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PARAMETRIC STUDY OF HIGH-TEMPERATURE VOLUMETRIC SOLAR ABSORBERS

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Abstract. *Solar energy has been a major target for renewable energy investment in the past years due to its extraordinary potential when compared to other energy sources, with the purpose of increasing the conversion efficiencies and making it profitable, while exploring new ways of harnessing more solar radiation. This study is focused on one of the most recent technologies regarding solar energy, the solar power tower, which is a prospective solution to replace the common coal power plants. It is concerned with the reticulated porous ceramic (RPC) material used to absorb the reflected radiation intensity at the top of the tower and transfer that energy to a working fluid. In this work, a simple one-dimensional computational model, taking into account the heat transfer by conduction, convection and radiation, is used to predict the performance of the RPC absorber. The influence of individual working parameters, such as the thermal conductivity, the porosity of the absorber and the fluid mass flow rate is investigated with the purpose of increasing the solar to thermal conversion efficiency at the volumetric receiver, and several case studies are presented.*

Keywords: *Solar energy, Solar power tower, Volumetric solar absorber, Reticulated porous material, Heat transfer model*

1. INTRODUCTION

During the last decade, with the consciousness of oil scarcity and inflated prices, many developing countries decided to put in place renewable energy support policies. Indeed, the number of countries with these policies tripled in the last 10 years, and emerging economies represent almost 68% of the global effort. Solar power, which represented 53% of the total investment in 2013, saw a record level of new installations despite, the 20% investment decrease on the area (REN21). Our sun, an average star, is a source of huge amounts of power when compared to other natural resources, like the wind or water streams, and even more impressive when compared with the total known reserves of fossil fuels, but it still represents no more than 3% of the total energy consumption worldwide (REN21).

The reasons for the low share in the energetic mix are connected with a number of factors that substantially reduce the effectively available energy at the planet surface, like the atmospheric conditions, as well as our inability to efficiently convert the solar radiation into useful forms of energy. The later reason is obviously bound with the state-of-the-art of the various technologies used to transform solar energy, either in electrical or thermal power, which have only begun to be truly developed very recently.

Concentrating solar power (CSP) is the name given to the most recent solar energy absorption technologies, which aim to be an alternative to fossil fuels (coal) as the heat source in common power plants. The concept is quite simple: with the help of mirrored surfaces, sunlight is focused on thermal receivers, also referred to as thermal absorbers, where a working fluid can reach high temperatures and then be used to generate the steam for the production of electric power at a turbine. There are currently 3 different types of CSP technologies (trough, dish/engine and power tower systems), all with the same objective of generating electricity using the concentrated solar irradiance as a heat source, and this work is concerned with the latter. As the name says, the solar thermal energy is concentrated at a tower, instead of individual receivers coupled with the reflectors. This technology allows for much higher temperatures as a field of flat reflectors, often called heliostats, focus the sunlight into a thermal receiver placed at the top of the central tower. The heat transfer fluid is then used to make steam, which produces electricity by actuating a turbine.

Cellular ceramic materials have been studied for decades, and have many heat transfer applications, including solar absorbers. Fu *et al*. (1998) studied the convective heat transfer in five different cellular ceramics and developed an empirical correlation for the Nusselt number. Wu *et al*. (2010) modelled the structure of a porous material at the pore level, to study the pressure drop in the flow. A correlation for the specific surface area was proposed as a function of the porosity and pore diameter. Mey *et al*. (2013) performed one dimensional studies to predict the temperature profiles and radiative losses and efficiencies on silicon carbide (SiC) receivers. Kribus *et al*. (2013, 2014) developed a simple model to investigate the performance of the solar absorber. A similar model is used in the present work, and several parametric studies are reported to investigate the dependence of the efficiency of the solar absorber on those parameters.

2. HEAT TRANSFER MODEL

2.1. System description

As stated above, the objective of this work is to study the solar receivers used in power tower systems, which are usually made of porous ceramic materials, like silicon carbides (SiC) or mullite. The radiation reflected by the heliostats hits the porous receiver at the top of the tower, heating it to very high temperatures, and creating a heat exchange between the solid matrix and the working fluid that is passing through the receiver. That hot fluid is then used in regular steam turbine systems as the source of heat. The thermal energy may be stored during production peak hours to ensure the generation of electricity even without sunlight.

