Conjugate simulation of solar honeycomb receiver for high temperature heat absorption at constant incident heat flux

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Abstract
Conjugate radiation-convection-conduction simulation was conducted for a solar volumetric receiver of silicon carbide honeycomb for high temperature heat absorption at 1,000°C and higher. Simulation was made for three cases of channel cell size: 0.6mm; 1.5mm; 2.9mm. At two levels of incident heat flux 1,400 kW m⁻² and 4,200 kW m⁻², air mass flux was changed variously for optimization of working conditions. When the cell size is reduced from \( d = 2.9 \) mm to 0.6 mm, the receiver efficiency together with the air temperature at the receiver exit increase at each level of incident heat flux. At 1,400 kW m⁻², the receiver efficiency exceeds 0.8 when the air temperature is as high as 1000°C in the case of the smallest cell size: \( d = 0.6 \) mm. At 4,200 kW m⁻², the efficiency surpasses 0.80 when the air temperature is almost 1500°C in the case of \( d = 0.6 \) mm. The heat losses from the receiver was analyzed through budget of energy balance equation. It was found that the thermal radiation was attenuated by reduction of channel cell size which resulted in enhancement of the receiver efficiency. The mean temperature at the top edge of the receiver decreased with the reduction of channel size in consistency with the attenuation of thermal radiation. The numerical result demonstrated that the reducing cell size is essential to absorb concentrated solar light at very high temperatures beyond 1000°C and higher.

Keywords: Honeycomb receiver, Receiver efficiency, Conjugate radiation-convection-conduction, Simulation

1. Introduction

Solar power generation was commercialized in Europe and United States. Conventional solar power plants use molten nitrate salt as heat transfer and storage medium (Avila-Marin, 2011; Jafari Mosleh and Ahmadi, 2019). The nitrate salt can be workable below the temperature limitation of 600°C since the thermal medium pyrolyzes beyond the criterion (Turchi et al., 2018). Recently, studies have been made for volumetric receiver using air as heat transfer fluid at the higher temperatures than 600°C (Fend et al., 2004; Pabst et al., 2017). Such high temperature solar heat is promised to pave the way to highly efficient solar Brayton cycle (Giostrì et al., 2020; Ssebabi et al., 2019) and solar fuel production (Hoskins et al., 2019; Kodama, 2003).

Up to now, research works have been made for volumetric receiver for high-temperature solar heat absorption. Wu et al. (2011) used two-equation model to solve non-equilibrium thermal fields in the air and solid parts of a porous receiver. They proved that the mean cell size are dominant for local equilibrium using P1 approximation for thermal radiation. Using the same method, Wu and Wang (2013) simulated transient behavior of a receiver. Fend et al. (2013) simulated flow and temperature along the exact pore geometry using Monte Carlo ray tracing method for calculation of incoming radiation. Storch et al. (2015) studied indirectly heated solar reforming with open solar receiver. They proposed to increase heating value of natural gas by concentrated solar radiation. Chen et al. (2015) simulated a double layer volumetric receiver for improving receiver efficiency.

Although the compilation of knowledge continues to this moment, there are still ambiguity on characteristics at high
temperatures. Research works have been conducted on a honeycomb receiver irradiated by a beamdown concentrating sun simulator (Nakakura et al., 2017; Nakakura et al., 2018; Nakakura et al., 2019; Nakakura et al., 2020). It was proved that nearly symmetrical temperature field created by the beamdown optics prevented buoyant flow, which increased receiver efficiency (Nakakura et al., 2017). Conjugate radiation, convection and conduction solver was developed for direct simulation of heat transfer along the actual shape of a honeycomb channel (Nakakura et al., 2018). Efficiency loss analysis were made based on energy balance of the receiver (Nakakura et al., 2019). Cell size effects analysis revealed that increasing the air mass flow deteriorated the receiver efficiency at the power over air mass (POM) kept constant but decreasing cell size recovered the efficiency (Nakakura et al., 2020).

