-\frac{dp}{dx} = -\frac{\Delta P}{L} = \frac{\mu}{K_1} v + \frac{\rho}{K_2} v^2 \quad (15)$$

where  $v$  is the air inlet velocity,  $\mu$  and  $\rho$  are the dynamic viscosity and density of the fluid, respectively; and  $K_1$  and  $K_2$  are the viscosity and inertial coefficients, respectively.

The three ICs following Darcy's law with the Forchheimer extension calculated using Eq. (15) were fitted to the CFD data. Table 8 presents the viscosity and inertial permeability coefficients for each IC, along with the corresponding MeAE for each case.

Furthermore, it is important to point out the difference in the viscosity coefficient ( $K_1$ ) between cases #1 and #3, a difference of 4 orders of magnitude, whereas the inertial coefficient ( $K_2$ ) is around 5 times lower. These differences in the viscosity and inertial permeability coefficients show the effect of the different geometries. In addition, cases #2 and #3 show a good fit as it is indicated by MeAE values of 0.5 and 0.8 %, respectively. However, the IC of case #1 deviates more from the CFD data. For a more complete picture of the differences, Table 9 compares the CFD results to those inferred from the IC at three different velocities. Cases #2 and #3 have deviations lower than 3 % for the

Table 8

Viscosity and inertial permeability coefficients,  $K_1$  and  $K_2$ , and mean absolute error for each IC of the  $\Delta P/L$ .

| Case | d, mm | $K_1$ , m <sup>2</sup> | $K_2$ , m | MeAE, % |
|------|-------|------------------------|-----------|---------|
| #1   | 0.7   | 2.50E-05               | 6.72E-03  | 20.5    |
| #2   | 0.4   | 3.85E-08               | 5.38E-03  | 0.5     |
| #3   | 0.1   | 2.51E-09               | 1.21E-03  | 0.8     |

Table 7

Dimensionless coefficients, mean absolute error and range of validity of the HTC-GC at thermal equilibrium.

| Case | $a_v$ , m <sup>-1</sup> | $d_h$ , mm | " $m_1$ " | " $m_2$ " | " $m_3$ " | "F"  | MeAE, % | Re range |
|------|-------------------------|------------|-----------|-----------|-----------|------|---------|----------|
| #1   | 1,138                   | 2.81       |           |           |           |      | 5.8     | 6–222    |
| #2   | 1,998                   | 1.60       | 0.00875   | 1.352     | 2.051     | 3.19 | 3.5     | 4–133    |
| #3   | 8,081                   | 0.39       |           |           |           |      | 2.9     | 1–31     |

**Table 9**Comparison between the numerical CFD  $\Delta P/L$  and the IC values for each case and 0.5, 2 and 5 m/s inlet velocity.

| v<br>m/s | Case #1 |         |      | Case #2 |         |     | Case #3  |         |     |
|----------|---------|---------|------|---------|---------|-----|----------|---------|-----|
|          | CFD     | IC      | D    | CFD     | IC      | D   | CFD      | IC      | D   |
|          | Pa/m    | Pa/m    | %    | Pa/m    | Pa/m    | %   | Pa/m     | Pa/m    | %   |
| 0.5      | 108.4   | 44.1    | 59.3 | 289.4   | 294.6   | 1.8 | 3,823.8  | 3,919.9 | 2.5 |
| 2        | 795.8   | 701.8   | 11.8 | 1,841.7 | 1,834.7 | 0.4 | 18,558.2 | 18,591  | 0.2 |
| 5        | 4,520.9 | 4,380.7 | 3.1  | 7,877.9 | 7,867.8 | 0.1 | 60,968.5 | 61,035  | 0.1 |

velocities presented, which confirms the accuracy of the IC in predicting the  $\Delta P/L$ . However, the IC of case #1 does not fit the CFD results as well as the ICs of the remaining cases at low velocities, reaching a deviation of almost 60 % at 0.5 m/s. In addition, the lower the inlet velocity, the larger the deviation in all cases. In [section 4.2.3](#) the explanation is addressed.

