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Modeling solar cavity receivers: a review and comparison of natural convection heat loss correlations

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Abstract

As the main difficulty of modeling cavity receivers is determining natural convection heat losses, this paper presents a survey of the different natural convection correlations developed by several authors to model natural convective heat losses from cavity receivers of solar thermal power tower plants. The different correlations studied for modeling convective heat losses are later compared by performing simulations on an implemented cavity receiver. For this work a model of a PS10-like cavity receiver, using Solar Salt as the heat transfer fluid, is implemented in Modelica. The simulations have shown how the results of four out of the five studied correlations do agree, while one of the analyzed correlations clearly overestimates convective heat losses for the simulations performed.

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Keywords: Concentrated solar power; Cavity receiver; Natural convection heat loss correlations

1. Introduction

Since the first commercial solar thermal power tower plant, PS10 in Sanlucar La Mayor, Spain, was connected to the grid in 2007, the interest in this technology has largely increased. Currently, solar tower technologies account for 457 MW operating worldwide, 210 MW under construction and about 6000 MW planned [1].

In CSP technologies, as in any other technology, any design needs to be tested through simulation in order to be validated theoretically. This is the reason why there is a latent need to develop accurate models for solar thermal power tower plants and particularly, there is a great interest in developing simulation models for cavity receivers, which have proven to have larger efficiencies than external receivers. For modeling these receivers, the most critical part is the accurate modeling of the heat losses, which determine its efficiency. However, modeling the heat losses

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of a cavity receiver can be a challenge considering the complexity of the heat transfer mechanisms and the infinite number of possible geometric configurations. Already proposed solar cavity receiver models can be subdivided into two groups, (i) three dimensional numerical models with high spatial resolution, namely CFD models, and (ii) numerical as well as empirical models of much lower spatial resolution. Although CFD models can analyze the influence of the geometry in the heat loss process with a very accurate and realistic approach, the computational cost of CFD studies is so high that these models cannot be applied to long-term simulations of complete solar thermal power tower plants.

The underlying purpose of this work is the development of receiver models for performing long-term transient simulations, in particular focusing on the evaluation of convective heat loss correlations. Convective heat losses are defined as the heat flow rate flowing out of the cavity aperture as the resulting process of heating the air inside the cavity or air streams due to wind loads and can be calculated by using natural convection and forced convection heat transfer expressions. These expressions are typically empirical correlations adjusted by performing experiments under certain conditions to specific geometries.

In this paper, an extensive survey of the specific related bibliography as well as the comparison of the proposed expressions for calculating the convective thermal losses in solar cavity receivers is presented. This work has been carried out in order to find an appropriate set of correlations to develop simulation models for cavity receivers with a focus on the compromise between model accuracy and computational cost, so as to include the developed models in an energy yield simulation program on system level. In order to compare the identified correlations, a flexible model for a cavity receivers has been implemented. The model, a PS10-like geometry cavity receiver using molten salts as the heat transfer fluid, has been simulated under the same boundary conditions using each of the identified correlations for calculating the convective heat losses. The heat transfer fluid outlet temperature is fixed by means of an adaptive controller; this way the active surfaces are at the same temperature for all the simulations, so the differences in the receiver performance are only caused by the natural convection correlation used in each case.

2. Modeling convection heat losses

Convection heat losses play a major role in the performance of a cavity receiver. In the last 20 years several correlations for the Nusselt number have been developed by different authors; these expressions have been later used to model both convective heat losses mechanisms: natural convection heat losses and forced convection heat losses. However, the focus of this paper is on natural convection heat losses, being the modeling of forced convection heat losses out of the present scope.

The correlations provided are generally based in the Nusselt number, Nu, that allows calculating the heat transfer coefficient, h, needed to calculate the corresponding convection heat losses, $\dot{Q}_{conv}$ as shown in Eqs. (1) and (2).

$$\dot{Q}_{conv} = hA_s(T_s - T_{amb}) \tag{1}$$

$$Nu = \frac{hL_c}{k} \tag{2}$$

Where:

- $\boldsymbol{T_{amb}}$ is the ambient temperature,
- $\boldsymbol{T_s}$ is the surface temperature,
- $\boldsymbol{L_c}$ is the characteristic length,
- k is the thermal conductivity, and
- $\boldsymbol{A_s}$ is the surface area.