One of the main limitations of these high temperature systems are the thermal losses. Being subjected to the heat flux equivalent to several hundred suns it is expected that the receiver reaches unusually high temperatures, which is excellent for the heat transfer rate, but causes considerable heat losses due to radiation at the front surface.

The so-called volumetric effect, expected in these porous receiver systems, is defined as the existence of lower temperature at the front side of the absorber, such that the emission of radiation to the environment is reduced, and occurs due to the higher heat transfer capacity of the working fluid at the front of the absorber. However, this volumetric effect has not been experimentally obtained, as the convective heat transfer fails to compensate the high influx of radiation, limiting the absorber efficiency to around 75% for high fluid temperature values.

Higher efficiencies up to 95% are predicted using local thermal equilibrium (LTE) models, which assume that the fluid and solid states have the same temperature at any point of the absorber, thus producing the volumetric effect. Although these models predict much better results, they have not been confirmed by experimental results and constitute an over-simplification of the problem. Therefore, a local thermal non-equilibrium (LTNE) model will be used in this work to predict the heat transfer results in the receiver.

LTNE models provide a more detailed analysis of the problem, by solving coupled equations for convective, diffusive and radiative heat transfer, and are known to predict realistic results. These results, however, do not yield a volumetric effect, as the temperature of the solid absorber is not artificially forced by the fluid inlet temperature.

2.1.1. Balance equations and discretization

Since the absorbers are made from relatively low thermal conductivity materials, the most significant heat transfer is due to the flow and radiation transport along the axial direction, perpendicular to the front surface of the absorber. This allows for a one-dimensional simplification of the problem, as a small loss in the results precision is compensated by the reduced computational time.

The volumetric absorbers, made of open-cell ceramic foams (reticulated porous ceramics, RPC), are modeled as homogeneous effective media combining two continuous phases (solid and fluid). By averaging the transport equations on a representative elementary volume, smaller than the problem scale, this approach constitutes a significant reduction in modeling complexity without compromising the validity of the results.

The porosity, as well as the mean pore diameter and solid thermal conductivity, are assumed to be constant throughout the length of the absorber. The heat balance is applied separately to the solid and fluid phases, consistent with the LTNE model. The porosity determines the available conduction area and effective thermal conductivity of the absorber. The balance equation for the solid phase is written as follows (Kribus *et al*., 2014):

$$(1-\phi)k_s \frac{d^2 T_s}{d x^2} = h_v \left[T_s(x) - T_f(x)\right] + \frac{d q_R}{d x} \tag{1}$$

where ϕ is the porosity, representing the volumetric fraction of "empty space" in the total volume of the absorber, k_s is the solid thermal conductivity, h_v is the volumetric heat transfer coefficient, T_s and T_f represent the temperatures of solid and fluid phases, respectively, and the last term is the divergence of the radiative heat flux, which is a balance between the absorbed and emitted radiation by the solid phase.

Transport by conduction is not taken into account for the fluid phase, since it is expected to be significantly smaller than the transport by advection. It is also assumed that the fluid is completely transparent to radiation. The balance equation for the fluid phase is given by (Kribus *et al*., 2014):

$$\dot{m} c_p \frac{d T_f}{d x} = h_v \left[T_s(x) - T_f(x)\right] \tag{2}$$

In Eq. (2), $\dot{m}$ represents the fluid mass flow rate per unit area, which is constant through each cross section of the absorber due to the one-dimensional assumption, and c_p is the specific heat of the fluid. The pressure drop in the porous medium along the axial direction is calculated by a Darcy-Forcheimer type correlation taken from experimental and numerical studies of ceramic foams for solar receiver applications (Wu *et al*., 2010). The fluid considered in this work is air, as it is the most common one in these applications, and the one that offers a wider database of properties and correlations for the required working conditions.

The volumetric heat transfer coefficient was estimated using the correlation of Fu *et al*. (1998). A comparison of several correlations reported by Kribus *et al*. (2014) showed that this correlation gave the lowest deviation (within ±20%) from experimental results, with reasonable parametrical dependence and a wide range of applicability. The selected correlation yields an average value of the Nusselt number across the entire absorber. This means that, although the properties of the air and the flow velocity are calculated for each node in every iteration, only one volumetric heat transfer coefficient is prescribed for the entire absorber using the averaged values. The convective heat transfer coefficient in the developed region was obtained by dividing the volumetric value by the surface to volume ratio. This ratio was determined using an empirical equation developed by Wu *et al*. (2010) for the material under consideration.