#### 4.2.3. Generalised Correlation (GC)

This section provides a GC that promises to be more useful in optimisation studies than the ICs due to its easier implementation and saving both computational time and storage when a range of geometrical properties is evaluated. This GC, derived from the current study of 80 % porosity wire mesh stacks with an IL arrangement, modifies Ergun's equation to predict the  $\Delta P/L$  similarly to that in the literature [52,64]. The expression of the GC for a wire mesh geometry is presented in Eq. (16), which differs from the IC based on Darcy's Law with Forchheimer extension, Eq.(15).

$$-\frac{dp}{dx} = -\frac{\Delta P}{L} = \alpha f_1 \cdot M^2 \cdot \mu \cdot v + \beta f_2 \cdot M \cdot \rho \cdot v^2 \quad (16)$$

Here  $f_1$  and  $f_2$  are calculated and  $\alpha$  and  $\beta$  fitted through the MATLAB fitting tool in accordance with the literature [45]. This GC expression takes into account the geometrical properties of the wire meshes with the mesh count parameter,  $M$ . The GC for 80 % porosity with an IL pattern is defined by Eq. (17).

$$-\frac{dp}{dx} = -\frac{\Delta P}{L} = 67.95 \cdot M^2 \cdot \mu \cdot v + 0.3274 \cdot M \cdot \rho \cdot v^2 \quad (17)$$

[Table 10](#) presents the MeAE between the CFD and the GC values. The highest MeAE is equal to 7 % for the largest wire diameter, case #1. Comparing the deviations of the GC with the IC ([Table 8](#)), the ICs fit better to the CFD values in cases #2 and #3 with MeAEs under 1 % whereas for the GC, both cases present MeAEs under 3 %, indicating acceptable fit. On the contrary, in case #3, the GC shows better fitting to the CFD values than the IC with MeAEs values of 6.2 and 20.5 % respectively.

In addition, [Fig. 10](#) illustrates the comparison of the  $\Delta P/L$  between the numerical CFD, and the IC and the GC values for case #1. In general, at low inlet velocities the  $\Delta P/L$ -GC fits to the  $\Delta P/L$ -CFD better than the  $\Delta P/L$ -ICs whereas at higher inlet velocities the  $\Delta P/L$ -IC has smaller deviations than the  $\Delta P/L$ -GC.

#### 4.3. Numerical validation

The results obtained in the previous sections were validated using a porous model or homogeneous equivalent model (HEM). The porous model is a macro-scale numerical simulation method which applies the

**Table 10**Mean absolute errors between CFD and the GC values for each case of study in the  $\Delta P/L$ .

| Case | d, mm | M, mm <sup>-1</sup> | MeAE, % |
|------|-------|---------------------|---------|
| #1   | 0.7   | 0.35                | 6.2     |
| #2   | 0.4   | 0.61                | 2.4     |
| #3   | 0.1   | 2.44                | 1.3     |

physics from representative 3D detailed simulations to real volumetric absorbers adopting the assumptions and boundaries presented in this study. This section follows the same organisation as before, presenting the validation of the HTC results first, and then the  $\Delta P/L$  results.

#### 4.3.1. Validation of the HTC

**4.3.1.1. HEM geometry of the HTC and governing equations.** The porous model geometry and boundary conditions used are presented in [Fig. 11](#). The 2D geometry of the porous model follows the same structure as the 3D detail simulations presented in [Fig. 4](#). The different absorber lengths are presented in [Table 3](#) and the inlet dimensions have the same size as the lengths in each case. The air chamber at the front is defined by the length equivalent to seven wire diameters, this air chamber is needed because when the air gets close to the wires, it is heated by convection mechanism. In the HEM simulations, the inclusion of an air volume at the rear is unnecessary because the effective properties introduced in the porous model are based on the assumption that thermal equilibrium has been achieved along the HEM geometry.

Eq. (18) to eq. (20) define the governing equations used in the HTC HEM model, providing validation for the results obtained in the micro-scale simulations.

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

$$\nabla \cdot (\rho \cdot \vec{v}) = 0 \quad (18)$$

Momentum equation for the porous region in steady state conditions:

$$\frac{1}{\phi^2} \nabla \cdot (\rho \cdot v \cdot v) = -\nabla \langle P_f \rangle^f + \frac{\mu}{\phi} \nabla^2 v - \left[ \frac{\mu}{K_1} - \frac{\rho}{K_2} [v] \right] \cdot v \quad (19)$$

Energy equation for the fluid region in steady state conditions:

$$\nabla \cdot (\rho \cdot c_{pf} \cdot T_f \cdot v) = \nabla \cdot (k_{eff} \cdot \nabla T_f) + h_v \cdot (\langle T_s \rangle^f - \langle T_f \rangle^f) \quad (20)$$

The energy equation that governs the porous region is not included because the temperature of the wires ( $\langle T_s \rangle^f$ ) is constant and uniform along the volumetric absorber (700, 1100, 1500 K). As a result, radiation and conduction effects are not considered.

