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Study and Modeling of a Pressurized Air Receiver to Power a Micro Gas Turbine

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Abstract: In the present work a solar receiver with reticulated porous ceramic foam bounded by two concentric cylinders, horizontal axis and length L is selected and studied. A receiver pre-sizing study based on the optimization work of Hischier allowed us to find the dimensions and the receiver input variables. We have developed a mathematical model based on the Representative elementary volume to model the flow and heat transfer within the absorber. The numerical solution of equations set was obtained with FLUENT. The power of 75 kW wanted in this study is obtained with a thermal efficiency equal to 87%. The fields of temperature and velocities from the simulation are analyzed and it is clear from this study that the temperature profiles show the excellent ability of the receiver to transfer the heat to the fluid. The influences of the porosity and mass flow on the thermal efficiency are analyzed also. It emerges from this study that the mass flow rate and porosity are very critical parameters on the thermal performance of the receiver.

INTRODUCTION

The development and implementation of renewable energies are essential as an alternative to the concerns caused by the depletion of fossil reserves and CO_2 emissions. Nowadays solar thermodynamic is attracting a lot of interest because its benefits are large including high energy efficiency, as well as great prospects of technological improvements [1]. Central receiver systems using solar tower technology have a higher potential thanks to strong achievable concentration. In these types of plants, the receiver plays a crucial role, it intercepts and converts incident sunlight into thermal energy and turns this heat into a fluid, so it has to meet the specific requirements of the installation. To do this, a better knowledge of receiver functioning is required. The goal of this work is to model a receiver and to analyze the influence of the input parameters and the physical properties of the porous medium on the fields of velocities and temperature as well as the thermal efficiency of the receiver so that it can meet the different micro-turbine needs.

DESCRIPTION AND PRE-SIZING OF THE SOLAR RECEIVER

Description of the Model

The receiver studied here is a receiver with a reticulated porous ceramic absorber which is simple and cheaper. This type of receiver may obtain an air outlet temperature of 1000 ° C with an efficiency of 78% [2]. This concept is shown schematically in Figure 1 [3]

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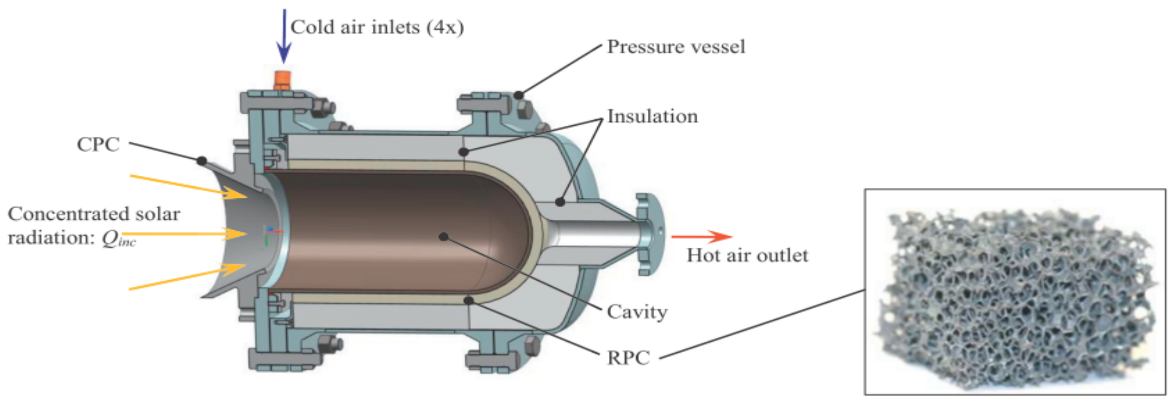
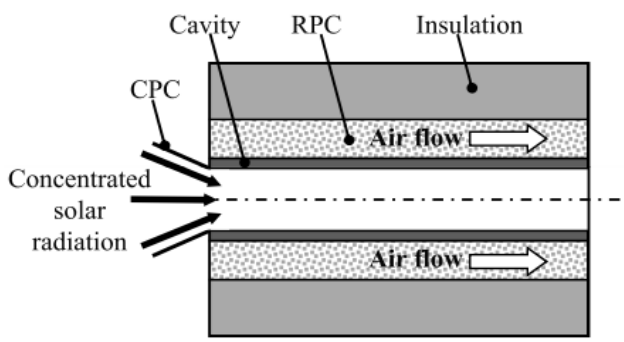


(a)



(b)

FIGURE 1. Presentation of the receiver model

The components of the receiver design are:

- CPC, which is the second concentrator, incorporated in the front to minimize radiation losses.