This paper is devoted to numerical analysis in the same direction as the previous works (Nakakura et al., 2017; Nakakura et al., 2018; Nakakura et al., 2019; Nakakura et al., 2020). The former papers varied air mass flow rate at the constant POM; however, this paper changes working conditions at the constant incidence flux from practical viewpoint. Exergy gain and mechanical power loss are examined for optimization of cell size and working condition for very high exit temperatures beyond 100°C.

2. Nomenclature

| Symbol | Description |
|--------|-------------|
| $A$ | area [m$^{-2}$] |
| $A_{out}$ | specific surface area [m$^{-2}$] |
| $A_{top}$ | area of outlet boundary [m$^{-2}$] |
| $c_p$ | specific heat at constant pressure [J kg$^{-1}$ K$^{-1}$] |
| $d$ | diameter of flow path inner [mm] |
| $e$ | exergy of air at the receiver exit [W m$^{-3}$] |
| $g$ | gravitational acceleration [m s$^{-2}$] |
| $G$ | incident radiation [W m$^{-2}$] |
| $h$ | specific enthalpy [kJ kg$^{-1}$] |
| $I$ | Radiation intensity [W m$^{-2}$ sr$^{-1}$] |
| $j_m$ | air mass flux [kg m$^{-2}$ s$^{-1}$] |
| $L$ | channel length [mm] |
| $m$ | air mass flow rate for unit channel [kg s$^{-1}$] |
| $MPD$ | mechanical power dissipation [W m$^{-2}$] |
| $n$ | refractive index [-] |
| $n_i$ | normal vector [-] |
| $p$ | pressure [Pa] |
| $POM$ | power over air mass [kJ kg$^{-1}$] |
| $q_a$ | incident radiative heat flux [W m$^{-2}$] |
| $q_{in}$ | incident radiation to the wall [W m$^{-2}$] |
| $q_{out}$ | net radiation emitted from the wall surface [W m$^{-2}$] |
| $q_{rad}$ | wall radiative heat flux [W m$^{-2}$] |
| $Q_a$ | absorption gain at the channel [W] |
| $Q_{conv}$ | convective heat transport [W] |
| $Q_{rad}$ | re-radiation which exits from the receiver [W] |
| $Q_{ref}$ | reflected radiation which exits form the receiver [W] |
| $Q_{trans}$ | transmitted radiation which exits from the receiver [W] |

| Subscripts |
|------------|
| $\Omega, \Omega'$ | solid angle [sr] |
| $\Phi$ | scattering phase function [sr$^{-1}$] |
| $\sigma$ | Stefan–Boltzmann coefficient ( = 5.669 x 10$^{-8}$) [W m$^{-2}$ K$^{-4}$] |
| $\sigma_{sea}$ | scattering coefficient [m$^{-1}$] |
| $\eta_{Carnot}$ | Carnot efficiency [-] |
| $\eta_e$ | exergy-based receiver efficiency |
| $\kappa$ | absorption coefficient [m$^{-1}$] |
| $\lambda$ | thermal conductivity [W m$^{-1}$ K$^{-1}$] |
| $\mu$ | dynamic viscosity [kg m$^{-3}$] |
| $\rho$ | density [kg m$^{-3}$] |
| $\nu$ | velocity components in Cartesian coordinates [m s$^{-1}$] |
| $\alpha$ | beam width of angle [deg] |
| $\delta$ | wall thickness [mm] |
| $\varepsilon$ | wall emissivity [-] |
| $T_{solid-top}$ | temperature of receiver top surface [K] |
| $T_{out}$ | temperature of air stream at a receiver exit [K] |
3. Numerical Method

This study deals with a honeycomb receiver with concentrated solar irradiation as depicted in Fig. 1. Air stream passes through the honeycomb, and the air is heated by the hot wall with sensible enthalpy created by incident light. Fig. 2 shows a computational domain. Only a single channel is simulated with using symmetry condition at four sections. Concentrated light enters from the upper inlet with beam width. The solid walls emit thermal radiation according to the Stefan Boltzmann law, $\varepsilon\sigma T^4$, assuming a gray wall. The walls absorb a part of the incoming light and diffusively reflect the rest. Similarly, the walls absorb and reflect the reflected light and thermal radiation emitted from the wall. The absorption of the reflected light may increase the rate of absorbed heat against the incident heat higher than the absorptivity of a wall surface.