#### 4.3.1.2. Assumptions and boundary conditions for the HEM simulations.

The HEM simulations used to validate the HTC are based on the following assumptions and boundary conditions:

- A velocity inlet boundary is applied at the inlet surface, with air inlet velocities ranging from 0.5 to 5 m/s, as analysed in [section 3.1.2](#). A laminar flow regime is used, as demonstrated in [Appendix 2](#).
- The lateral surface is characterised by adiabatic walls, assuming the HEM represents the real dimensions of the volumetric absorber.
- A pressure outlet boundary is imposed to the outlet surface, with the pressure set to atmospheric.
- All the wires were defined with a constant and uniform temperature ( $\langle T_s \rangle^f = 700, 1100, 1500$  K). This replicates the working conditions of [section 3.1](#), isolating the effect of the convective heat transfer coefficient from other heat transfer mechanisms. As a result, radiation and conduction effects are not considered.

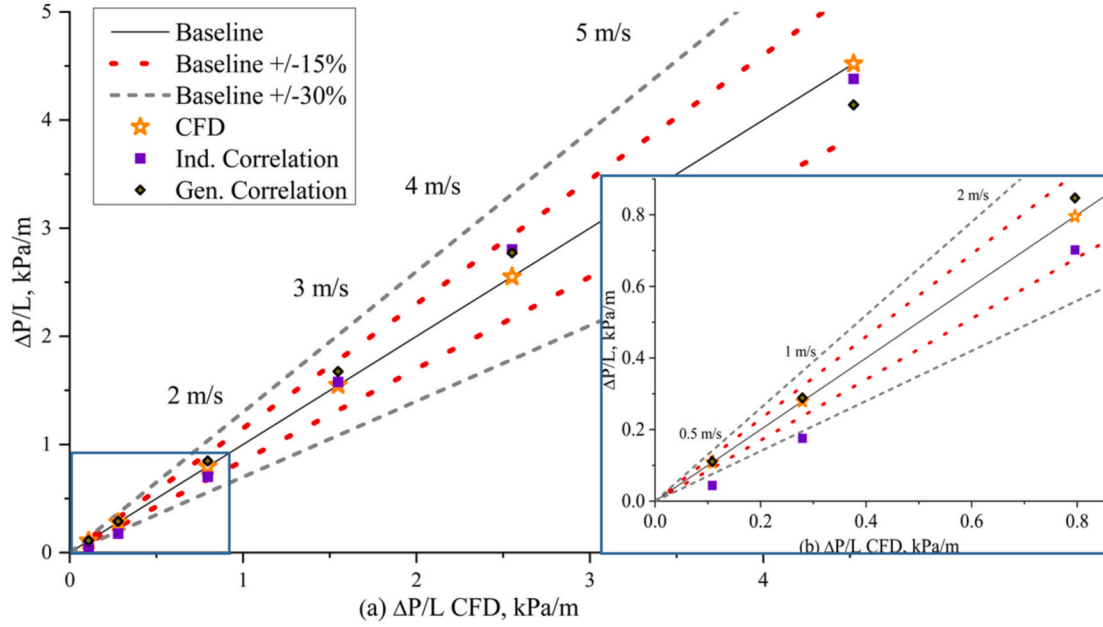


**Fig. 10.**  $\Delta P/L$  comparison for case #1 between the  $\Delta P/L$ -CFD values, the  $\Delta P/L$ -IC and the  $\Delta P/L$ -GC: (a) normal view of the results and (b) zoomed view of the results.

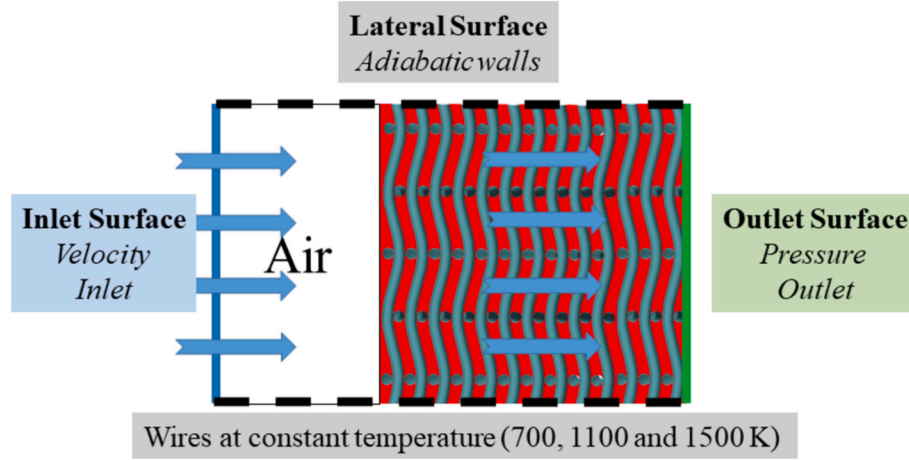


**Fig. 11.** Porous model geometry and boundary conditions for the validation of the HTC results.

- The air is modelled as an ideal gas, with properties that vary depending on the temperature.