- The cavity that absorbs concentrated solar flux and forwards it by conduction to the RPC

- The RPC which acts as absorber is made of ceramic foam.

- The outer cylinder acts as an insulator and maintains the pressure at a certain value [4].

The material used for the CPC is aluminum chosen due to its good resistance to high temperatures and its high reflectivity (0.84). For the cavity and the porous matrix silicon carbide SiC is adequate and Al_2O_3-SiO_2 metal alloy can be used as the outer cylinder. The properties of these materials are taken from the thesis of Hischier [4]. For the optical properties of the ceramic foam including the coefficients of absorption, scattering and destruction, we got them through formulas of Wu Zhiyong and Wang Zhifeng [5].

$$D_v = 1.5(2-\alpha)\frac{(1-\varphi)}{d} = 97m^{-1} \tag{1}$$

$$\alpha_v = 1.5\alpha\frac{1-\varphi}{d} = 80m^{-1} \tag{2}$$

$$\beta = \alpha_v + D_v = \frac{3(1-\varphi)}{d} = 177m^{-1} \tag{3}$$

Presizing of Receiver

Receiver dimensions (table 1.) are closely linked to the operating conditions and heliostat field. In our case we will dimension the receiver without the CPC and in accordance with the turbine parameters. We have a preset value of the supplied electric power which is 25 kW. Taking into account the performance of the always predefined cycle that used is 33.33% we obtain an input power equal to 75 kW. This power corresponds to the thermal output heat of the receiver. Based on the thesis of Kretzschmar [2] we said earlier that this type of receiver has efficiency equal to 78%. The power at the aperture is finally equal to 96.15 kW. This value allowed us to use initially the geometric dimensions of solar receivers with an incident flux q = 100 kW found in the optimization work in Hischier's thesis [4]. With an arbitrary CPC's efficiency equal to 0.8 the incident flux at the aperture of the receiver without the CPC is equal to 76.92 kW.

TABLE 1. Dimensions of the receiver

Parameters	Dimensions (m)
Diameter of cavity	0.5
Diameter of aperture	0.25
Thickness of cavity	0.01
Thickness of RPC	0.02
Thickness of insulator	0.1
Length of cavity	0.5

MODELLING OF THE AIR FLOW AND HEAT TRANSFERS WITHIN THE RECEIVER

Mathematical Formulation

Reference Framework: Representative Elementary Volume (REV)

To quantify a physical problem by using mathematical tools, it is necessary to have a continuous medium to establish relationships with partial derivatives. In porous media physical properties (porosity, permeability...) are discontinuous at the microscopic level. To overcome this discontinuity we introduce the notion of representative elementary volume (REV) which consists in associating mathematically the properties of a volume sufficiently representative to define or measure the average properties of the volume [6].

Physical Problem and Simplifying Assumptions

We aim to study the forced convection flow in an annular space filled with a porous material delimited by two concentric cylinders having a horizontal axis of length L. At the two entrances, air is injected with a velocity and temperature inlet.

To model the transfers we accept that

- The porous medium is homogeneous and isotropic
- The regime is permanent
- The flow in the channel is incompressible, turbulent and two-dimensional
- The fluid is Newtonian
- The viscous dissipation in the energy equation and the work of the pressure forces are negligible
- The assumptions of hydrodynamic and thermal boundaries layers are valid