![Fig. 1 Conjugate convection-conduction-radiation heat transfer problem for honeycomb structure.](image)

![Fig. 2 (a) Conditions of incident rays and fluid flow, and (b) computational domain of conjugate heat transfer problem for unit honeycomb channel.](image)
The governing equations are the continuity, Navier-Stokes, energy and radiative transfer equations:

\[
\frac{\partial p y}{\partial x} + \frac{\partial p y v}{\partial y} + \frac{\partial p y w}{\partial z} = 0 \tag{1}
\]

\[
\frac{\partial p y u^2}{\partial x} + \frac{\partial p y u v}{\partial y} + \frac{\partial p y v w}{\partial z} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left( \mu \frac{\partial w}{\partial x} \right) + \frac{\partial}{\partial y} \left( \mu \frac{\partial w}{\partial y} \right) + \frac{\partial}{\partial z} \left( \mu \frac{\partial w}{\partial z} \right) \tag{2}
\]

\[
\frac{\partial p y u w}{\partial x} + \frac{\partial p y u^2}{\partial y} + \frac{\partial p y v w}{\partial z} = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \left( \mu \frac{\partial w}{\partial x} \right) + \frac{\partial}{\partial y} \left( \mu \frac{\partial w}{\partial y} \right) + \frac{\partial}{\partial z} \left( \mu \frac{\partial w}{\partial z} \right) \tag{3}
\]

\[
\frac{\partial p y u w}{\partial x} + \frac{\partial p y u^2}{\partial y} + \frac{\partial p y w^2}{\partial z} = -\frac{\partial p}{\partial z} + \frac{\partial}{\partial x} \left( \mu \frac{\partial w}{\partial x} \right) + \frac{\partial}{\partial y} \left( \mu \frac{\partial w}{\partial y} \right) + \frac{\partial}{\partial z} \left( \mu \frac{\partial w}{\partial z} \right) - \rho_f g \tag{4}
\]

\[
\frac{\partial}{\partial x} \left( \lambda_s \frac{\partial T_s}{\partial x} \right) + \frac{\partial}{\partial y} \left( \lambda_s \frac{\partial T_s}{\partial y} \right) + \frac{\partial}{\partial z} \left( \lambda_s \frac{\partial T_s}{\partial z} \right) = 0 \tag{5}
\]

\[
\nabla \cdot (I(\vec{r},z)^2) = -(\kappa + \sigma_{\text{sc}})I(\vec{r},z) + \kappa n^2 \frac{\sigma T^4_s}{4\pi} + \sigma_{\text{sc}} \int_{4\pi} I(\vec{r},z') \Phi(\vec{r},z') d\Omega' \tag{6}
\]

The last term on the right hand side of Eq. (5) is written as

\[
\nabla \cdot q_{\text{rad}} = \kappa (4\sigma T_s^4 - G) \tag{7}
\]

This study assumes the refractive index, absorption and scattering coefficients, respectively as \( n = 1, \kappa = 0, \sigma_s = 0 \). The volumetric heat source term vanishes consequently. The heat generation due to radiation occurs interface between solid and fluid parts. At the interface, heat flux coincides between the solid and fluid sides as

\[
\lambda_s \nabla T_{s,\text{interface}} \cdot \vec{n} = \lambda_f \nabla T_{f,\text{interface}} \cdot \vec{n} + q_{\text{rad}} \tag{8}
\]

In this equation, the last term of the right hand side is the radiation heat flux defined by

\[
q_{\text{rad}} = q_{\text{out}} - q_{\text{in}} = -\varepsilon_w \int_{\Omega_{\text{out}}} I_\lambda \cdot \vec{n} d\Omega + n^2 \varepsilon_w \sigma T^4_{s,\text{interface}} \tag{9}
\]

Through (9) and (10), the radiation intensity is related to the heat source at the interface. Nakakura et al. (2020) explained the detailed derivation of (9) and (10).