The same sensitive study carried out in [section 3.1](#) was adopted for the porous model, based on six air inlet velocities between 0.5 and 5 m/s and the solid temperature at 700, 1100 and 1500 K.

The effective properties of the volumetric absorber in this HEM model are summarised in [Table 11](#). The convective HTC applied in the HEM is different depending on which HTC correlation is used:

- Eq.(21) is implemented when ICs are used to obtain the HTC, with the corresponding a, b and c coefficients provided in [Table 6](#).

- Eq. (22) is implemented when GC is used to obtain the HTC, with the corresponding  $m_1$ ,  $m_2$  and  $m_3$  coefficients provided in [Table 7](#).

The coefficients used in the correlations are obtained when the fluid phase has reached thermal equilibrium.

These effective properties are inputs for the porous model as they determine how the volumetric absorber behaves under the working conditions established in the micro-scale simulations. The porous model simulations obtained the convective HTC along the volumetric absorber by setting a fixed and uniform temperature for the solid matrix while the fluid phase heats up until thermal equilibrium is achieved. This numerical validation methodology was already demonstrated in the literature [\[43\]](#).

**Table 11**

Effective properties of the volumetric absorber for the validation of the HTC.

|                                       |      |
|---------------------------------------|------|
| $Nu = a \cdot Re^b \cdot Pr^c$        | (21) |
| $Nu = (m_1) \cdot Re^{m_2} + P^{m_3}$ | (22) |

**4.3.1.3. HTC Results of the HEM simulations.** [Table 12](#) shows the results of the HTC at thermal equilibrium obtained in previous sections with the IC ([section 4.1.2](#)) and GC ([section 4.1.3](#)) compared to the results obtained when these correlations ([Table 11](#)) are implemented in the porous model or HEM for case #2 at a wire temperature of 1100 K.

**Table 12**

Validation of the TEZ HTC results for the CFD and porous model (HEM) with the IC and GC at a solid temperature of 1100 K for case #2.

| $v$<br>m/s | TEZ HTC-IC                     |                                |        | TEZ HTC-GC                     |                                |        |
|------------|--------------------------------|--------------------------------|--------|--------------------------------|--------------------------------|--------|
|            | CFD-IC<br>kW/m <sup>3</sup> ·K | HEM-IC<br>kW/m <sup>3</sup> ·K | D<br>% | CFD-GC<br>kW/m <sup>3</sup> ·K | HEM-GC<br>kW/m <sup>3</sup> ·K | D<br>% |
| 0.5        | 299                            | 299                            | 0.1    | 310                            | 312                            | 0.8    |
| 1          | 320                            | 321                            | 0.2    | 314                            | 316                            | 0.9    |
| 2          | 343                            | 345                            | 0.6    | 323                            | 327                            | 1.2    |
| 3          | 356                            | 359                            | 0.7    | 335                            | 339                            | 1.2    |
| 4          | 364                            | 368                            | 1.1    | 347                            | 353                            | 1.5    |
| 5          | 371                            | 373                            | 0.5    | 363                            | 367                            | 1.1    |

Additional results can be found in [Appendix 4](#).

The deviations are lower than 1.5 % for all the velocities and correlations analysed. The use of IC presents lower deviations than the use of GC in the HEM with respect to the CFD results. However, the results with both the IC and GC correlations, are acceptable, indicating that at thermal equilibrium when used in HEM to analyse the thermal performance of OVRs in CRS are valid.

#### 4.3.2. Validation of the $\Delta P/L$

**4.3.2.1. HEM geometry of the  $\Delta P/L$  and governing equations.** The validation of the  $\Delta P/L$  results was carried out with the porous model under the 2D geometry and boundary conditions presented in [Fig. 12](#). The geometries used were similar to those studied in the 3D detail simulations with the same lengths in each absorber ([Table 4](#)) and with inlet dimensions of the same size as the length. No air chamber at the frontal or rear is needed because in this HEM simulations is obtained the  $\Delta P/L$  maintaining the wires at a temperature of 300 K.

Eq. (23) and (24) define the governing equations used in the  $\Delta P/L$  HEM model, providing validation for the results obtained in the micro-scale simulations. The fluid temperature is held constant at 300 K, so energy equations are not applied.