Under the conditions of validity of the model of Forchheimer - Wooding, equations of continuity, momentum and energy are written in cylindrical coordinates

$$\frac{\partial u_r}{\partial r} + \frac{\partial u_z}{\partial z} = 0 \tag{4}$$

$$\frac{\rho}{\varphi^2}\left(u_r \frac{\partial u_z}{\partial r} + u_z \frac{\partial u_z}{\partial z}\right) = -\frac{\partial P}{\partial z} + \mu\left\{\frac{1}{r}\frac{\partial}{\partial r}\left[r.\frac{\partial u_z}{\partial r}\right]\right\} + \left[-\frac{\mu}{K} - F\rho\left|u_z\right|\right].u_z \tag{5}$$

$$(\rho C_p)(u_r \frac{\partial T}{\partial r} + u_z \frac{\partial T}{\partial z}) = k_{eff} \left\{ \frac{1}{r} \frac{\partial}{\partial r} \left[.r \frac{\partial u_z}{\partial r} \right] \right\} - div(\vec{q}_r) \tag{6}$$

$\frac{\mu}{K} u_z$: Expression of ***Darcy,*** $F \rho |u_z| . u_z$: expression of ***Forchheimer*** (microscopic inertial forces) with F coefficient of ***Forchheimer***, $\vec{q}_r$ is the radiative flux density vector. k_{eff} and μ_{eff} are the dynamic viscosity and the effective thermal conductivity of the porous medium.

The divergence of the radiation flow that occurs at the heat transfer equation is obtained by integration of the radiative transfer equation.
In the event that fluid particles are animated by a very low barycentric speed compared with the radiation speed, the radiative equation transfer integrated across the spectrum is reduced to the following quasi stationary equation

$$\frac{1}{\alpha} div[\vec{e}.L(M,\vec{e})] + L(M,\vec{e}) = \frac{n^2.\sigma.T^4}{\pi} \tag{7}$$

The divergence of the radiative flux density vector is given by

$$div(\vec{q}_r) = 4.n^2.\sigma.\alpha_p.T^4 - \int_{\Omega=4\pi} \left[\int_{\nu} \alpha_{\nu}.L(M,\vec{e}).d\nu \right] d\Omega \tag{8}$$

Boundaries Conditions

Hydrodynamic Boundaries Conditions

At the entrance z=0

$$u_z(r,0) = u_{débit} \tag{9}$$

Where $u_{débit} = \dot{m} / \rho.\pi.R^2$; $u_r(r,0) = 0$

At the external and internal walls we have

$$u_z(R_e,z) = u_r(R_e,z) = u_z(R_i,z) = u_r(R_i,z) = 0 \tag{10}$$

Thermal Boundary Conditions

- *Boundary conditions at the wall* $r = R_e$

This wall is adiabatic we have $\vec{q}_{cond}.\vec{n}_p = 0$.

- • *Boundary conditions at the wall* $r = R_i$

The face of the inner cylinder in contact with the porous matrix is in thermal equilibrium. It is therefore

$$0 = (\vec{q}_{cond} + \vec{q}_{r,f}).\vec{n}_p \tag{11}$$

Where $q_{r,f} = k_{eff}.\partial T / \partial r$

The rear face of the inner cylinder is subjected to an assumed constant solar flux of density q_{sol}.
The heat balance is expressed by:

$$q_{sol} = (\vec{q}_{cond} + \vec{q}_r).\vec{n}_p \tag{12}$$

Where $\vec{q}_{cond}$ and $\vec{q}_r$ vector densities conductive and radiative flux. Assuming that the wall of the inner cylinder is thermally thin, we get

$$q_{sol} + q_{r,p} = 0 \tag{13}$$

The expression of the net radiative flux density is achieved through the radiosity method if the temperature is not uniform on the wall [7]. We get the following expression;

$$q_{r,p}(R_i,z) = J_p(R_i,z) - \Sigma_l^L \int_{Sl} J_l K(\vec{r}_p;\vec{r}_l).dS_l \tag{14}$$

Where $\vec{r}_p$ et $\vec{r}_l$ are the positions of the elementary surfaces dS_p et dS_l and $K(\vec{r}_p;\vec{r}_l)$ the ring defined by

$$K(\vec{r}_p;\vec{r}_l)=\frac{(\vec{n}_p.\vec{r}_{p,l}).(\vec{n}_l.\vec{r}_{l,p})}{\pi.\left|\vec{r}_{p,l}\right|^4} \tag{15}$$