Equations (1) and (2) have been discretized along the domain using finite differences and solved iteratively to find the temperature values for both phases in each point of the mesh. A uniform mesh along the absorber thickness was used. A central differencing scheme was used to discretize the diffusion term for the solid phase, while an upwind differencing scheme was used to discretize the advection term for the fluid phase.

2.1.2. Thermal boundary conditions

The boundary condition of the energy equation for the solid phase, at the front surface of the absorber (x=0), states that the net radiative energy, i.e., the difference between the absorbed and the emitted radiative energy, per unit of time, is equal to heat transferred by conduction to the absorber and by convection to the fluid (Kribus *et al*., 2014):

$$\alpha q_{in} = \alpha \sigma T_s^4(0) - k_s \left.\frac{d T_s}{d x}\right|_0 + h_a \left(T_s(0) - \frac{T_f(0) + T_{f,in}}{2} \right) \tag{3}$$

In Eq. (3), all the terms are related to the solid phase, and so the solid fraction is omitted in all of them. $T_{f,in}$ is the fluid temperature upstream of the absorber, q_{in} is the incident solar radiation, σ is the Stefan-Boltzmann constant and α is the surface absorptance. The heat transfer coefficient at the aperture, $h_{a,}$ was set to 1.6 times the convection coefficient in the developed region, based on the analysis of Wu *et al*. (2011).

The boundary condition of the energy equation for the fluid phase, at the front surface of the absorber (x=0), is expressed as follows:

$$\dot{m} c_p \left(T_f(0) - T_{f,in} \right) = (1-\phi) h_a \left(T_s(0) - \frac{T_f(0) + T_{f,in}}{2} \right) \tag{4}$$

Similarly to the thermal balance equations (1) and (2), the boundary conditions (3) and (4) are not independent, as both include the solid and fluid phase temperatures at the front surface. The last thermal condition to be defined is the second boundary condition for the solid phase. Like the first one, it is a balance between the conduction in the solid absorber, convection to the working fluid and radiation (Kribus et al. 2014):

$$-k_s \left.\frac{d T_s}{d x}\right|_L = q_{net,ex} + h_b \left(T_s(L) - \frac{T_f(L) + T_{f,ex}}{2} \right) \tag{5}$$

In this equation, $T_{f,,ex}$ is the temperature of the fluid downstream of the absorber and $q_{net,ex}$ is the net radiative power for the solid phase at the back surface of the absorber. If the back wall is a perfect reflector, there are no radiative losses at that surface. The convective heat transfer coefficient at the back surface, h_b, is obtained from the volumetric value and the surface to volume ratio, and is used to calculate the heat transfer between the solid and fluid phases at x=L.

2.2. Radiation model

2.2.1. Discrete ordinates method

The absorber is assumed to be a homogeneous medium with effective radiative properties, as far as radiative transfer is concerned. The discrete ordinates method was selected to solve the radiative transport problem in the volumetric absorber, which was modeled as a plane slab of a homogeneous participating medium. The choice of this method is bound to its good accuracy and the relatively low computational requirements when compared to other methods. The S_4

approximation was employed in the present calculations, since it was found to give an accurate representation of the spatial and angular distribution of the radiation intensity in the participating medium.

The discretization of the radiative transfer equation for a direction of propagation defined by direction cosine ξ^m may be written as follows (Modest, 2003):

$$\xi^m \frac{d I^m}{d x} = -\beta I^m + \beta(1-\omega)\phi\, I_b + \frac{\beta \omega}{4\pi}\sum_{j=1}^{M} w_j\, I^j\, \Phi_{jm} \tag{6}$$

where m denotes a discrete direction, M being the total number of discrete directions, I^m represents the radiation intensity along the mth direction, I_b is the blackbody radiation intensity, β denotes the extinction coefficient, ω is the scattering albedo, and Φ_{jm} is the scattering phase function for the pair of directions j and m.

The radiative balance equations, like the thermal balance equations, are not independent, since the scattering term couples together all the radiation intensities for all discrete directions. However, due to symmetry reasons, only 4 different directions need to be calculated to represent the 24 discrete directions of the S_4 quadrature for one-dimensional problems, namely two for positive x directions and two for negative x directions.