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

$$\nabla \cdot (\rho \cdot \vec{v}) = 0 \quad (23)$$

Momentum equation for the porous region in steady state conditions:

$$\frac{1}{\phi} \nabla \cdot (\rho \cdot \vec{v} \cdot \vec{v}) = -\nabla \langle P_f \rangle^f + \frac{\mu}{\phi} \nabla^2 \vec{v} - \left[ \frac{\mu}{K_1} - \frac{\rho}{K_2} |\vec{v}| \right] \cdot \vec{v} \quad (24)$$

**4.3.2.2. Assumptions and boundary conditions for the HEM simulations.** The HEM simulations used to validate the  $\Delta P/L$  are based on the following assumptions and boundary conditions:

- A velocity inlet boundary is applied at the inlet surface, with air inlet velocities ranging from 0.5 to 5 m/s, as analysed in [section 3.2.2](#). A laminar flow regime is used, as demonstrated in [Appendix 2](#).
- The lateral surface is characterised by adiabatic walls, assuming the HEM represents the real dimensions of the volumetric absorber.
- A pressure outlet boundary is imposed to the outlet surface, with the pressure set to atmospheric.
- An adiabatic wall is set as a boundary condition in the wires as a result of the absorber being set at room temperature (300K). This replicates the working conditions of [section 3.2](#), isolating the pressure drop from the temperature. When the correlations are incorporated into a porous model under real operating conditions, the air properties ( $\rho$  and  $\mu$ ) will change with temperature, resulting in fluctuations in the pressure drop with the temperature.
- The air is modelled as an ideal gas with a constant density, assuming properties corresponding to a temperature of 300 K.

A parametric study was carried out using the porous model, six air inlet velocities between 0.5 and 5 m/s and keeping the wires temperature at 300 K as in the 3D detailed numerical  $\Delta P/L$  study.

The effective properties of the volumetric absorber in this HEM model are summarised in [Table 13](#). The pressure drop applied in the porous model is different depending on which  $\Delta P$  correlation is used.

- Eq. (25) is implemented when ICs are used to obtain the  $\Delta P$ , with the  $K_1$  and  $K_2$  provided in [Table 8](#).
- Eq. (26) is implemented when GC is used to obtain the  $\Delta P$ .

These effective properties are inputs for the porous model as they determine how the volumetric absorber behaves under the working conditions established in the micro-scale simulations. The porous model simulations obtained the pressure drop along the volumetric absorber, enabling the determination of the pressure drop per unit length ( $\Delta P/L$ ) of the absorber. This numerical validation methodology was already demonstrated in the literature [\[43\]](#).

**4.3.2.3.  $\Delta P/L$  Results of the HEM simulations.** [Table 14](#) shows the results of the  $\Delta P/L$  obtained in previous sections with IC ([section 4.2.2](#)) and GC ([section 4.2.3](#)) compared to the results obtained when these correlations are implemented in the porous model or HEM for case #2 and three air inlet velocities, 0.5, 2 and 5 m/s. Further results can be found in [Appendix 5](#).

The deviations of the  $\Delta P/L$  are lower than 1.0 % for the velocities and correlations analysed. The use of IC was equivalent to that of the GC in the HEM, both correlations presenting acceptable deviations with respect to the CFD results. Consequently, the validation of the  $\Delta P/L$  when used in HEM to analyse the fluid flow of OVRs in CRS.

## 5. Conclusions

OVRs are a promising technology in CRS. The design of an OVR is dependent on two main thermo-fluid-dynamic attributes, the convective HTC and the pressure drop. This work presents the convective HTC and  $\Delta P/L$  for IL woven wire mesh, with an 80 % constant porosity and a range of wire diameters (case #1–0.7 mm, case #2–0.4 mm and case #3–0.1 mm). For all the cases, ICs and GCs were fitted to the CFD results, as they are used in OVR optimisation studies.

**Table 13**

Effective properties of the volumetric absorber for the validation of the  $\Delta P/L$ .

$$-\frac{dp}{dx} = -\frac{\Delta P}{L} = \frac{\mu}{K_1} v + \frac{\rho}{K_2} v^2 \quad (25)$$

$$-\frac{\Delta P}{L} = 67.95 \cdot M^2 \cdot \mu \cdot v + 0.3274 \cdot M \cdot \rho \cdot v^2 \quad (26)$$

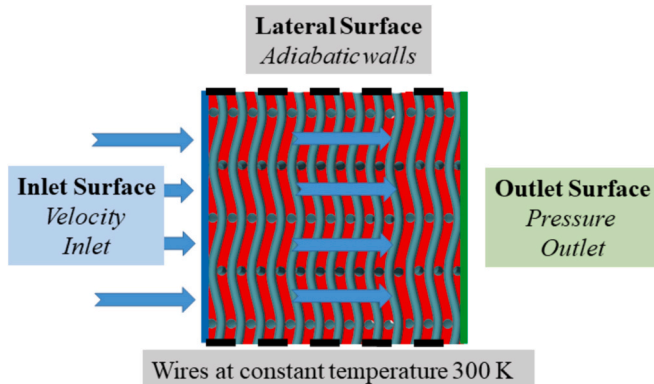


**Fig. 12.** Porous model geometry and boundary conditions for the validation of the  $\Delta P/L$  results.

**Table 14**

Validation of the  $\Delta P/L$  results for the CFD and porous model with the IC and the GC case #2.

