Optimized volumetric solar receiver: Thermal performance prediction and experimental validation

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ABSTRACT

In the last decade, different absorber geometries, such as foams and honeycombs, have been tested at laboratory or industrial scale in order to achieve high performance in the conversion of the solar radiation into usable heat, with the current state-of-the-art, the HiTRec-II monolithic honeycomb, characterized by a square-channel section and made out of siliconized silicon carbide (SiSiC). Such geometry has been so far the best compromise for large-scale application thanks to the low production costs, easy manufacturability through extrusion procedure and overall acceptable performance. However, it does present some drawbacks, since the geometry is not able to contain the radiative heat losses, especially from the front surface. An optimized absorber geometry, capable to reduce overall thermal losses, is presented in this work, being able to increase the final thermal efficiency of more than 12% compared to the current state-of-the-art and showing the presence of the so-called volumetric effect, since the outlet fluid temperature is higher than the solid inlet temperature. A test sample has been produced for laboratory-scale experiments, in the form of a 3:1 scaled prototype through additive manufacturing procedure, using a titanium-aluminium alloy (Ti6Al4V) and the experimental results were in good agreement with the numerical calculation, with a deviation of 3%, computed considering a 3:1 Ti6Al4V scaled-up sample. As the manufacturing technology will progress and become cheaper in the near future, it will be possible to improve the overall Solar Power Tower (SPT) plants performance using advanced micro-geometry for open volumetric receivers.

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1. Introduction

Open volumetric receivers represent the core of a particular type of solar power tower plants (SPT) where the reflected sunlight, coming from the heliostat field, is converted into thermal energy for further industrial uses. The volumetric receiver is characterized by a porous solid (absorber) crossed by atmospheric air that is heated by contact with the inner walls. In such technology, the volume of the receiver mainly has a thermal function. While with linear receivers the wall temperature is always higher than the fluid temperature, the use of volumetric receivers could allow the presence of the so-called volumetric effect. In this case, the outlet temperature of the air can reach values of temperature higher than the ones of the solid front surface.

Several types of volumetric absorbers have been built and tested in the past decades, such as wire mesh, fibres, packed bed particles, honeycombs and foams, using both metals and ceramics as manufacturing materials. The entire variety of different structures can be characterized by parameters and coefficients, defined as effective properties, like the porosity ($\varepsilon$), the extinction coefficient ($\xi$) and the volumetric heat transfer coefficient ($hA_{\text{g}}$). The first is the ratio between the void volume and the total volume of the porous absorber, while $\xi$ defines the attenuation of the incoming sunlight in the absorber inner volume. The volumetric heat transfer coefficient is the product between the convective heat transfer coefficient ($h$), defined as the ratio of the heat flux and the thermodynamic driving force (that in this case is the temperature difference between the atmospheric flow and the inner surfaces of the absorber), and the specific surface area ($A_{\text{s}}$), which is the index of the exchange surface in porous materials and is defined as the ratio of the wet surface and
Volumetric receivers showed promising results in previous scientific works; however, they also showed a wide margin of improvement, since the volumetric effect has never been achieved.

The shape of the solar absorber plays an important role in the utilization of the incoming sunlight, particularly at the inlet section where the radiation hits the volume. For this reason, the optimization of the porous structure to the different thermodynamic needs is a key aspect to improve the efficiency of the energy conversion.

The first example reported of a ceramic volumetric absorber is the open cavity receiver developed by Sander Associates Inc. [1], in which the prototype, able to produce 250 kWth, was characterized by silicon carbide honeycomb structure and designed for 1100 °C air outlet temperature. A new concept of volumetric absorber, consisting of a hexagonal absorber structure and a siliconized silicon carbide (SiSiC) cup placed on a stainless steel supporting structure. The experiments showed good agreement with the calculated performance with an outlet air temperature of 800 °C. No hotspots were observed and eventually the receiver showed reduced start-up times and easy operability and maintenance. However, during the test the stainless steel structure was deformed, which made it not acceptable for industry-scale production.

In 1998 DLR, Ciemat and Inabensa started the development of the evolution of the aforementioned HitRec, realizing a new concept called HitRec-II [6]. The receiver was characterized by 32 cups in hexagonal ceramic modules, both of the same material as the HitRec I. The supporting structure was made of steel capable of very high working temperature and presenting a similar expansion coefficient of SiSiC. The improved receiver was tested for 155 h without showing any sign of damage, operating with an outlet air temperature between 700 °C and 800 °C.