At the inlet boundary of the computational domain, the mass flux of fluid is given the prescribed value listed in Table 1. At the inlet, the pressure is fixed at 101 kPa and the temperature is 288.15 K through all the computations. At the inlet, the radiation intensity is assumed to enter with beam width \( \alpha \):

\[
I = 0, 0 \leq \theta < \pi - \alpha/2, 0 \leq \varphi < 2\pi \tag{11}
\]

\[
I = I_{\text{in}}, \pi - \alpha/2 \leq \theta < \pi, 0 \leq \varphi < 2\pi \tag{12}
\]

At the exit boundary, the boundary layer approximation is used with neglecting streamwise diffusion terms of the basic equations.

The governing equations including the two energy equations for two phases are numerically solved with considering the full interaction between radiation, convection in fluid and conduction in solid parts. The SIMPLEC algorithm is employed for coupling the continuity and Navier-Stokes equations. The second order upwind scheme is used for the convection terms of momentum and radiative transfer equations. The QUICK was applied for convection terms of the energy equation. Assuming that flow is steady and laminar, steady state is simulated without a turbulence model. This assumption is confirmed valid since the channel Reynolds number at the exit ranges 3.6 to 1360 through the computations.

Table 1 summarizes computational conditions. The domain includes a honeycomb channel with length \( L = 50 \text{ mm} \) and inlet and outlet regions with \( L_{\text{i}} = 10 \text{ mm} \) and \( L_{\text{o}} = 20 \text{ mm} \), respectively. Simulation was conducted for the honeycomb
channels of three levels of cell size: $d = 0.6\text{mm}; 1.5\text{mm}; 2.9\text{mm}$. The porosity of the smallest and middle channels are 0.44, and that of the largest channel is 0.43. Each channel was simulated at the incident heat flux of $1,425$ kW m$^{-2}$ and $4,200$ kW m$^{-2}$. The air mass flux was changed so that the power over air mass ($POM$) prevails 400-5000 kJ kg$^{-1}$. Silicon carbide (SiC) is assumed as the material of a honeycomb channel. The simulation considered temperature dependency of properties of solid and gas phases besides optical property. Only the optical properties are set constant. The emissivity (absorptivity) of the solid wall is 0.9 with assuming the gray wall of a channel.

$$POM \,[\text{kJ kg}^{-1}] = \frac{q_0 \,[\text{kW m}^{-2}]}{j_m \,[\text{kg m}^{-2} \text{s}^{-1}]}$$

The previous work (Nakakura et al. (2019)) examined the effects from grid resolution and DO separation for the case of low incident heat flux $q_0 = 501$ kW/m$^2$. The work proved that 5x5x8 and finer DO separation satisfactorily converged the numerical variables. In this paper, the grid convergence test was conducted for 2.9mm channel at high incident heat flux $q_0 = 1,425$ kW m$^{-2}$, $4,200$ kW/m$^2$. For each level of heat flux, simulation employed two kinds of grid system with resolution 46,080 and 11,520 as tabulated in Table 2. Air mass flux is given so that $POM = 3000$ kJ/kg. DO separation is 18 x 18 x 8. Table 3 summarizes the result. As shown in the tables, grid effects are not prominent in this range of resolution. The difference of receiver efficiency is 1.6 % and 0.5 % for the lower and higher heat flux case, respectively. Based on the test, this paper adopted the grid system of 28,000, 11,520 and 46,080 grid points for 0.6mm, 1.5mm and 2.9 mm channels, respectively, in the following discussion. These grid resolutions are equivalent or finer than that of grid convergence test for 2.9mm channel.

| Channel | Cell size, $d$ [mm] | $\delta$ [mm] | $L$ [mm] | Porosity [-] | $q_0$ [kW m$^{-2}$] | $j_m$ [kg m$^{-2}$ s$^{-1}$] | $POM$ [kJ kg$^{-1}$] |
|---------|---------------------|--------------|---------|-------------|---------------------|---------------------|------------------|
| 0.6ch   | 0.6                 | 0.2          | 50      | 0.44        | 1425                | 4200                | 0.28-3.56, 0.84-10.50 | 400-5000        |
| 1.5ch   | 1.5                 | 0.5          | 50      | 0.44        | 1425                | 4200                | 0.28-3.56, 0.84-10.50 | 400-5000        |
| 2.9ch   | 2.9                 | 1.0          | 50      | 0.43        | 1425                | 4200                | 0.28-3.56, 0.84-10.50 | 400-5000        |