The dependence between the radiative transfer equation and the thermal balance equations is also explicit here, as the absorber temperature field is required to calculate the blackbody radiation intensity term. The influence of the radiation on the thermal balance equations appears in the radiative source term of Eq. (1), which may be evaluated as follows (Modest, 2003):

$$\frac{d q_R}{d x} = \beta(1-\omega)\left[4\pi\phi\, I_b - \sum_{j=1}^{M} w_j\, I^j\right] \tag{7}$$

2.2.2. Radiative boundary conditions

The incoming radiation intensity at x=0 along the discrete directions was evaluated from the incident solar radiation in such a way that the first and second moments of the radiation intensity are exactly satisfied by the discrete ordinates method. The first moment of the radiation intensity corresponds to the incident heat flux. The directional distribution of the solar radiation intensity was defined as uniform within a cone of 45º, and zero outside that cone.

At the back boundary of the absorber, the incoming radiation intensity is calculated by solving a simple problem for radiative transfer in an enclosure with a non-participating medium and bounded by three diffuse and opaque surfaces, namely the solid absorber edge, the void space of the absorber, and the back wall. Details on the implementation of the radiative boundary conditions may be found in Kribus *et al*. (2014).

2.2.3. Radiative properties

The scattering phase function for the porous medium is based on an analytic expression taken from Hendricks and Howell (1996), which was found to yield reliable results when compared to detailed analysis of ceramic foams (Petrasch et al., 2007):

$$\Phi_{ij} = \frac{8}{3\pi}\left(\sin\theta_{ij} - \theta_{ij}\cos\theta_{ij}\right) \tag{8}$$

The volumetric extinction coefficient of the effective homogeneous medium depends on the porosity and mean pore diameter, and is calculated using an expression based on geometric optics analysis of porous ceramics (Hendricks and Howell, 1996):

$$\beta = C\,\frac{1-\phi}{d} \tag{9}$$

where C is a semi-empirical non-dimensional constant, taken as 4.8, which gives the best fit for a variety of materials, according to the work of Kribus *et al*. (2014). The volumetric absorption and scattering coefficients, α_v and σ_v, respectively, can then be calculated by defining the absorptivity of the material, α:

$$\alpha_v = \alpha\beta, \qquad \sigma_v = (1-\alpha)\beta \tag{10}$$

2.3. Absorber performance

The efficiency of the absorber is related to the amount of incident energy that is effectively transferred to the working fluid. The temperature of the exiting fluid is obtained from the energy balance at the end of the absorber:

$$\dot{m} c_p \left(T_{f,ex} - T_f(L)\right) = (1-\phi) h_b \left(T_s(L) - \frac{T_f(L) + T_{f,ex}}{2} \right) \tag{11}$$

After solving the systems of discretized equations with the respective boundary conditions and obtaining the desired convergence, the efficiency of the volumetric receiver can be calculated as the thermal power per unit area transferred to the fluid, divided by the incident flux:

$$\eta = \frac{\dot{m}}{q_{in}} \int_{T_{f,in}}^{T_{f,ex}} c_p(T)\, dT \tag{12}$$

2.4. Convergence

The convergence criterion defined and used in the developed code prescribed a maximum error of 10^{-5} (0.001%) in the thermal balance equations, in the radiative balance equations, and between two consecutive iterations of the radiative source term. This ensures that both solid and fluid phase temperature distributions are fully converged, as well as the radiative transport equations. The number of grid nodes was defined to be high enough so that it does not influence the results and yields temperature distributions with the required accuracy, but without excessive computational time. For all the studies presented in this work, a uniform mesh with 100 grid nodes was used to discretize the governing equations. Regarding the convergence of the algorithm, besides the convergence criterion, a relaxation factor was implemented for the divergence of the radiative heat flux.

3. RESULTS AND DISCUSSION

The main objective of this work was to better understand the influence of parametrical changes on the efficiency of the absorber. The heat transfer model described above was used for this purpose, and a few parameters were analyzed and their influence investigated.

Reference parameters were defined and used in the simulations reported below, with exception of the particular parameter under investigation. The absorptance at the back wall of the receiver was set to zero, meaning that all the radiation reaching the back wall is reflected back into the absorber, and so the only losses occur at the front surface. The thermal conductivity, porosity, pore diameter and absorptance are listed in Tab. 1. These values are typical of ceramic materials used in this type of applications (SiC foams).