The HitRec II represents the current state-of-the-art for what concerns open volumetric air receivers in pre-commercial industrial scale. Nonetheless, several innovative concepts have been tested on experimental scale in order to improve the thermal performance of the absorber and to achieve the volumetric effect.

Fend et al. investigated the performance of several absorber samples through experimental analysis. The complete set of porous structures included three different foam samples with different cell density and three different materials (SiSiC, cordierite and clay-bound silicon carbide), SiC fiber mesh material and different configurations of improved metallic structures based on the Catrec technology (different surface densities and one sample modified...
introducing small chokes in each channel). Furthermore, a structure characterized by the combination of silicon carbide honeycomb with parallel channels and fiber mesh was also considered in the study. The experimental results showed high and promising performances for the foam, for the metallic structure with the largest surface density and for the combined honeycomb/fiber mesh structure. With those structures, more than 800 °C air outlet temperature has been achieved at highest radiative loads. In 2013, an advanced geometry based on the HiTRec II technology was investigated by Smirnova et al. [8]. The geometry was characterized by a parallel channel honeycomb with square cross section openings of 1.4 mm length and a wall thickness of 0.4 mm. Those dimensions characterized the lowest limit of the manufacturing process. The advanced sample was numerically analysed, showing an improvement of about 5% in terms of thermal performance. However, when the sample was tested at the DLR experimental facility, it showed a lack of agreement with the previous theoretical evaluation, mainly due to the occurrence of hot spots and inhomogeneous flow.

A preliminary optimization procedure of the solar absorber structure for volumetric solar receivers has been presented in Capuano et al. [9]. In this work, an extensive parametric numerical analysis has been carried out, where effective quantities such as porosity, volumetric heat transfer coefficient and radiation extinction have been varied in order to obtain an optimized selection for future design processes. A combination of high porosity, high volumetric heat transfer coefficient and increasing extinction within the absorber volume resulted to be the very important to improve the overall conjugate heat transfer between the porous solid and the air flow in volumetric solar receivers, drastically reducing the radiative heat losses, thus obtaining high value of thermal performance and the achievement of the volumetric effect. Pfeifer et al. [10] introduced a new concept of volumetric receiver using ultra-high temperature ceramics. Their concept combines the behaviour of spectral selectivity of selective coatings and the volumetric effect observed with refractory porous ceramics, showing improved performance compared to conventional porous ceramic foams with same shape characteristics. In the work of Shuja and Yilbas [11], different designs of volumetric solar receivers are presented. The structures are characterized by different configurations of absorbing blocks, arranged to obtain different behaviours concerning radiation absorption and pressure drops. The energy transfer is increased for certain arrangements in which pressure drops are minimized and absorption is distributed within the structure uniformly.

Pabst et al. [12] also obtained very good results with the use of an advanced cellular metallic honeycomb, consisting of winded pairs of flat corrugated metal foils, widely used in the automotive industry for the after-treatment of exhaust gases. As an outcome, air outlet with more than 800 °C has been reached in a 500 kW experiment, supporting the improved performance by theoretical calculations.

In the present work, an innovative micro-geometry is presented, based on the numerical outcome of the work of the same author previously published [9]. The structure presents a particular pin-shaped inlet zone, coupled with an inner staggered-honeycomb geometry, allowing a better distribution of the incoming solar radiation, reducing the inlet radiative losses and optimizing the overall heat transfer process. The thermal performance of the new geometry has been predicted using a three-dimensional CFD numerical approach, in which a unit element of such geometry is used as control volume. The computational simulation has been carried out taking into account conjugate heat transfer between the irradiated porous solid and the ambient air flowing through the structure. Thus, the predicted results have been validated using a 3:1 scaled up demonstrator in small-scale laboratory experiments, showing a good agreement between each other. Improved thermal performance have been obtained compared to the current state-of-the-art, eventually achieving the volumetric effect.

2. Methods and materials

The numerical results of the new geometry analysis have been compared to the ones characterizing the current state-of-the-art and the advanced honeycomb presented in the work of Smirnova et al. [8]. Afterwards, thermal and flow performances have been experimentally evaluated with the use of a solar simulator and an in-house developed experimental setup for the flow analysis. Thus, the outcome of the experimental campaign has been finally used as a validation of the previously predicted numerical results.

2.1. Numerical model

The discrete numerical model presented in this study is characterized by a definite representation of the three-dimensional solid geometry. Particularly, due to the complexity of such structures, unit elements and symmetry boundaries are used to represent the entire porous medium.