Where $\vec{n}_p$et $\vec{n}_l$ the normal unit respectively of dS_p and dS_l. In this relationship we pose

$$\vec{r}_{p,l} = -\vec{r}_{l,p} = \vec{r}_l - \vec{r}_p \tag{16}$$

The radiosity leaving the wall is given by

$$J_p(R_i,z) = \varepsilon_p \sigma T_p^4 - (1-\varepsilon_p)\sum_{l=1}^{l}\int_{S_l} J_l.K(\vec{r}_p;\vec{r}_l).dS_l \tag{17}$$

Considering the input and output as black surfaces to the temperature T_i and T_0 we obtain

$$J_p(R_i,z)=\varepsilon_p\sigma T_p^4-(1-\varepsilon)\sigma.\int_{S_e}T^4(r,z=0).K(\vec{r}_p;\vec{r}_e)dS-(1-\varepsilon_p)\sigma\int_{S_S}T_S^4(r;z=L).K(\vec{r}_p;\vec{r}_s)dS_S-(1-\varepsilon_p)\int_{S_{p'}}J_p.K(\vec{r}_p;\vec{r}_{p'})dS_{p'} \tag{18}$$

The last term represents the radiative contribution inter-area.

Nondimensionalized Equations

To display these nondimensionalized numbers introduce the variables following references: Reference length L_r = L the length of the absorber. Reference Velocity: $V_r = u_{débit}$ flow velocity. Reference pressure $P_r = \rho V_r^2$. Reference flux density $q_r = q_{sol}$: net solar flux on the outside of the inner cylinder or cavity. Reference temperature difference calculated from the conductive flux density. The nondimensionalized equations are:

$$\frac{\partial u_r^*}{\partial r^*}+\frac{\partial u_z^*}{\partial z^*}=0 \tag{19}$$

$$\frac{1}{\varphi^2}(u_r^*\frac{\partial u_z^*}{\partial r^*}+u_z^*\frac{\partial u_z^*}{\partial z^*}) = -\frac{\partial P^*}{\partial z^*}+\frac{\mu^*}{\mathrm{Re}_L}\left\{\frac{1}{r^*}\frac{\partial}{\partial r^*}\left[r^*.\frac{\partial u_z^*}{\partial r^*}\right]\right\}+-\frac{1}{Da.\mathrm{Re}_L}u_z^* - F.L.\left|u_z^*\right|.u_z^* \tag{20}$$

$$u_r^*\frac{\partial\theta}{\partial r^*}+u_z^*\frac{\partial\theta}{\partial z^*}=\frac{k^*}{Pe_L}\left\{\frac{1}{r^*}\frac{\partial}{\partial r^*}\left[r^*.\frac{\partial\theta}{\partial r^*}\right]\right\}-\frac{1}{Bo}.div^*(\vec{q}_r^*) \tag{21}$$

$$\frac{1}{\alpha^*}div^*\left[\vec{e}.L^*(M,\vec{e})\right]+L^*(M,\vec{e})=\frac{n^2.\theta^4}{\pi} \tag{22}$$

Thus the values of these numbers and characteristic variables allow us to obtain information on the solution before the problem is solved.

RESULTS AND DISCUSSIONS

Results

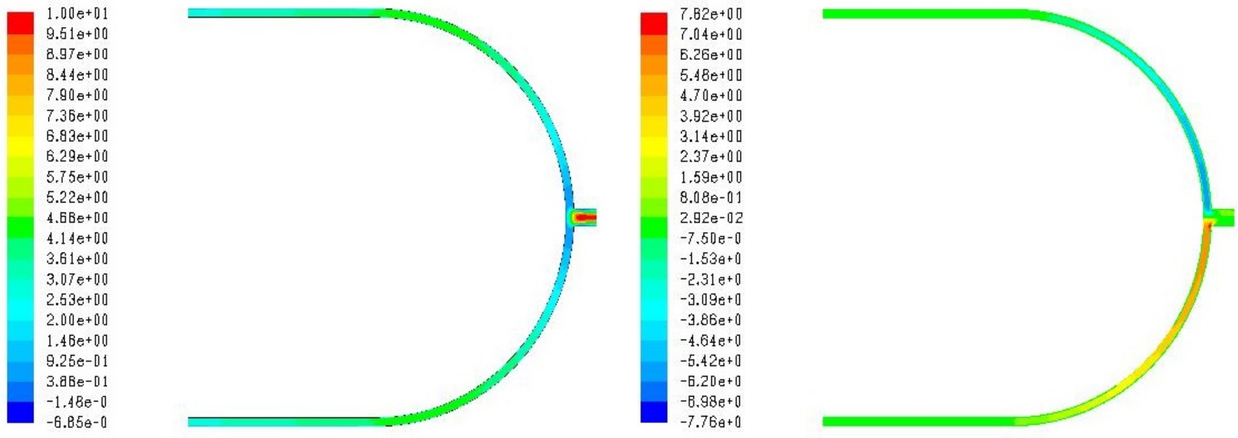
The figure is drawn on GAMBIT and dimensions given earlier are respected. To validate our computational code we compared the results with those obtained by Hischier [4] and the agreement is acceptable.