Figure 1 shows the temperature of the solid and fluid phases along the absorber. The solid phase temperature is higher at the front surface, since the absorption of radiation is more significant there, causing higher thermal losses due to diffuse emission. This is confirmed by the increase of the radiative heat flux in the backward directions near the front surface, as shown in Fig. 1. The efficiency of this reference case is 73.7%.

Table 1. Reference values for simulation parameters

Absorber mean pore diameter	0.003 m
Absorber porosity	0.85
Absorber thickness	0.020 m
Absorber thermal conductivity	50 W K^{-1} m^{-1}
Absorber surface absorptance	0.85
Air inlet temperature	300 K
Air mass flow rate per unit area	0.55 kg s^{-1} m^{-2}
Incident solar flux	600 kW m^{-2}

Table 2 – Data used for the variable thermal conductivity case study

Absorber mean pore diameter	0.004 m
Absorber porosity	0.92
Absorber thickness	0.020 m
Absorber thermal conductivity (first	100 W K^{-1}
Absorber thermal conductivity (second	5 W K^{-1} m^{-1}
Absorber surface	0.85
Air inlet temperature	300 K
Air mass flow rate per unit area	0.75 kg s^{-1}
Incident solar flux	800 kW m^{-2}

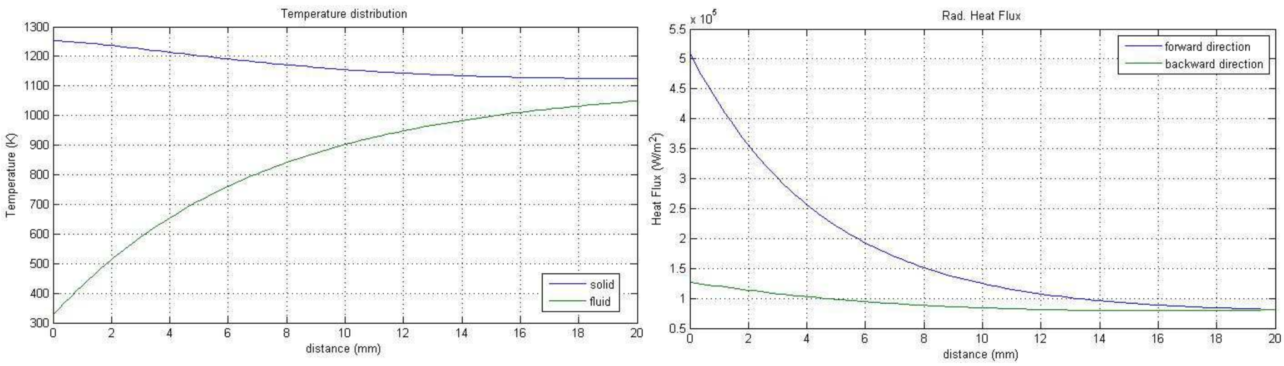

Figure 1. Temperature distributions of solid and fluid phases (left) and radiative heat flux along the forward and backward directions (right) using the reference parameters listed in Tab. 1

3.1. Incident solar radiative flux

The incident solar radiative flux represents the effective available energy that may be transferred to the working fluid. Increasing or decreasing this value can be achieved by having more or fewer heliostats in the field, concentrating the solar radiation at the power tower receiver.

It was expected that higher incident heat fluxes would generate higher temperatures, and therefore better heat transfer conditions and higher efficiencies, but that was not entirely true. Higher solid phase temperatures are in fact obtained, but in the present situation, the air is not able to completely remove the excess of heat, resulting in lower efficiencies. Figure 2 illustrates the previous statement. The difference between the solid and fluid temperatures at the back surface increases with the increase of the incident solar radiative flux, yielding a lower ratio of the converted to the available energy.

3.2. Mass flow rate per unit area

The second parameter evaluated was the mass flow rate, which was expected to have a direct effect on the efficiency of the receiver, as more mass of working fluid results in more capacity to remove heat from the solid absorber. On the other hand, a higher mass flow rate of air requires much more energy to reach the same temperatures, resulting unsurprisingly in lower fluid exit temperatures. Figure 3 shows the increasing efficiency for increasing values of mass flow rate. Values greater than 0.65 kg/s.m^2 were not considered, since lower fluid exit temperatures would be obtained, which is undesirable as far as the global efficiency of the solar power plant is concerned.