For the numerical calculation of the velocity and pressure fields the Navier-Stokes equations have been used in the model, according to the following equations:

$$-\nabla p + \mu \left(\nabla u + \left(\nabla u^T\right)^T\right) = \rho(u \nabla) u$$

(1)

where \( p \) is the pressure, \( \mu \) is the dynamic viscosity, \( u \) is the velocity and \( \rho \) is the density of the fluid and:

$$\nabla (\rho u) = 0, \quad \rho = \rho(p, T_f)$$

(2)

with \( T_f \) as the temperature of the fluid.

The energy exchange has been treated using Fourier’s law for the solid and the fluid phases, through the following equations:

$$\rho c_p u \nabla T_f = \nabla \left( k_f \nabla T_f \right)$$

(3)

where \( c_p \) is the specific heat capacity at constant pressure of the fluid and \( k_f \) is the thermal conductivity of the fluid;

$$- (k_s \nabla T_s) = q_r$$

(4)

where \( k_s \) is the thermal conductivity of the solid and \( q_r \) is the radiative heat density;

$$q_r = e_i (1 - \sigma_B T^4)$$

(5)

where \( e_i \) is the superficial emissivity of the solid, \( \sigma_B \) is the Stefan-Boltzmann constant and \( I \) is the direct radiative intensity source.

The direct irradiation source \( I \) is set through a built-in ray-tracer algorithm, taking into account the setup of the HLS solar simulator and it has been distributed on different vectors on planes XZ and YZ, as shown in Figs. 1 and 2. Radiative heat transfer is treated by using the direct area integration method through view factor calculations. Thus, the irradiation \( I \) can be defined as follows:

$$I = I_m + F_{amb} \sigma_B T_{amb}^4 + I_{ext}$$

(6)

where \( I_m \) represents the mutual irradiation coming from other volume boundaries; \( F_{amb} \) is the ambient view factor, whose value is
equal to the fraction of the field of view that is not covered by other boundaries. Therefore, by definition, the correlation $0 \leq F_{\text{amb}} \leq 1$ is valid for all points. $T_{\text{amb}}$ is the assumed far away ambient temperature in the directions included in $F_{\text{amb}}$, while $I_{\text{ext}}$ represents the sum of the products, for each external source, between their view factors and the corresponding source radiosity. The latter is defined as the sum of the reflected and emitted radiation that leaves the generic surface taking part in the heat transfer process as absorbing/emitting medium.

The radiation is coupled to the heat conduction equation and the following boundary condition is set on the solid surfaces:

$$n(-k_{s} \nabla T_{s}) = \varepsilon_{i} (I - \sigma_{b} T_{s}^{4}) + n \left( k_{f} \nabla T_{f} \right)$$  \hspace{1cm} (7)

where $n$ is the directional unit vector and $\varepsilon_{i}$ is the superficial emissivity of the solid surface.

External walls are set as non-absorbing surfaces, while for the fluid phase fixed values of the mass flow rate and the ambient temperature have been considered at the inlet.

The key parameter for performance evaluation of the analysed structures is the thermal efficiency, defined as the ratio between the power transmitted to the fluid and the incoming radiative power, according the following equation:

$$\eta = \frac{\Delta H}{Q_{r}}$$  \hspace{1cm} (8)

where $\Delta H$ is the energy gained by the fluid during the heat transfer process and $Q_{r}$ is the incoming radiative power and is equal to:

$$Q_{r} = I \cdot A_{i}$$  \hspace{1cm} (9)

where $A_{i}$ is the projected surface at the inlet.

$\Delta H$ can be calculated according to the following equation:

$$\Delta H = \dot{m} \left( h_{f,o} - h_{f,i} \right)$$  \hspace{1cm} (10)

where $h_{f,o}$ and $h_{f,i}$ are the fluid enthalpy evaluated at the outlet and inlet sections respectively.

Thus, Eq. (8) becomes:

$$\eta = \frac{\dot{m} \left( c_{p,o} \cdot T_{f,o} - c_{p,i} \cdot T_{f,i} \right)}{(I \cdot A_{i})}$$  \hspace{1cm} (11)

### 2.2. Experimental setups

The experimental campaign focused on the analysis of the thermal performance of the newly designed absorber sample and its flow characteristics. The experiments have been performed using two different experimental setups that will be presented in the next sections.