The correlations presented in this paper are used to determine either a global heat transfer coefficient for the cavity or a specific heat transfer coefficient for every surface inside the cavity. Eq. (1) and Eq. (2) are applied to every surface inside the cavity if the correlation is able to estimate a specific heat transfer coefficient for every surface or to a single surface at a temperature equal to the cavity average temperature if only a global heat transfer coefficient can be estimated by the correlation provided.

In the following sections, a review of Nusselt number correlations provided by different authors for the natural convection heat losses from solar cavity receivers is conducted. The correlations presented by Clausing in 1983 [2], Sieber and Kraabel in 1984 [3], Clausing et al. in 1987 [4], Leibfried and Otjohann in 1995 [5] and Paitoonsurikarn and Lovegrove in 2006 [6] are considered to be the most relevant ones, and thus hereafter analyzed.

2.1. Clausing, 1983

Clausing [2] presented a correlation for the Nusselt number that takes into account the surface orientation, the surface temperature and the ambient temperature, this correlation allows calculating the appropriate heat transfer coefficient for each surface.

The cavity shown in Fig. 1 is the one studied by the author. It is divided into two different zones, a stagnant zone, and a convective zone. The density of the air entering the cavity is typically a factor of three or four higher than the density of the air at the temperature of the refractory surfaces in the upper portion of the cavity. Hence, if the aperture is in the lower portion of the cavity, the air inside the cavity is expected to be stratified and relatively stagnant in the upper region. For this reason, the interior volume is divided into a Convective Zone, Zone I, and a Stagnant Zone, Zone 2 [2]. The stagnant zone is the part of the receiver where the air stays still, the convective zone is the zone where the majority of the convection heat losses occur due to the velocity induced in the air by the difference between the ambient temperature and the surfaces temperatures, called buoyant velocity.

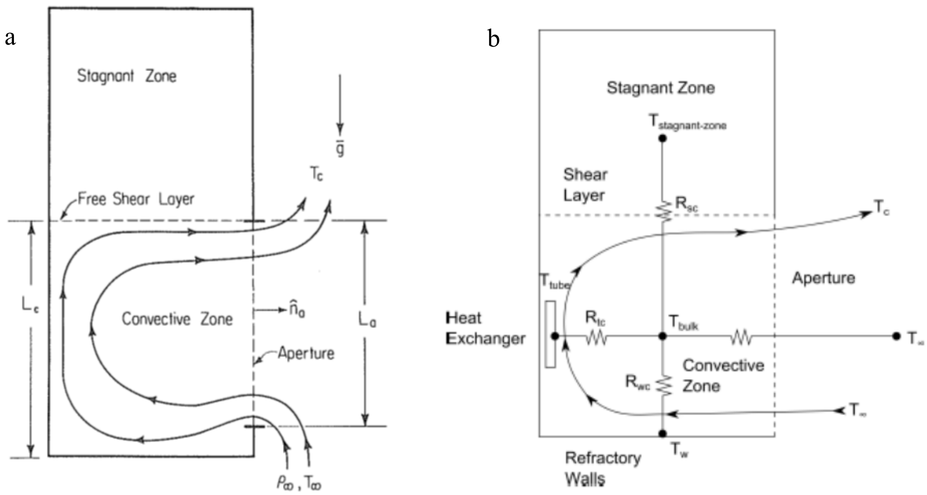

Fig. 1. (a) Cavity receiver studied by [2]; (b) Network representation used by [2].