3.3. Porosity

The porosity has a large influence on the performance of the absorber, since it affects many aspects of the heat transfer, like the surface to volume ratio, the extinction coefficient of the medium, and even the amount of radiation transmitted to the absorber at the front boundary. The first expectation was that by having less solid material at the front boundary, the absorber would reflect less radiative influx to the ambient. It was further expected that lower radiation would be absorbed/emitted at the front surface, providing more available energy across the receiver for the same incident solar flux. Both expectations were confirmed by the predicted results, shown in Fig. 4, with the extra benefit of the reduced front surface temperature for higher porosity cases, resulting in lower thermal losses to the environment.

3.4. Absorber thermal conductivity

The importance of the thermal conductivity for the design of volumetric ceramic receivers is essential, as it is responsible for the transport of thermal energy by conduction across the solid absorber. Higher thermal conductivities mean lower temperature gradients and more uniform temperature profiles from the front to the back surfaces, while lower thermal conductivities allow for more abrupt variations, as illustrated in Fig. 5.

A lower front surface temperature, in addition to the increase in the difference between the solid and fluid phase temperatures throughout the second half of the absorber, reduces the thermal losses and improves the heat transfer by convection, thus increasing the overall efficiency.

As previously explained, and shown in the results, at the front surface there is a tendency for the temperature gradient of the solid phase to be negative, and for that reason higher thermal conductivities help to keep the temperature distribution lower and more uniform. On the second half of the absorber, however, the temperature gradient may be positive, due to the reduced convection and the increase in radiative heat flux caused by lower porosity, but the increase

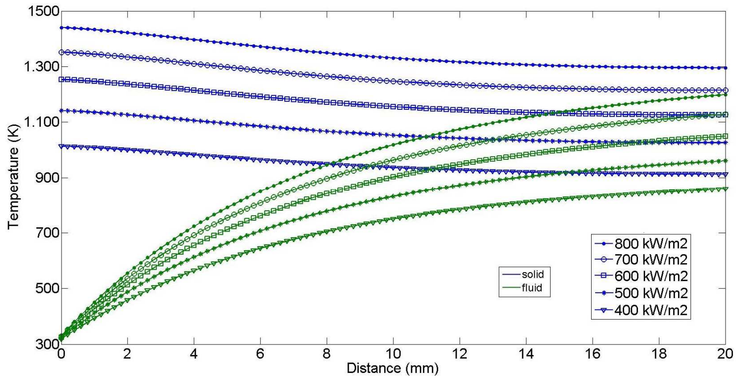

Figure 2. Influence of the incident solar flux on the solid and fluid temperature distributions

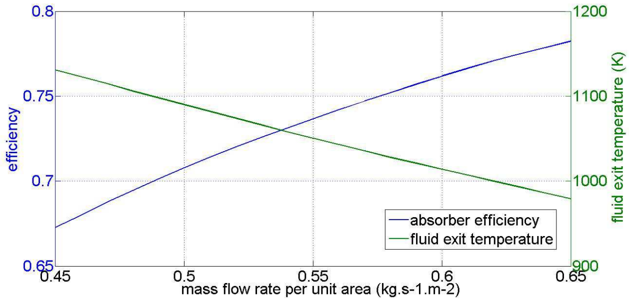

Figure 3. Efficiency and air exit temperature as function of the air mass flow rate per unit area.

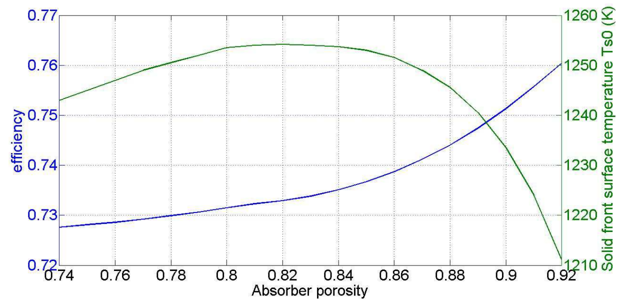

Figure 4. Efficiency and front surface solid temperature variations for different absorber porosity values.