| $v$<br>m/s | $\Delta P/L$ -IC |                |        | $\Delta P/L$ -GC |                |        |
|------------|------------------|----------------|--------|------------------|----------------|--------|
|            | CFD-IC<br>Pa/m   | HEM-IC<br>Pa/m | D<br>% | CFD-GC<br>Pa/m   | HEM-GC<br>Pa/m | D<br>% |
| 0.5        | 294.6            | 296.6          | 0.7    | 292.3            | 294.3          | 0.7    |
| 2          | 1,834.7          | 1,846.2        | 0.6    | 1,874.4          | 1,886.1        | 0.6    |
| 5          | 7,867.8          | 7,907.7        | 0.5    | 8,211.3          | 8,251.9        | 0.5    |

Concerning the HTC at thermal equilibrium, the GC fits the CFD results better than the ICs in cases #1 and #2. However, case #3 is better predicted by the IC. The GC obtained in this work for the HTC at thermal equilibrium presents acceptable MeAE, in two out of the three cases with better accuracy than the ICs.

For the pressure drop analysis results offer similar conclusions to those of the HTC. While some ICs were found to fit the CFD better, such as the finer wires in cases #2 and #3, in general the GC presented better overall performance. Generally, at low velocities (between 0.5 and 2 m/s), the GC presents smaller deviations than those of the ICs. However, at high velocities, the ICs fit the CFD results better.

In addition, the HTC at thermal equilibrium and the  $\Delta P/L$  results obtained with the ICs and GCs were validated through a HEM with deviations of less than 3 %. All the methodologies used in this work have been validated in the literature, which reinforces the validity of the presented correlations (ICs and GCs).

All results considered, this study concludes that the geometrical parameters that define a wire mesh lattice need to be included in the prediction of both thermo-fluid-dynamic attributes, HTC and  $\Delta P/L$ .

Furthermore, the GCs calculated and validated for the HTC at thermal equilibrium and the  $\Delta P/L$  in the study indicate that they could be used in macro-scale simulations of wire mesh OVRs. A balance between both attributes is important for optimising the overall performance of metallic open volumetric receivers. Later optimisation studies with a porous model or HEM under LTNE state could use the GCs to predict the convective HTC at thermal equilibrium and the pressure drop in real IL wire mesh absorbers with wire diameters ranging from 0.1 to 0.7 mm and 80 % volumetric porosity.

## Nomenclature

|             |                                                       |
|-------------|-------------------------------------------------------|
| $M$         | Mesh count, $\text{mm}^{-1}$                          |
| $d$         | Wire diameter, mm                                     |
| $w$         | Wire spacing, mm                                      |
| $c_f$       | Compactness factor                                    |
| $\Phi$      | Porosity, %                                           |
| $\Phi_{SM}$ | Single mesh porosity, %                               |
| $\Phi_{IL}$ | Inline mesh porosity, %                               |
| $C$         | Unit cell mesh constant                               |
| $a_v$       | Specific surface area, $\text{m}^{-1}$                |
| $d_h$       | Hydraulic diameter, mm                                |
| $\rho$      | Density, $\text{kg m}^{-3}$                           |
| $k$         | Thermal conductivity, $\text{W m}^{-1} \text{K}^{-1}$ |
| $c_p$       | Specific heat, $\text{J kg}^{-1} \text{K}^{-1}$       |
| $v$         | Superficial velocity at the inlet, m/s                |
| $\mu$       | Dynamic viscosity, $\text{kg m}^{-1} \text{s}^{-1}$   |
| $K_1$       | Viscosity coefficient, $\text{m}^2$                   |
| $K_2$       | Inertial coefficient, m                               |
| $T$         | Temperature, K                                        |
| $T_s$       | Temperature of the solid, K                           |
| $T_f$       | Temperature of the fluid, K                           |