In Fig. 1 (a), T_c is the temperature of the air leaving the cavity, T_b is the bulk air temperature, L_a is the length of the aperture and L_c is the length of the convective zone. The author expressed the natural convection heat losses flowing out through the aperture with the expression shown in Eq. (3).

$$\dot{Q}_a = \rho_{amb} v_a A_a c_p (T_c - T_{amb}) \quad (3)$$

With v_a being an effective flow velocity calculated as the quadratic mean of wind velocity and buoyancy induced velocity, ρ_{amb} the density, c_p is the specific heat of the ambient air and A_a is the aperture area through which mass flows into the cavity (assumed to be half of the total aperture area). The involved air velocities are estimated through equations Eq. (4) and Eq. (5).

$$v_b = [g\beta(T_c - T_{amb})L_a]^{1/2} \quad (4)$$

$$v_a = 0.5\left[{v_b}^2 + \left(\frac{v_{wind}}{2}\right)^2\right]^{0.5} \quad (5)$$

Where, v_b is defined as the buoyancy induced velocity, an air mass flow driven by the high surface temperatures inside the cavity.

The author proposed solving the net of resistances represented in Fig. 1 (b).This network leads to the Eq. (6):

$$\dot{Q}_{nc} = \frac{T_t - T_b}{R_{tc}} + \frac{T_w - T_b}{R_{wc}} + \frac{T_s - T_b}{R_{sc}} \quad (6)$$

Where $R = 1/hA$.

The heat transfer coefficient is then calculated thanks to the Nusselt number correlation provided by Eq. (7).

$$Nu = 0.082Ra^{\frac{1}{3}}\left[-0.9 + 2.4\frac{T_s}{T_{amb}} - 0.5\left(\frac{T_s}{T_{amb}}\right)^2\right]z(Z_w) \tag{7}$$

The Nusselt number is determined as a function of Rayleigh number, wall and ambient temperature and the surface orientation $z(Z_w)$, where Z_w is the angle between the z-axis and the normal vector to the surface. The surface orientation is calculated according to Eq. (8).

$$z(Z_w) = \begin{cases} 1 & if\ 0 < Z_w < 135deg \\ \frac{2}{3}\left[1 + \frac{sin(Z_w)}{\sqrt{2}}\right] & if\ \ Z_w > 135 \\ \frac{2}{3} & downward\ facing\ heated\ surface \end{cases} \tag{8}$$

The surface heat transfer area between stagnant zone and convective zone is $A_{stag} = 0.3A_{ceiling}$.

According to the authors, this model is valid for $Ra > 1.6 \times 10^9$ and $1 < T_s/T_{amb} < 2.6$. For temperatures ratios greater than 2.6 its value should be limited to 2 [2] until data in this region is available. The surface temperatures for a cavity receiver might achieve values greater than 600ºC, so the temperatures ratio, evaluated in K with an ambient temperature of 20ºC, might be around 2, within the validity range, the Ra number during operation is also within the validity range ($3.6\cdot10^{12}$-$5.3\cdot10^{12}$ for the active panels).

2.2. Siebers and Kraabel 1984

Siebers and Kraabel [3] determined a global heat transfer coefficient for the cavity receiver as a function of the Grashoff number, the cavity average temperature and the ambient temperature, this coefficient is weighted by the areas of the cavity to calculate the total heat transfer coefficient. The authors studied the cubical receiver in Fig. 2 and provided a global heat transfer coefficient for the cavity.

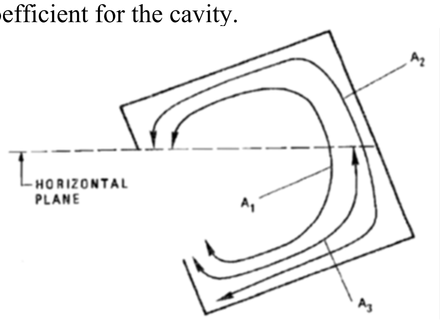

Fig. 2. *Cavity studied [3].*

In this model a heat transfer coefficient expression is provided for air at normal ambient temperatures without a previous calculation of the Nusselt number, $h_{nc,0}$, Eq. (9).To account for partial covering of the cavity opening such as a lower and upper lip Eq. (10) is used.

$$h_{nc,0} = 0.81(T_{cav} - T_{amb})^{0.426} \tag{9}$$

$$h_{nc} = h_{nc,0}\left(\frac{A_1}{A_2}\right)\left(\frac{A_3}{A_1}\right)^{0.63} \tag{10}$$

Where T_{cav} is the average cavity temperature, $A_1$is the total cavity surface, A_2 the cavity surface area excluding the area of the lower lip and A_3 the cavity surface area excluding the area of the upper lip as shown in Fig. 2. No validity range is provided for this correlation.