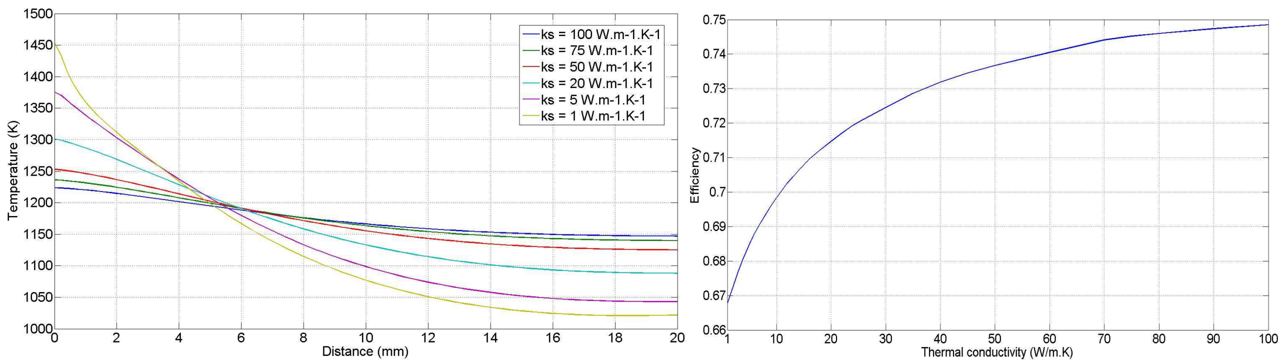

Figure 5. Influence of the absorber thermal conductivity on the solid phase temperature profile (left) and on the efficiency (right).

of temperature is limited by the high thermal conductivity.

Hence, a study was performed using an absorber with two materials in series, of equal length but different thermal conductivities. The thermal conductivity of the material on the first half of the absorber was set to a relatively high value, to avoid high temperature gradients near the front surface, while keeping the porosity as high as possible (within the validity range of the Nusselt correlation), so that more radiative energy can be transmitted to the absorber. The fluid mass flow rate was increased to help the absorption of the extra influx of energy. The values used in this last study are shown in Tab. 2.

This study predicted an efficiency of 80.6% and yields a higher fluid mass flow rate at higher exit temperature (1097 K), as shown in Fig. 6. The most important note goes to the volumetric effect predicted in this case, as the solid temperature at the back of the absorber is 50 K higher than the front surface temperature, due to the lower thermal conductivity of the second half of the absorber.

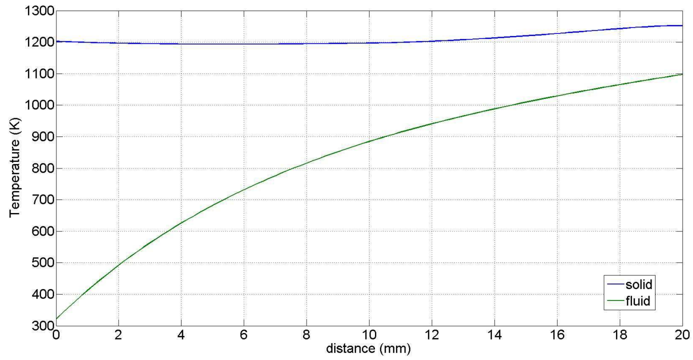

Figure 6. Temperature distributions for the variable thermal conductivity case study

4. CONCLUSIONS

The following conclusions may be drawn from the parametric analysis carried out:

i) Lower solar incident heat fluxes offer better conversion efficiencies, but the lower available energy results in less absorbed energy and lower temperature values as well.

ii) Better efficiencies often require higher mass flow rates or lower solid phase volumetric percentage, resulting in lower fluid exit temperatures.

iii) The larger the fluid to solid ratio, the higher the efficiency. This increase of efficiency is often achieved at the expense of a lower fluid exit temperature, but still a higher fluid mass flow rate provides more capacity to absorb energy by convection, and higher porosity increases the amount of fluid percentage at each cross section, both resulting in better solar to thermal energy conversion.

iv) The thermal conductivity of the solid phase has a large influence on the performance. In the case of uniform thermal conductivity, lower thermal conductivities cause higher solid temperature at the front surface, increasing the thermal losses, while higher thermal conductivities provide a more uniform temperature distribution, resulting in smaller radiative losses and increasing the performance. In the case of an absorber constituted by two materials, with a higher thermal conductivity of the upstream material, the efficiency is significantly improved.

5. ACKNOWLEDGEMENTS

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