Table 1 Computational conditions.

| Channel | Number of grids | $\Delta x, \Delta y$ | $\Delta z$ |
|---------|-----------------|----------------------|------------|
| 0.6ch   | 28,000          | 0.1 mm               | 0.1 mm     |
| 1.5ch   | 11,520          | 0.25 mm              | 0.25 mm    |
| 2.9ch   | 11,520          | 0.48 - 0.5 mm        | 0.25 mm    |
| 2.9ch   | 46,080          | 0.24 - 0.25 mm       | 0.25 mm    |

Table 2 Grid system

| Channel | $q_0 = 1425$ kW m$^{-2}$ | $q_0 = 4200$ kW m$^{-2}$ |
|---------|--------------------------|--------------------------|
| Number of grids | 11,520 | 46,080 | 11,520 | 46,080 |
| Receiver exit temp. | 1756 K | 1733 K | 1747 K | 1741 K |
| Receiver top temp. | 2377 K | 2390 K | 1807 K | 1811 K |
| Convection power | 19.46 W | 19.41 W | 6.56 W | 6.53 W |
| Receiver efficiency | 0.551 | 0.542 | 0.547 | 0.544 |

Table 3 Grid convergence test for 2.9 mm channel
4. Results and discussion

Figure 3 shows the temperatures of air stream at the receiver exit at two levels of incident heat flux. In each, the temperature increases when the air mass flux decreases. The maximum temperature exceeds 1600°C in the case of 1400 kW m$^{-2}$, and does 2000°C in the case of 4200 kW m$^{-2}$. In the case of higher heat flux, the exit temperature rises with decrease of channel size especially when the air mass flux is smaller than 5.0 kg m$^{-2}$ s$^{-1}$. The figure also includes the receiver efficiency which is defined by the ratio between the incident heat and the sensible enthalpy of air stream:

$$\eta = \int_{A_{out}} \rho |v| h \ dA \int_{A_{top}} q_0 \ dA = Q_{conv} / Q_0$$ \hspace{1cm} (14)

Comparison of the efficiency with the air exit temperature reveals that the lower incident heat flux produces a hot air of 1000°C at the efficiency beyond 0.8 and the higher incident heat flux does a hot air of 1500°C at the efficiency above 0.8.

Figure 4 shows temperature contour for three cases of different cell size at $q_0 = 4200$ kW m$^{-2}$ and $j_m = 2.8$ kg m$^{-2}$ s$^{-1}$. The contour maps are extended laterally for visibility. In the figure, the thermal boundary layer develops in the longitudinal direction and the temperature field is nearly developed in each channel. When decreasing the cell size, the development of temperature fields becomes more rapid and the thermal inlet length shortens. However, the undeveloped effect seem not remarkable for producing hot air since the thermal field is almost developed in three cases. It is more important to note that the temperature at the receiver top decreases with decreasing the cell size. It is not so clear in the contour map and will be discussed later in more detail.

Table 4 cites the receiver efficiency, and Table 5 quotes the air temperature for some conditions. It is evident that reducing cell size comes to the increase of the receiver efficiency and the air exit temperature. At the incident heat flux of 1425 kW m$^{-2}$, the receiver efficiency exceeds 0.8 when the air temperature is 1087°C in the case of the smallest cell size, $d = 0.6$ mm. At the incident flux of 4200 kW m$^{-2}$, the efficiency surpasses 0.8 when the air temperature is 1504°C in the case of $d = 0.6$ mm. Therefore, the smaller cell size less than 1.0 mm is essential for solar absorption at very high temperatures beyond 1000°C and 1500°C.
Exergy-based receiver efficiency is defined as

\[
\eta_E = \left( \int_{A_{out}} \rho |w| h \, dA - MPD \times A_{out} \right) \times \left[ 1 - \left( 298.15 / T_{out} \right) \right] / \int_{A_{top}} q_0 \, dA
\]

\[
= (Q_{conv} - MPD \times A_{out}) \times \eta_{Carnot} / Q_0
\]