2.3. Clausing 1987

In Clausing et al. [4], the same cavity receiver than in [2] is analyzed, this time providing a global Nusselt number for the cavity receiver. A global heat transfer coefficient for the cavity receiver in terms of three parameters is calculated. The first parameter, g, represents the natural convection correlation from the internal cavity surfaces to the surroundings for constant properties, the second one, f, is the quantity which accounts for variable property influences because of the difference of wall and ambient temperature and the last parameter, b, accounting for the effects that occur when the temperature of the air inside the cavity is significantly different than the ambient temperature.

$$Nu = gfb \tag{11}$$

Table 1. "f" and "g" parameters as a function of the flow regime.

$Ra < 3.2 \times 10^8 = Ra_l$	$g = 0.63Ra^{1/4}$ $f_l = 1$	(12)
$Ra_l < Ra < 1.6 \times 10^9 = Ra_{turb}$	$g = 0.63Ra^{1/4}$ $f_t = (f_{turb} - 1)\left[\left(Ra^{\frac{1}{3}} - Ra_l^{\frac{1}{3}}\right)\Big/\left(Ra_{turb}^{\frac{1}{3}} - Ra_l^{\frac{1}{3}}\right)\right] + 1$	(13)
$Ra > Ra_{turb}$	$g = 0.108Ra^{1/3}$ $f_{turb} = 0.2524 + 0.9163\frac{T_w}{T_\infty} - 0.1663\left(\frac{T_w}{T_\infty}\right)^2$	(14)

$$b = 1 - 1.57\left[\frac{gfb\left(\frac{k_f}{k_{amb}}\right)}{\left(Ra_{amb}Pr_{amb}\frac{L_a}{L_c}\right)^{\frac{1}{2}}\left(\frac{A_a}{A_{convective\ zone}}\right)}\right]^{\frac{2}{3}} \tag{15}$$

In this expression, properties must be evaluated at the film temperature. The characteristic length is the aperture height plus half of the total cavity height $L_c = L_a + \frac{L}{2}$.

$A_{convective\ zone}$ is constituted by all surfaces that are not in the stagnant zone and the aperture area. According to the authors, the provided correlations are valid for $1 < T_s/T_{amb} < 3$, $L^2/18 \leq A_a \leq L^2$ and $3 \times 10^7 < Ra < 3 \times 10^{10}$.

2.4. Leibfried and Otjohann 1995

Leibfried and Otjohann [5] studied the flow configuration in a hemispherical tilted cavity receiver, determining a global Nusselt correlation accounting for the ambient temperature, the cavity average temperature and the orientation of the cavity. According to the authors, heat transfer mainly occurs in the boundary layer, although only buoyancy driven flow is considered, in some regions of the cavity forced convection patterns are observed.

The Nusselt number derived takes into account the orientation of the cavity $h(\overline{\theta})$, the total area of the cavity, A_{cav} , the area of the aperture, A_a, the Grashoff number and the ratio of the cavity average temperature and the ambient temperature, see Eq. (16) .

$$Nu = 0.106Gr^{\frac{1}{3}}\left(\frac{T_w}{T_\infty}\right)^{0.18}\left(\frac{4.256A_a}{A_{cav}}\right)^s h(\overline{\theta}) \tag{16}$$

$$s = 0.56 - 1.01\left(\frac{A_a}{A_{cav}}\right)^{\frac{1}{2}} \tag{17}$$

$$h(\overline{\theta}) = \frac{1}{h_0}\left[1 - cos\left(\overline{\theta}^{\ 0.85}\pi\right)\right] \tag{18}$$