#### 2.2.1. Thermal efficiency analysis – setup

The High Flux Solar Simulator (HLS), coupled with a tubular test-bed, has been used to investigate the thermal performance characteristics of the absorber sample. With this configuration, it is possible to simulate boundary conditions similar to real-case applications in solar power plants for a receiver sample. The setup is characterized by a set of 10 short-arc Xenon lamps with ellipsoid reflectors, providing up to 4.5 MW m$^{-2}$ of concentrated irradiation. The latter is focused on the investigated sample, placed in a test-bed at a distance of ca. 150 cm from the lamps [7].

Only 4 lamps have been used for the performance evaluation, in order to provide a mostly homogeneous irradiation on the small front surface of the demonstrator. In Fig. 3, the setup is shown: short-arc Xenon lamps are displayed on the right side of the picture, while on the left side there is the test-bed used for the experiments. In Fig. 4, a more detailed picture of the test-bed front zone is shown. The housing of the absorber sample is surrounded by the insulating material and a liquid circuit used to cool down the part of the structure nearby the sample when the lamps are operating.

The concentrated radiation coming from the lamps ($I_{\text{exp}}$), hit the inlet surface of the absorber sample that is heated up. The blower is used to provide the pressure difference that generates the air flow in the circuit, as described in the flow-scheme in Fig. 5.

Cold ambient air flowing through the sample and then in the circuit, is heated up and its temperature is measured by three thermocouples with an accuracy of ±1%, as described in the work of
The thermocouples are placed in the positions $T_A$ and $T_B$. A counter flow air/water heat exchanger has been used in order to protect auxiliary components, like the blower and the mass flow meter, from overheating.

Additionally, surface temperature of the absorber has been measured with the use of an infrared camera while the radiation flux on the aperture of the sample has been measured using the Flux And Temperature Measurement System (FATMES) with an accuracy of ±5% [13].

For the evaluation of the thermal performance of the sample, the key parameter is the thermal efficiency, calculated according to Eq. (8). The irradiation power is calculated as follows:

$$ Q_r = I_{\text{exp}} \cdot A_i $$

where $I_{\text{exp}}$ is the homogeneous value of the irradiation source, coming from the Xenon lamps, while $A_i$ is the aperture area of the cylindrical absorber sample.

The power transmitted to the fluid is calculated according to Eq. (10). In the latter, the outlet air temperature is the average of the hot air temperatures detected by the thermocouples at position $T_B$.

2.2.2. Pressure drop analysis — setup

The analysis of the pressure drop inside the porous sample has been performed adapting the experimental setup according to Reutter et al. [14] and shown in Fig. 6. The setup is characterized by a tubular structure where a blower is used to generate the air stream inside the circuit (1). The tube is long enough to let the air flow be fully developed. Then, a standardized orifice (diameter: 22.7 mm) is placed perpendicular to the flow direction (2) and is used for the evaluation of the mass flow rate. The latter is calculated taking into account the pressure difference measured with an accuracy of ±1% by two sensors that are placed before and behind the orifice. The pressure drop across the sample has been measured, together with the air temperature, using two additional sensors placed right next to the porous sample and presenting an accuracy of ±1% [14].

For the comparison and the validation of the results obtained through numerical simulation, the specific pressure drop ($\Delta p/L$) will be considered. It is defined through the following equation:

$$ \Delta p/L = \left( p_o^2 - p_i^2 \right) / 2 \rho L $$

where $p_o$ and $p_i$ represents the air pressure evaluated at the outlet and at the inlet of the porous sample respectively, and $L$ is the sample length.

3. Results

The thermal performance of the new absorber structure, in its original design dimensions and in the demonstrator form, has been evaluated through a discrete-based simulation and compared with the current state-of-the-art HiTRec-II honeycomb absorber and an improved honeycomb geometry [15].

Eventually, the outcome of the experimental campaign will be introduced, showing a final validation concerning thermal performance and flow behaviour.

3.1. Numerical test and comparison

3.1.1. Original design

Single unit elements and corresponding fluid zones have been used as control volume for this set of simulations. The unit element of the optimized geometry is shown in Fig. 7. It presents a channel width of 1.1 mm and a wall thickness of 0.2 mm. Moreover, it is characterized by a graded open porosity ($\varepsilon$) with the highest value reached in the first section of the pin zone (0.97), down to a minimum value of 0.71 for the honeycomb back section. The specific surface area ($A_s$) ranges from the lowest value of 960 m$^2$ m$^{-3}$ in the...
first section of the pin zone, up to 2860 m² m⁻³ in the honeycomb back zone.