The values of temperatures in the boundary conditions were calculated by a numerical code in FORTRAN. Edge effects have been neglected that is upon the basics sections temperatures. Then with a text file we have incorporated the results in FLUENT. The turbulence model used is k-epsilon with the option Realizable. The radiation model used in the RPC is the discrete ordinate method.

The thermal efficiency is given by $\eta_{thermique} = \dot{m}(h_{oulet} - h_{inlet})/q_{incident}$

Where h_{outlet} and h_{inlet} are the enthalpies of the fluid respectively at outlet and inlet. And $q_{incident}$: Incident flux.

The power 75 kW wanted at the receiver outlet is obtained with a mass flow inlet $\dot{m}$ =0.14 kg/s and a temperature inlet T_{inlet} =450 K; we obtain a T_{outlet}=945 K and a thermal efficiency equal to 87%. The Figure 2 shows the contours velocities



(a) (b)

FIGURE 2. Velocities contours along x (a) and along y (b)

On the horizontal contour the velocity on x increases until the descent which can be assimilated to an elbow. Just after the bend we note a considerable decrease in velocity on x but is met by an increase of velocity on y. This is due to the continuity equation. The figures show that the average velocity on x is higher than that on y. But local velocities on y are higher than on x. Therefore this result confirms our previous assumptions in the momentum equation, the velocity on x is greater than on y but the velocities change are more important.

The Fig 3 shows the coolant temperature and contours of the walls of the absorber.

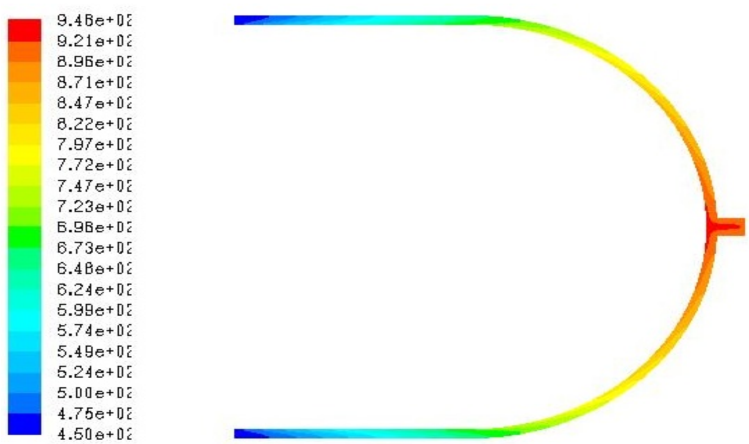


FIGURE 3: Temperature contours of the absorber

The fluid comes in both blue inlets as shown with the same inlet temperature 450 K and the same velocity. And we have the exit at right side. The dimensions given earlier have been respected. The figure shows the temperature profile of the fluid inside the absorber. This figure illustrates the good ability of the receiver to transfer the heat to the fluid.

There are many parameters which can affect the thermal efficiency of our receiver. We chose in this work two of them.

Influence of the Porosity on the Efficiency

The porosity characterizes the physical absorber nature, Fig. 4. shows its influence on the efficiency.