Where $\overline{\theta}$ is defined as $\overline{\theta} = \left(\theta - \theta_{stag,eff}\right)/\ (\theta_{max} - \ \theta_{stag,eff})$, being θ_{stag} the angle up to which the heated part of the cavity is situated in the stagnant zone, generally 90 deg. The expression for $\theta_{stag,eff}$ is found in Eq. (19).

$$\theta_{stag,eff} = \theta_{stag} + \frac{\left(90\ deg - \theta_{stag}\right)\left(\theta_{stag} - \theta\right)}{\theta_{stag}} \tag{19}$$

Where θ is the tilt angle of the cavity, 90 *deg* when the cavity is upward facing and 0 deg when the normal vector to the aperture of the cavity is horizontal.

$$h_0 = 1 - cos(\overline{\theta}(\theta = 0)\ ^{0.85}\pi) \tag{20}$$

The authors claim that these expressions are valid as long as $A_{aperture}/A_{cavity} < 0.2$, the aperture area of the receiver studied is 126.5 m^2 and the total area of the cavity is 563.88 m^2,so the ratio is 0.224, slightly greater than the provided limit.

2.5. Paitoonsurikarn and Lovegrove 2006

Paitoonsurikarn and Lovegrove [6] established a Nusselt correlation depending on the Grashoff and the Prandtl number for solar dish cavity receivers. The characteristic length is calculated as a function of the cavity geometry. Even though it was developed for solar dish cavity receivers, it was also tested in large cavities, providing accurate results. The authors claim that for the case of a 400 m^2 dish receiver the correlation correlates as high as 70% of the data from within ±20%, and up to 100% coverage within ±50%.

Eq. (21) shows the Nusselt correlation provided by the authors in [6]. Where L_s is the cavity length scale and it is calculated according to Eq. (22).

$$Nu_L = 0.0196 Ra_L^{0.41} Pr^{0.13} = \frac{hL_s}{k} \tag{21}$$

$$L_s = \left| \sum_{i=1}^{3} a_i cos(\emptyset + \Psi_i)^{b_i} L_i \right| \tag{22}$$

In this expressions, a_i, Ψ_i, b_i are parameters given by the authors based on empirical adjustments. The proposed values are reproduced in Table 2.

Table 2. Constants for the evaluation of the ensemble cavity length scale Ls [6].

i	a_i	b_i	Ψ_i
1	4.08	5.41	-0.11
2	-1.17	7.17	-0.3
3	0.07	1.99	-0.08

The meaning of the different dimensions L_i is specified in Fig. 3.

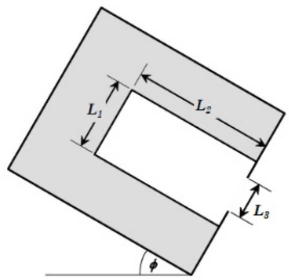

Fig. 3. Cavity geometrical parameters used in the definition of the ensemble cavity length scale L_S [6].

In this model, the properties are evaluated at film temperature, the mean temperature of the cavity average temperature and the ambient temperature, $T_f = (T_{cav} + T_{amb})/2$.

3. Simulations

The receiver model has been implemented in Modelica as a part of a master thesis [7] and allows a flexible geometry definition. To compare the results provided by the different natural convection correlations, the geometry of the receiver is fixed according to Fig. 4 (a).