|              |                                                                                  |
|--------------|----------------------------------------------------------------------------------|
| $h_v$        | Volumetric convective heat transfer coefficient, $\text{W m}^{-3} \text{K}^{-1}$ |
| $q_s$        | Solid heat flux, W                                                               |
| $Q$          | Heat flux, W                                                                     |
| $T_{ls}$     | Local temperature of the solid, K                                                |
| $T_{mlf}$    | Fluid mass average temperature, K                                                |
| $f_1$        | Function 1, $\text{kg m}^{-1} \text{s}^{-1}$                                     |
| $f_2$        | Function 2, $\text{kg m}^{-3}$                                                   |
| $Re$         | Reynold number                                                                   |
| $Pr$         | Prandlt number                                                                   |
| $Nu$         | Nusselt number                                                                   |
| $F$          | Geometrical factor                                                               |
| $\Delta P/L$ | Differential pressure per unit length, Pa/m                                      |
| $D$          | Deviation, %                                                                     |
| $MeAE$       | Mean absolute error, %                                                           |
| $CFD$        | Computational Fluid Dynamics                                                     |
| $CRS$        | Central Receiver System                                                          |
| $CSP$        | Concentrated Solar Power                                                         |
| $OVRs$       | Open volumetric receivers                                                        |
| $PPI$        | Pore per inch                                                                    |
| $LTNE$       | Local thermal non equilibrium                                                    |
| $HTC$        | Heat transfer coefficient                                                        |
| $FIZ$        | Fluid inlet zone                                                                 |
| $FHZ$        | Fluid heating zone                                                               |
| $TEZ$        | Thermal equilibrium zone                                                         |
| $IL$         | Inline                                                                           |
| $ST$         | Staggered                                                                        |
| $IC$         | Individual Correlation                                                           |
| $GC$         | Generalised Correlation                                                          |

## 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. **Meryem Farchado:** Writing – review & editing, 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, Conceptualization.

## 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.

## Data availability

Data will be made available on request.

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## Appendix 1. Grid analysis for the convective HTC

Case #1 presented a convective HTC at thermal equilibrium of  $101 \text{ kW/m}^3\cdot\text{K}$  for an inlet velocity of 3 m/s with a temperature of the wires of 700 K. The grid used had a base size of 0.245 mm and 42,895,029 cells. When the base size was reduced to 0.225 mm the number of cells was increased to 47,832,540 and the convective HTC was  $100 \text{ kW/m}^3\cdot\text{K}$  under the same working conditions (3 m/s and 700 K). These results showed a deviation of 1.2 %.

The base size selected in case #3 was 0.125 mm with 5,493,819 cells. The convective HTC at thermal equilibrium was  $3.59 \text{ MW/m}^3\cdot\text{K}$  for an inlet velocity of 3 m/s with a wire temperature of 700 K. The convective HTC with the 0.125 mm base size showed a deviation of 3 % with respect to the convective HTC with a value of  $3.48 \text{ MW/m}^3\cdot\text{K}$ . The last convective HTC was obtained with a base size of 0.06 mm with 10,777,901 cells and the same working conditions (3 m/s and 700 K).

## Appendix 2. Analysis of the flow regime

**Table A.1**

Comparison between the turbulence and laminar flow solvers for the studied cases at an air inlet velocity of 1 m/s and a wire temperature of 700 K.

| Case      | TEZ HTC with Laminar flow<br>kW/m <sup>3</sup> ·K | TEZ HTC with Turbulent flow (SST-Komega)<br>kW/m <sup>3</sup> ·K | Deviation<br>% |
|-----------|---------------------------------------------------|------------------------------------------------------------------|----------------|
| #1–0.7 mm | 76.6                                              | 76.3                                                             | 0.4            |
| #2–0.4 mm | 228.8                                             | 228.8                                                            | 0.005          |
| #3–0.1 mm | 3,586.9                                           | 3,587.4                                                          | 0.015          |

## Appendix 3. Grid analysis for the $\Delta P/L$

Table A.2 showed the influence of the grid analysis in the  $\Delta P/L$  for case #2 for an inlet velocity of 3 m/s. The deviation in all the cases was lower than 3 % resulting in acceptable grids for the study. The same procedure was followed for the rest of the cases.

Case #1 presented a  $\Delta P/L$  of 1,547 Pa/m for an inlet velocity of 3 m/s. The grid used had a base size of 0.245 mm and 35,557,723 cells. When the base size was reduced to 0.225 mm the number of cells rose to 39,392,795 and the  $\Delta P/L$  was 1,547 Pa/m under an inlet velocity of 3 m/s. These results showed a deviation of 1.7 %.

The base size chosen in case #3 was 0.125 mm with 8,714,384 cells. The  $\Delta P/L$  was 30,914 Pa/m for an inlet velocity of 3 m/s. The  $\Delta P/L$  obtained with the 0.125 mm base size presents a 0.9 % deviation with respect to the  $\Delta P/L$ , with a value of 31,191 Pa/m. The last  $\Delta P/L$  value was based on a grid size of 0.1 mm, with 12,145,765 cells and an inlet velocity of 3 m/s.