The unit elements of the structures considered for the numerical comparison are the HiTRec-II and an improved honeycomb geometry and their unit element are displayed in Fig. 8. The state-of-the-art is characterized by channels 2.0 mm wide, a wall thickness of 0.8 mm and 50.0 mm depth, resulting in $\varepsilon = 0.51$ and $A_v = 1020 \text{ m}^2 \text{ m}^{-3}$. The improved honeycomb has a similar shape and depth compared to the HiTRec-II but the channels are 1.4 mm wide and the wall thickness is 0.4 mm, so that $\varepsilon = 0.60$ and $A_v = 2560 \text{ m}^2 \text{ m}^{-3}$.
A tetrahedral mesh has been adopted for the discretization of the solid volume, as reported in Fig. 9. The mesh consists of 8.5 \times 10^5 elements, presenting an average growth rate of the elements of 1.7 and a maximum growth rate of 8.5. The growth rate factor is defined as the ratio between the volume of a single element and the volume of the smallest elements characterizing the mesh. A combination of tetrahedral and hexahedral elements has been used for the discretization of the fluid volume, as reported in Fig. 10. Hexahedral elements have been used for the meshing of the fluid layers in the immediate vicinity of the bounding solid surfaces. Here the effects of viscosity are significant and a strong gradient of the velocity is present. Thus, since hexahedral elements provide more precise solutions and can be better adapted to the shape of the interface, they are preferable compared to tetrahedral elements. The mesh consists of 1.6 \times 10^6 tetrahedral elements, characterized by an average growth rate of 1.4 and a maximum growth rate of 51.1, and 7.3 \times 10^3 hexahedral elements, presenting an average growth rate of 1.2 and a maximum growth rate of 10.7.

For this set of simulations, initial conditions are set according to the lay-out parameters of the Solar Tower in Jülich [16], as reported below:

- \( I_0 = 6.5 \times 10^4 \text{ W m}^{-2} \);
- \( \dot{m} = 1.1 \times 10^{-6} \text{ kg s}^{-1} \) (corresponding to: \( U_0 = 0.5 \text{ m s}^{-1} \));
- \( T_{\text{fl}} = 318.3 \text{ K} \);
- \( P_{\text{fl}} = P_{\text{amb}} = 1.0 \times 10^5 \text{ Pa} \).

The direct irradiation source \( I_0 \) is set according to the setup of the HLS solar simulator of DLR, as previously described in Section 3.1.

Siliconized silicon carbide (SiSiC) has been considered for the different geometries and temperature dependent material properties have been implemented in the model according to Munro et al. [17]. Humid air with 60% relative humidity has been considered for the fluid volume.

Mean temperature profiles of each case along the flow direction \( z \) are displayed in Fig. 11. They are the result of the interaction between the porous solid structure and the flowing air under conjugate radiation, convection and conduction. As the radiation hits the structure, the part which is not reflected and emitted from the inlet solid surface penetrates into it, generating a conductive heat flux and heating up the volume and the outer surfaces of the solid. This energy is then transmitted through convection to the fluid which is heated up to the highest value reached at the outlet.

The higher the porosity, the higher the amount of radiation that enters the porous structure; thus, the thermal performance of the absorber is enhanced. On the other hand, the thermal performance is affected by higher optical losses from the inlet solid zone in case of low porosity structures. In fact, reflection and emission losses from the inlet are proportional to the amount of surface directly hit by solar radiation and to the temperature of the solid surface to the fourth power respectively.

Moreover, when low porosity is joined by a low specific surface area, the convective heat exchange is also affected, causing a bigger temperature difference with thermal equilibrium achieved later in the structure between the solid and fluid phases.

Material thermal conductivity also plays a role in the overall heat transfer process, as it directly affects the conduction of the heat into the solid. Whenever the volumetric effect is reached and the temperature of the solid phase is characterized by a monotone rising trend, a low thermal conductivity can be beneficial in the conjugate heat transfer process, since it slows down the transport of heat by conduction to the front. Because of this, the rise in temperature toward the front is prevented, keeping a low inlet solid temperature and hence improving the thermal performance [18].