(15)

\[
MPD = \int \frac{\nu}{A} \, dp = \int \frac{f m}{\rho} \, dp
\]

(16)

The right hand side of (15) means the ratio of the exergy of exit air stream against the incident heat. The left hand side of (16) MPD is the dissipation of mechanical power due to the wall friction through honeycomb channels. The exergy-based efficiency is shown in Fig. 5 in a similar way to the previous figure. The quantities in Fig. 5 decreases from the enthalpy-based receiver efficiency due to the Carnot efficiency. The decrease is more conspicuous in the high air mass flux region at the low exit temperatures, which makes the curve of the diagram maximize at 1.3 kg m\(^{-2}\) s\(^{-1}\) for the lower incident flux, and at 2.8 kg m\(^{-2}\) s\(^{-1}\) for the higher incident flux. In the exergy-based, the low incident heat flux achieves 1000°C exit temperature at the efficiency of almost 0.6, and the high incident heat flux does 1500°C at the efficiency about 0.6.

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**Table 4** Receiver efficiency for some conditions.

| \(q_0\) [kW m\(^{-2}\)] | \(j_m\) [kg m\(^{-2}\) s\(^{-1}\)] | \(\eta\) [-] |
|-----------------|-----------------|------|
| 1425            | 1.0             | 0.81 | 0.79 | 0.74 |
| 4200            | 1.0             | 0.57 | 0.55 | 0.48 |
| 4200            | 2.0             | 0.80 | 0.72 | 0.60 |

**Table 5** Outlet air temperature for some conditions.

| \(q_0\) [kW m\(^{-2}\)] | \(j_m\) [kg m\(^{-2}\) s\(^{-1}\)] | \(T_{out}\) [°C] |
|-----------------|-----------------|--------|
| 1425            | 1.0             | 1087   | 1065 | 997  |
| 4200            | 1.0             | 2069   | 2002 | 1774 |
| 4200            | 2.0             | 1504   | 1373 | 1158 |

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![Temperature contour](image)

Fig. 4 Temperature contour for three cases of cell size at \(q_0 = 4200\) kW m\(^{-2}\) and \(j_m = 2.8\) kg m\(^{-2}\) s\(^{-1}\).
In Fig. 6, the exergy-based efficiency is presented against the air temperature at the receiver exit. The figure also presents the theoretical values based on Carnot efficiency. It is observed in this figure that the exergy-based efficiency increases closer to the Carnot efficiency when the channel size reduces. In the case of incident flux 1425 kW m\(^{-2}\), the efficiency slightly decreases from channel size 1.5 mm to 0.6 mm, which is caused by the huge loss of mechanical power from the friction in the narrow channel.

Figure 7 shows the exergy of air at the receiver exit against the dissipated power defined by above equation. Although the reduced channel size increases the receiver efficiency as shown in Fig. 5, the exergy analysis in Fig. 7 indicates that the reduction of channel size incurs huge load of fluid pumping. When the dissipated power giving the maximum exergy compared between difference channel sizes, the power increases seven to eight times at each level of the incident flux. The dissipated power is less than one tenth of exergy at most. The dissipated power is thus comparably small compared with exergy in all the cases. However, there are tremendous increases of dissipated power with channel size reduction 1.5 mm to 0.6 mm whereas the peak of exergy changes only slightly decreases at two levels of the incident flux. Therefore,
the huge load for mechanical power in the reduced channel size should be noted in designing a solar receiver system.

The energy balance of the receiver is written as follow:

\[
Q_0 = Q_{ab} + Q_{ref} + Q_{trans} = (Q_{conv} + Q_{rad}) + Q_{ref} + Q_{trans}
\] (15)

As shown in this equation, incident energy \( Q_0 \) is distributed to absorption to the receiver \( Q_{ab} \), the reflection from and transmission through it, respectively, \( Q_{ref} \) and \( Q_{trans} \). Absorption \( Q_{ab} \) is released from the receiver due to air convection \( Q_{conv} \) and thermal radiation \( Q_{rad} \). Figure 8 shows absorption, reflection and transmission for three channels. Three variables are independent of the incident flux and air mass flux, but depend on the channel geometry. The channel shape is not completely similar since the cell size changes with a constant channel length. Decreasing the cell size at the fixed channel length resulted in slight increase of absorption and decrease of reflection as shown in the figure.