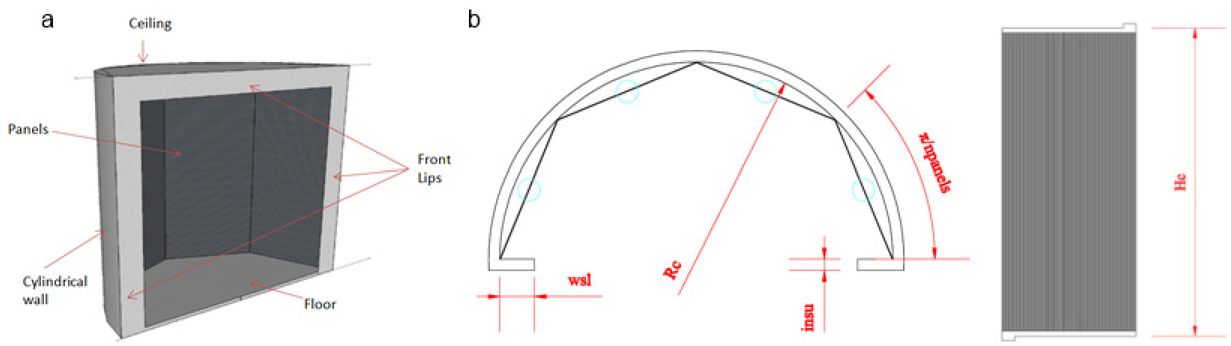

Fig. 4. (a) PS10-like cavity receiver geometry; (b) Schematic of the receiver and its geometry,

Eight parameters are taken as inputs by the model to completely define its geometry, as shown by the graphical representation in Fig. 4 (b).

- R_c: cavity radius, which defines the inner radius of the cavity
- H_c: receiver height, which is the internal height of the cavity and thus the height of the active surfaces
- e_{insu}: width of the insulation surfaces, this parameter is the same for all the passive surfaces already mentioned
- w_{sl} and w_{ul}: width of the side lips and upper lip respectively
- n_{tubes}: number of tubes inside each panel

The parameters taken to define the geometry of the receiver are shown in Table 3 (a); these dimensions are similar to the ones of the PS10 cavity receiver. The fluid used as HTF is molten salts typically a mixture of 60% $NaNO_3$ and 40% KNO_3 (weight percent).

Table 3. (a) PS-10 like cavity receiver geometric parameters; (b) Simulation Conditions

(a) Geometric parameters		
R_c	7	m
H_c	12	m
e_{insu}	0.4	m
w_{sl}	1.25	m
w_{ul}	1	m
n_{pan}	4	
n_{tubes}	300	
e_{pipe}	0.0025	m

(b) Simulation Parameters		
DNI	800	W/ m2
Inlet Temperature	280	ºC
Outlet temperature	565	ºC
Ambient temperature	20	ºC
Wind speed	0	m/s
Wind direction	0	rad
Relative Humidity	15	%
Atmospheric Pressure	101325	Pa
Solar Field Area	100500	m2
Focused Radiation	5,67E+07	W

The panels within the receiver, each of them consisting of 300 tubes, are connected in series, as shown in Fig. 5.

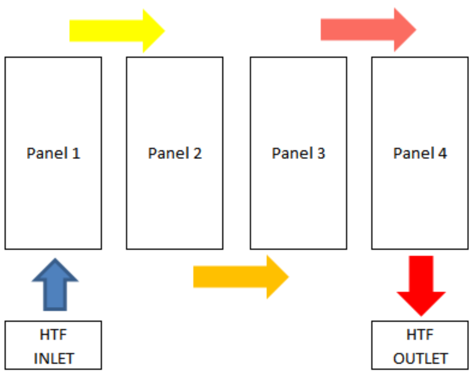

Fig. 5. HTF flow configuration through the panels.

The system used for the simulations consisted of a fluid source (at constant temperature) and a fluid sink connected to the inlet and outlet of the modeled receiver; the adaptive controller and a simple model representing the solar radiation focused on the receiver by a heliostat field.

Under the constant conditions stated in Table 3 (b), simulations for the different natural convection models were conducted. The simulation conditions were fixed in order to perform the comparisons for the steady-state conditions.

3.1. Comparisons between the natural convection models

For the fixed operating conditions, in Table 3 (b), the correlations studied in Section 2 to model natural convection heat losses are compared. The performance of the cavity receiver with the different convection models analyzed is provided in Table 4. In particular, the power absorbed by the heat transfer fluid, the efficiency of the receiver and the total heat losses are the results analyzed. The efficiency is defined as the ratio of the total power absorbed by the heat transfer fluid to the total solar power incident on the receiver.