The HiTRec-II honeycomb geometry is characterized by the lowest value of porosity and the highest initial solid temperature (1234 K). It also has the lowest specific surface area, therefore, thermal equilibrium is reached nearly at the end of the depth. The thermal conductivity of SiSiC also has an influence on this behaviour, as its value drops down as a consequence of the high initial solid temperature, increasing the thermal inertia in the initial section. The combination of all those aspects leads to the lowest value of the outlet air temperature (1015 K) and thermal efficiency (\( \eta_{\text{HiTRec-II}} = 0.70 \)) of the examined structures.

Performance raises when we consider the advanced honeycomb, as it presents a higher porosity and specific surface area compared to the current state-of-the-art (\( \eta_{\text{advanced honeycomb}} = 0.74 \)).

The new geometry shows the best performance and also the presence of the volumetric effect, since the front solid temperature (842 K) is lower than the outlet air temperature (1203 K). Thermal equilibrium is reached in correspondence of 12.0 mm while the
The geometry has the same open porosity ($\varepsilon$) as the original design, as the upscaling does not affect its ratio, but it does affect the specific surface area ($A_s$). In fact, this parameter ranges from 318 m$^2$ m$^{-3}$ in the entrance section of the pin inlet zone, up to 870 m$^2$ m$^{-3}$ reached in the back honeycomb zone. In Table 1, the characteristic geometric values of the demonstrator, the new geometry in the original design and the other structures used for the numerical comparison are reported. All the simulations herein presented have been carried out with the same initial and boundary conditions and material properties as the previous cases. Only the value of the mass flow has been adapted to the enlarged geometry, corresponding to the same value of incoming fluid velocity.

The temperature profiles along the flow direction $z$ are shown in Fig. 12, comparing the demonstrator geometry with the other structures.

The new structure in the demonstrator form shows the presence of the volumetric effect, since the solid inlet temperature (1012 K) is lower than the fluid outlet temperature (1144 K), corresponding to a thermal efficiency of 0.84. The solid temperature profile is characterized by a discontinuous trend: in the pin section ($0.0 \text{ mm} \leq z \leq 30.0 \text{ mm}$), the temperature decreases firstly, from a value at the inlet surface of 1012 K, down to a value of ca. 950 K at 10.0 mm. Then it rises again up to the peak value reached in correspondence of the end of the pin zone (ca. 1192 K). After that, the temperature decreases again down to the final value reached at the end of the structure (1144 K). On the other hand, the fluid temperature profile presents a continuous increase.

The solid temperature profile is influenced by the porosity, the specific surface area and the corresponding radiation extinction of the demonstrator. The high porosity at the front is helpful for the distribution of the radiation into the porous volume, as the low extinction of the incoming sunlight in the inlet pin section reduces the initial optical losses. However, the demonstrator is also characterized by the lowest values of $A_s$ among the geometries due to the upscaling, affecting the convective heat exchange. In particular, the first section of the pin zone has the lowest value of $A_s$ (318 m$^2$ m$^{-3}$). In this part of the structure, the heat gained by the radiation which is then transmitted through conduction is larger than the one transferred to the fluid through convection. For this reason, the solid temperature pattern shows a negative trend from the beginning to ca. 10.0 mm. Furthermore, in this section also the temperature of the fluid is influenced, as it is characterized by a slight increase. After this depth, since the cross section of the pins increases and hence also the value of $A_s$ becomes larger, the convective heat transfer is enhanced, resulting in a sharper increase of both solid and fluid temperatures. A peak characterizes the solid temperature profile at ca. 30.0 mm, where the honeycomb zone starts. Here, the porosity decreases and more radiation is absorbed due to the staggering at ca. 35.0 mm. Moreover, the specific surface area in the honeycomb section, even though it is larger than the initial pin section, is still characterized by a low value ($870 \text{ m}^2 \text{ m}^{-3}$). This leads once again to a condition where the heat gained through radiation and then transmitted through conduction is higher than the one transferred to the fluid through convection, causing the peak of the solid temperature and a lower increase in the fluid temperature.

3.2. Experimental validation

In Fig. 13, the demonstrator sample used for the experiments is reported. It is characterized by a cylindrical outer shape, with a
diameter of 70.0 mm and 60.0 mm depth. For its realization, titanium-aluminium alloy (Ti6Al4V) has been used, due to material availability constraints of the manufacturer.

The variation of key parameters, such as the outlet air temperature and the corresponding thermal efficiency for different values of the ratio “power on aperture/mass flow rate” ($P/\dot{m}$) will be compared with the outcome of detailed simulations. The analysis of the pressure drops will be also displayed and the results compared with the ones obtained through the discrete numerical analysis and further values regarding the current state-of-the-art.