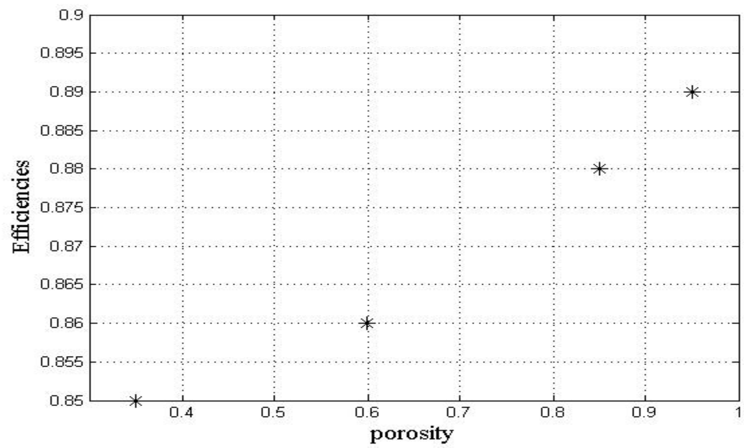


FIGURE 4: Thermal efficiencies based on the porosity

The porosity and permeability are the main properties of a porous medium. The porosity is defined as the quotient of the pore volume and the total volume of medium. Thus, if the porosity increases with the pore volume, the exchange surface between the fluid and the porous matrix increases, which significantly affects the convective flux thereby increasing thermal efficiencies. Porosity is a physical characteristic of the environment. The materials used for this type of receiver must be carefully chosen because its performance depends on them.

Influence of Mass Flow on Efficiency

Figure 5 shows the variation of the thermal efficiency based on the mass flow. The curve is plotted with MATLAB.

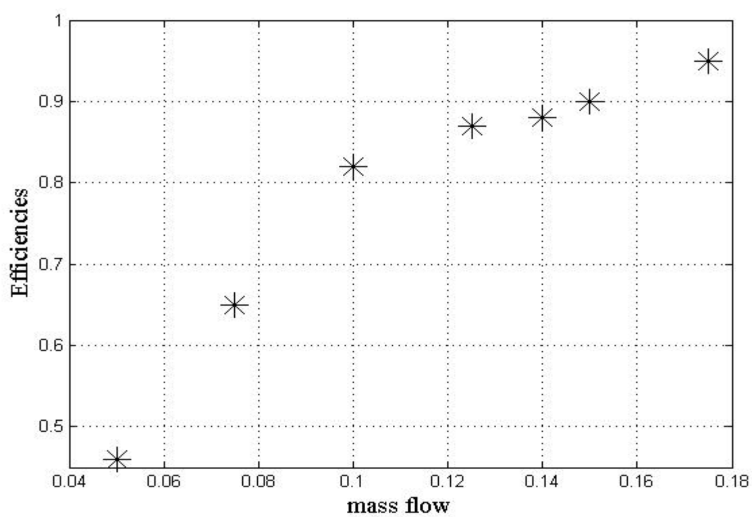


FIGURE 5 : Thermal efficiencies based on the mass flow

We note that it is not very interesting to work with low mass flows even if they provide higher outlet temperatures because the fluid spends more time in the medium and therefore exchanges well with the heating and the die cavity porous. With a mass flow equal to 0.05 kg/s we have a temperature outlet equal to 1103.5 K. So increasing the outlet temperature is at the expense of the thermal efficiency as shown in Fig 5. The thermal efficiency of the receiver varies proportionally with the mass flow but the variation is not linear because the mass flow also has an influence on the temperature outlet which in turn has an influence on the efficiency. The mass flow choice is very important for good performance of the receiver so it is very imperative to control them according to the needs.

We note a clear difference between our thermal efficiencies with those experimentally obtained by Hischier in Israel. This difference is explained by the fact that we got our power with a much higher mass flow in the order of a hundred g/s unlike him who worked with some units of g/s and also we got a lower outlet temperature. And the analyze of the Figure 5 shows that a higher mass flow gives a higher thermal efficiency and a lower outlet temperature. Thus the results show good agreement with references.

CONCLUSION

This study enables us to better understand the functioning of a solar receiver. We modeled the receiver by considering one phase in the RPC and we got interesting results which confirm the good ability of the receiver to provide hot fluid. It is also interesting to see the effect of the different parameters on the performance. The mass flow that characterizes hydrodynamic properties in the receiver plays a key role so it is very important to adapt its value according to the needs. The porosity characterizes the physical nature of the absorber and a good efficiency depends on its high number. In our future work we will study in detail the influence of the inlet temperatures and thermal operating conditions.

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