Table 4. Performance of the cavity receiver depending on the natural convection model used.

MODEL	GENERAL PERFORMANCE		%Heat Loss	
Clausing 1983	Power absorbed	5,177E+07		
	Efficiency	91,34%		
	Heat Losses	5,338E+06	Convection	27,39%
			Conduction	10,24%
			Radiation	62,37%
Clausing 1987	Power absorbed	5,118E+07		
	Efficiency	90,31%		
	Heat Losses	5,591E+06	Convection	35,57%
			Conduction	12,41%
			Radiation	52,02%
Siebers and Kraabel	Power absorbed	5,111E+07		
	Efficiency	90,17%		
	Heat Losses	5,705E+06	Convection	34,08%
			Conduction	12,62%
			Radiation	53,30%
Paitoonsurikarn and Lovegrove	Power absorbed	4,833E+07		
	Efficiency	85,28%		
	Heat Losses	7,848E+06	Convection	63,42%
			Conduction	3,50%
			Radiation	33,08%
Leibfried and Ortjohann	Power absorbed	5,140E+07		
	Efficiency	90,69%		
	Heat Losses	5,496E+06	Convection	30,23%
			Conduction	13,38%
			Radiation	56,40%

The total heat losses are defined as the total heat flow rate flowing out of the cavity. This means that they account for the radiation exiting the cavity through the aperture (solar and thermal radiation), the conduction heat losses through the insulation layer and the convection heat losses due to natural convection inside the cavity and forced convection. The heat flow rate lost by each of these mechanisms is also detailed.

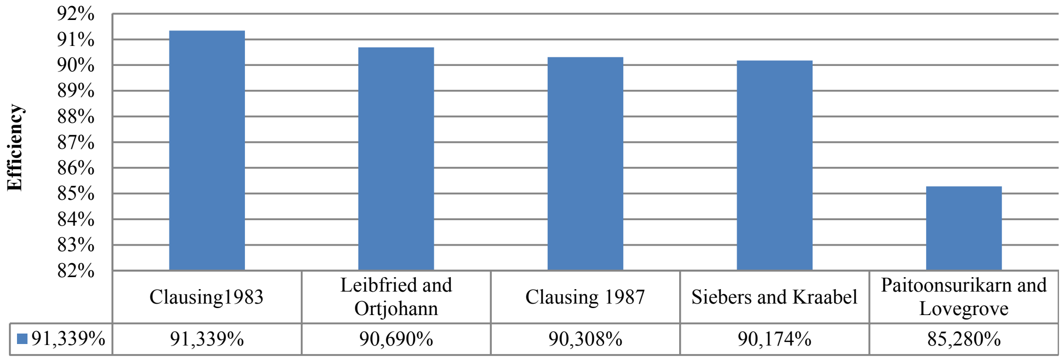

Fig. 6. Efficiencies for the different correlations.

Fig. 6 shows that the application of the correlation for natural convection provided by Clausing in 1983 results in the greatest efficiency: 91.339%. The model introduced by Leibfried and Ortjohann, Clausing in 1987 and Siebers and Kraabel in 1984 results in almost identical performance characteristics; 90.690%, 90.308% and 90.174% respectively, of the total incident radiation on the receiver is transferred to the heat transfer fluid. The correlation provided by Paitoonsurikarn and Lovegrove for Solar Dish cavity receivers, with a smaller area than cavity receivers in central tower systems and lower surface temperatures, provides the lowest power rate transferred to the heat transfer fluid; 85.280% of the incident radiation is absorbed by the heat transfer fluid. The results presented here agree quite well with the ones presented by Teichel [8].