### 3.2.1. Thermal efficiency analysis

The thermal efficiency measurements have been performed for different values of the air mass flow rate, keeping constant the power on aperture. The corresponding numerical simulations have been carried out taking into account the experimental conditions during the tests with the HLS, as stated below:

- $I_{\text{exp}} = 7.6 \cdot 10^5$ W m$^{-2}$ (corresponding power on aperture: $2.95 \cdot 10^3$ W);
- $\dot{m} = (33.2; 11.4; 8.3; 6.0; 5.2; 4.4; 4.1) \cdot 10^{-3}$ kg s$^{-1}$;
- $T_{\text{f,i}} = 309.7$ K;
- $p_{\text{f,o}} = p_{\text{amb}} = 1.0 \cdot 10^5$ Pa.

Furthermore, the thermal performances of HiTRec-II, calculated with the discrete numerical model in the same environmental conditions, are displayed for further comparison.

Fig. 14 shows the variation of the measured outlet fluid temperature for different values of the ratio “power on aperture/mass flow rate” ($P/\dot{m}$) and corresponding numerical results are reported for comparison. They are characterized by blue and red dots respectively, while the dark-grey dashed line represents the numerical results of the state-of-the-art.

Vertical error bars for the experimental results take into account the accuracy of the thermocouples and the standard deviation of the results for each operational point, corresponding to a variation on the temperature value of 2%. Horizontal error bars consider a cumulated error of ±5% on the $P/\dot{m}$ values, taking into account the accuracy of the FATMES system used for the evaluation of the power-on-aperture and a negligible uncertainty on the evaluation of the mass flow rate [13].

The highest mass flow rate and hence the lowest ratio $P/\dot{m}$ leads to the lowest value of the measured outlet fluid temperature in all the cases displayed on the chart. As the ratio increases, characterizing lower mass flow rates, the outlet temperature rises up to the highest value, obtained for $P/\dot{m} = 705$ kJ kg$^{-1}$, corresponding to $\dot{m} = 4.1 \cdot 10^{-3}$ kg s$^{-1}$. The correlation can be represented by a linear trend. Highest values of outlet fluid temperature are obtained with

### Table 1

Characteristic geometric values of the structures analysed in the numerical comparison.

<table>
<thead>
<tr>
<th>Structures</th>
<th>Channel width [mm]</th>
<th>Wall thickness [mm]</th>
<th>Channel depth [mm]</th>
<th>Porosity $\epsilon$</th>
<th>Specific surface area $A_s$ [m$^2$ m$^{-3}$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>HiTRec-II</td>
<td>2.0</td>
<td>0.8</td>
<td>50.0</td>
<td>0.51</td>
<td>1020</td>
</tr>
<tr>
<td>Advanced honeycomb</td>
<td>1.4</td>
<td>0.4</td>
<td>50.0</td>
<td>0.60</td>
<td>2560</td>
</tr>
<tr>
<td>New geometry – original design</td>
<td>1.1</td>
<td>0.2</td>
<td>20.0</td>
<td>ca. 0.97 – 0.71</td>
<td>ca. 990 – 2860</td>
</tr>
<tr>
<td>Demonstrator</td>
<td>3.6</td>
<td>0.6</td>
<td>60.0</td>
<td>0.97 ca. – 0.71</td>
<td>ca. 318 – 870</td>
</tr>
</tbody>
</table>
the demonstrator compared to the ones of the HiTRec-II, for each operational point. This is due to the lower radiative losses of the new geometry compared to the losses characterizing the state-of-the-art geometry. Furthermore, the numerical results of the demonstrator present a good agreement with the measurements after the value of $P/\dot{m} = 350 \text{ kJ kg}^{-1}$. Before this point, their values are outside the error limit of the measurements. This condition is caused by a transient effect of the test-bed, which alters the measurements in correspondence of high mass flow rate and thus of low values of the ratio $P/\dot{m}$ [7].

The trend of the corresponding thermal efficiency, calculated according to Eq. (8), is reported in Fig. 15. The propagation error has been calculated taking into account the accuracy of the instruments involved (thermocouples, mass flow and power on aperture measurement systems) and the standard deviation of the measured values of outlet fluid temperature. The variation of experimental results has been quantified in $\pm5\%$ on the calculated efficiency and the corresponding error bars are reported for the experimental results.