**Table A.2**

Influence of the grid analysis in the  $\Delta P/L$  for case #2 and an inlet velocity of 3 m/s.

| Base Size<br>mm | Number of cells | $\Delta P/L$<br>Pa/m | D<br>% |
|-----------------|-----------------|----------------------|--------|
| 1               | 1,840,171       | 3,587                | 7.6    |
| 0.5             | 7,555,151       | 3,472                | 4.1    |
| 0.3             | 17,405,135      | 3,412                | 2.3    |
| 0.2             | 27,143,878      | 3,334                | –      |

## Appendix 4. Additional results of the validation of the HTC

**Table A.3**

Validation of the TEZ HTC results for the CFD and porous model with the IC and GC at a wire temperature of 700 K for case #2.

| v<br>m/s | TEZ HTC-IC                     |                                |        | TEZ HTC-GC                     |                                |        |
|----------|--------------------------------|--------------------------------|--------|--------------------------------|--------------------------------|--------|
|          | CFD-IC<br>kW/m <sup>3</sup> ·K | HEM-IC<br>kW/m <sup>3</sup> ·K | D<br>% | CFD-GC<br>kW/m <sup>3</sup> ·K | HEM-GC<br>kW/m <sup>3</sup> ·K | D<br>% |
| 0.5      | 228                            | 231                            | 1.4    | 225                            | 226                            | 0.7    |
| 1        | 244                            | 248                            | 1.6    | 232                            | 234                            | 0.9    |
| 2        | 261                            | 266                            | 1.8    | 252                            | 255                            | 1.0    |
| 3        | 269                            | 277                            | 2.6    | 276                            | 279                            | 1.2    |
| 4        | 275                            | 283                            | 3.0    | 305                            | 307                            | 0.8    |
| 5        | 279                            | 286                            | 2.4    | 338                            | 338                            | 0.2    |

**Table A.4**  
Validation of the TEZ HTC results for the CFD and porous model with the IC and GC at a wire temperature of 1500 K for case #2.

| $v$<br>m/s | TEZ HTC-IC                     |                                |        | TEZ HTC-GC                     |                                |        |
|------------|--------------------------------|--------------------------------|--------|--------------------------------|--------------------------------|--------|
|            | CFD-IC<br>kW/m <sup>3</sup> ·K | HEM-IC<br>kW/m <sup>3</sup> ·K | D<br>% | CFD-GC<br>kW/m <sup>3</sup> ·K | HEM-GC<br>kW/m <sup>3</sup> ·K | D<br>% |
| 0.5        | 366                            | 359                            | 2.0    | 398                            | 402                            | 0.9    |
| 1          | 393                            | 386                            | 1.9    | 401                            | 405                            | 1.0    |
| 2          | 420                            | 414                            | 1.4    | 405                            | 411                            | 1.5    |
| 3          | 437                            | 432                            | 1.2    | 412                            | 419                            | 1.8    |
| 4          | 447                            | 443                            | 0.9    | 419                            | 427                            | 2.0    |
| 5          | 453                            | 450                            | 0.7    | 426                            | 435                            | 2.1    |

Appendix 5. Additional results of the validation of the ΔP/L

**Table A.5**  
Validation of the ΔP/L results for the CFD and porous model with the IC and the GC for case #1.

| $a$ | ΔP/L-IC        |                |        | ΔP/L-GC        |                |        |
|-----|----------------|----------------|--------|----------------|----------------|--------|
|     | CFD-IC<br>Pa/m | HEM-IC<br>Pa/m | D<br>% | CFD-GC<br>Pa/m | HEM-GC<br>Pa/m | D<br>% |
| 0.5 | 44.14          | 45.27          | 2.6    | 111.30         | 111.93         | 0.6    |
| 2   | 701.80         | 711.01         | 1.3    | 858.08         | 855.80         | 0.3    |
| 5   | 4,380.7        | 4,417.2        | 0.8    | 4,209.5        | 4,173.5        | 0.9    |

**Table A.6**  
Validation of the ΔP/L results for the CFD and porous model with the IC and the GC for case #3.

| $a$ | ΔP/L-IC        |                |        | ΔP/L-GC        |                |        |
|-----|----------------|----------------|--------|----------------|----------------|--------|
|     | CFD-IC<br>Pa/m | HEM-IC<br>Pa/m | D<br>% | CFD-GC<br>Pa/m | HEM-GC<br>Pa/m | D<br>% |
| 0.5 | 3,919.9        | 3,928.2        | 0.2    | 3,972.3        | 3,976.8        | 0.1    |
| 2   | 18,591         | 18,633         | 0.2    | 18,710         | 18,730         | 0.1    |
| 5   | 61,035         | 61,175         | 0.2    | 60,875         | 60,930         | 0.1    |

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