![Graphs showing energy balance](image)

**Fig. 7** The exergy of air at the receiver exit against the dissipated power: (a) \( q_0 = 1425 \) kW m\(^{-2}\) and (b) \( q_0 = 4200 \) kW m\(^{-2}\).

**Fig. 8** The absorption, reflection and transmission for three channels.

Figure 9 exemplifies the terms in energy balance Eq. (15) in the case of a channel size 0.6 mm at incident flux 4,200 kW m\(^{-2}\). In this figure, the absorption is split into the convection and the radiation as described in the second expansion of the equation. The reflection and transmission take low constant values across the computed range of the air mass flux.
The convection increases and reaches near plateau in the air mass flux of \( j_m > 4.0 \text{ kg m}^{-2} \text{ s}^{-1} \). Conversely, the radiation decreases and almost stagnates in the same condition. The increase of the convection means the enhancement of the receiver efficiency which is already presented in Fig. 3. The shape of convection in Fig. 9 is the exactly same as that of the efficiency in Fig. 3 since the convection term divided by the incident radiation yields the receiver efficiency. In Fig. 3, it is evidently shown that the attenuation of the thermal radiation causes the increase of the convection which is equivalent to the enhancement of the receiver efficiency.

Figure 9 also includes the mean temperature at the top end of the channel. The mean temperature decreases when the air mass increases. It is indicated that the decrease of mean temperature causes the attenuation of thermal radiation augmenting the receiver efficiency.

![Figure 9](image1.png)  
**Fig. 9** The energy balance in the case of a channel size 0.6 mm at incident flux \( q_0 = 4200 \text{ kW m}^{-2} \).

The thermal radiation and the mean temperature at the channel top are shown compared between different channel sizes in Fig. 10. Clearly, the thermal radiation decreases, and the mean temperature does with decreases of the channel size. Therefore, augmentation of the receiver efficiency with the channel size reduction is caused by the decrease of thermal radiation which is resultant from the temperature fall at the receiver top.

![Figure 10](image2.png)  
**Fig. 10** The thermal radiation and the mean temperature at the channel top for three channels at incident flux \( q_0 = 4200 \text{ kW m}^{-2} \).
5. Conclusions

Three-dimensional simulation was conducted for conjugate radiation, convection and conduction problem of honeycomb receivers with three levels of cell size (0.6 mm, 1.5 mm and 2.9 mm). The air mass flux is changed at two levels of incident heat flux (1425 kW m\(^{-2}\) and 4200 kW m\(^{-2}\)). The conclusions extracted from the numerical results can be summarized as follow:

1) The receiver exit air temperature exceeds 1000 °C when the air mass flux decreases less than 1.0 kg m\(^{-2}\) s\(^{-1}\) at incident flux 1425 kW m\(^{-2}\). Similarly, the exit temperature rises beyond 1500 °C at the air mass flux less than 2.0 kg m\(^{-2}\) s\(^{-1}\) at 4200 kW m\(^{-2}\). In these conditions, the receiver efficiency is relatively low in the case of the largest cell size, but the efficiency increases to near 80 % in the case of the smallest cell size.

2) Analysis was conducted for the exergy of the exhaust hot air behaves and the dissipated mechanical power due to pressure loss. It was found that the exergy are not very far between three channels whereas the dissipated power increases greatly with decrease of the channel cell size. Although the dissipated power is maximally 10 % of the exergy, the huge load of the dissipation from the small channel should be noted in the design of receiver system.

3) The energy balance of the receiver was discussed through presentation of convection, thermal radiation, reflection and transmission terms of the energy budget equation. The diagram of the energy balance revealed that decreasing cell size results in the decrease of the thermal radiation increasing the receiver efficiency.

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