The numerical trends of the demonstrator and the HiTRec-II are both characterized by a continuous decreasing trend, with the highest value obtained in correspondence of the lowest value of the ratio $P/\dot{m}$ and thus the highest value of mass flow rate. In those cases the temperature of the solid is low and hence the re-emission of the radiation, representing the highest contribution of thermal losses, is negligible [21]. However, since the main aim of the volumetric absorber is to obtain a high temperature fluid, this condition is not applicable. Thus, decreasing the mass flow rate, the system is capable to produce a high temperature air flow. As a consequence, the temperature of the solid rises, as well as the thermal losses, affecting the value of thermal efficiency. The comparison with the HiTRec-II single channel geometry shows an overall better behaviour of the demonstrator, with an increased efficiency up to 13\% thanks to the optimized inlet design that allows the minimization of radiative losses. Also in this case, the numerical results of the demonstrator present a good agreement with the measurements, characterized by the blue dots, after the value of $P/\dot{m}$ $= 350 \text{ kJ kg}^{-1}$, due to the transient effect of the test-bed on the measurements with low values of the ratio $P/\dot{m}$ [7].

3.2.2. Pressure drop analysis
The fluid pressure at the outlet has been measured for the demonstrator geometry using the experimental setup shown in section 3.2. The results are shown in the chart of Fig. 16 and they have been used for comparison with the corresponding numerical results obtained through the discrete numerical simulation. A further comparison has been made with the results regarding the HiTRec-II, obtained through detailed numerical analysis. Measurements have been performed at constant temperature for different values of incoming mass flow rate, as reported below:

- $T = 298$ K
- $\dot{m} = (16.0; 14.0; 12.0; 8.0) \times 10^{-3} \text{ kg s}^{-1}$.

The specific pressure drop ($\Delta p/L$) has been calculated according to Eq. (14). Error bars for the experimental results show a variation of $\pm3\%$, taking into account the accuracy of the instruments for the measurement of the outlet pressure and the standard deviation for each operational point. Experimental and corresponding numerical results are represented by blue and red dots respectively, while the dark-grey dashed line represents the results of the state-of-the-art.

The measured specific pressure drop ranges between 660 N m$^{-3}$ and 1670 N m$^{-3}$, corresponding to a total pressure drop of 20 Pa and 50 Pa, respectively. The numerical results calculated under the same experimental conditions show a good agreement with the experiments, with a maximum overestimation of 2\%.

As the mass flow rate increases, the pressure drop rises as well with a quadratic trend. This behaviour follows the Darby’s equation with the Dupuit/Forchheimer extension, where the specific pressure drop is expressed in relation to the flow velocity, dynamic viscosity ($\mu$) and density ($\rho$), permeability ($K_1$) and inertial ($K_2$) coefficients [22], according to the following equation:

$$\frac{\Delta p}{L} = \frac{\mu}{K_1} u + \frac{\rho u^2}{K_2}$$

where $u$ is the fluid velocity.

Thus, permeability and inertial coefficients have been calculated for the innovative geometry and are listed below:

- $K_1 = 3.0 \times 10^{-7}$ m$^2$;
- $K_2 = 4.9 \times 10^{-3}$ m.

In comparison to the HiTRec-II structure, the innovative geometry has a higher pressure drop. This is due to the staggered characterization of the demonstrator, as section variation leads to higher flow velocities and, thus, to higher values of $\Delta p/L$.

4. Conclusions
In the study presented in this article, a new geometry has been
presented to be used as solar absorber in volumetric receivers. The structure design is based on the previous results obtained by the author concerning the optimization of such structures for high temperature solar applications [9].

The new geometry has been numerically tested with the use of a discrete CFD simulation and a first performance comparison has been carried out taking into account the current state-of-the-art of volumetric receivers and an advanced honeycomb geometry produced at the DLR. Afterwards, a 3:1 scaled up demonstrator has been produced, taking into account current technology limits in manufacturing. Experimental validation of the numerical methodology and results has been performed showing good agreement between each other. Improved thermal performance and the achievement of the volumetric effect have been obtained with the innovative structure presented, resulting in a big step forward for volumetric solar receivers. However, current manufactory technology limits have forced the analysis on a scaled-up structure that still showed improved performance compared to the current state-of-the-art, but with lower performance in comparison to the original structure developed. Those performance can be enhanced even more in the mid-term future with the use of finer structures and this will be feasible thanks to the potential of manufacturing procedures in the mid-term future, like the electron beam sintering (EBM) used in the present study.

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References