It has to be said, that there is a big uncertainty associated with the correlations provided by the different authors. Clausing claims that with the correlation provided in 1987 the maximum deviation of any data point of his experiment, a cavity in a cryogenic wind tunnel, is 20%. When using this expression in other cavities with different geometries the deviation might be greater. Clausing also claims that for the correlation recommended in 1983, the average absolute deviation for the 119 data points is 1%. However, the uncertainty is higher in this case, since some of the values of the Rayleigh number are out of the validity range. Paitoonsurikarn and Lovegrove states that their correlation, in the worst case tested, correlates only 40% of the model receiver data within ±20% and up to approximately 80% of the data within ±50%. For better cases, the correlation correlates as high as 70% of the data within ±20%, and up to 100% coverage within ±50%. The rest of the correlations also present great uncertainties when they are used for cavities with geometries different from those as studied by the authors, but no explicit values are provided.

With such a big uncertainty associated with the correlations used to model the convection heat losses all the models can be considered to be valid except the one developed for solar dish cavity receivers. Even though the authors [9] claim that their correlation is valid for larger cavity receivers, the results do not agree with the ones of other correlations.

There is only one model that provides the heat losses on every surface of the cavity receiver; Clausing 1983. The rest of the models provide a total heat transfer coefficient for the cavity receiver and thus later distribution of the convective heat losses has to be done for all the surfaces inside the cavity so that the heat losses are proportional to the surface area. This heat transfer coefficient is calculated for a cavity mean temperature calculated as an area averaged temperature. It has to be considered that the panels are cooled by the heat transfer fluid, thus the temperature of their surface is kept relatively low, at values ranging from 290-580ºC. However, the passive surfaces are not cooled, in such way the temperatures of these surfaces can be greater than the temperatures of the coolest active ones. As the natural convective heat losses of any surface can be treated separately, the heat losses of the panel surfaces are kept at their real values. In contrast, for the rest of the models as the temperature of the passive surfaces can increase the cavity average temperature, the total convective heat losses increase too. When these heat losses are distributed to every surface by weighting the area, the natural convection heat losses from the panel surfaces are greater than their real natural convection heat losses. In such way the efficiency of the receiver can be reduced.

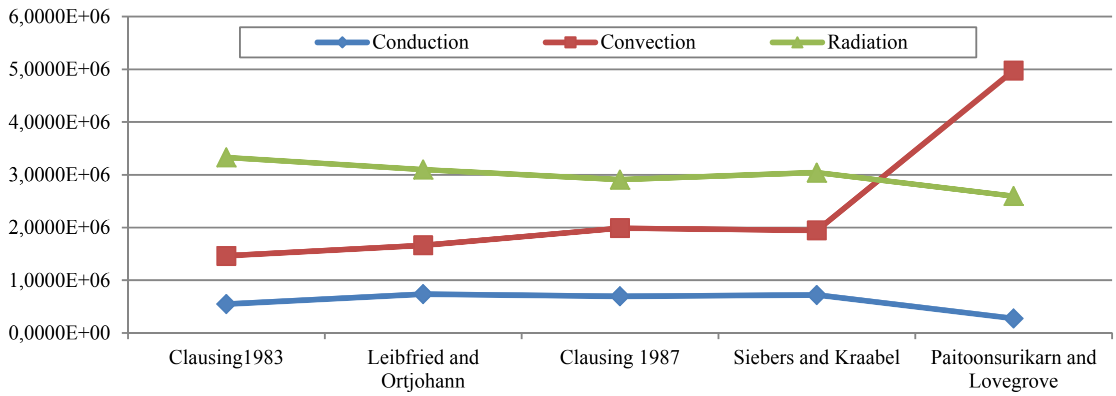

Fig. 7. Comparisons of the heat transfer mechanisms for the different natural convection models analyzed.

As it is shown in Fig. 7 conduction and radiation heat losses are quite constant for all the convection models, as expected. This is because the radiosity method implemented for the simulation is not changed, and neither the characteristics of the insulation layers are. The slight variability observed is due to the variation in the natural convection heat loss model.

To sum up, even though four of the analyzed correlations provide similar results, and thus the four of them seem to be valid for the cavity receiver simulated in this work, within the uncertainty declared in the papers, the method proposed by Clausing in [2] is considered to be the best choice if a detailed performance model has to be developed for a cavity receiver.

4. Conclusions

In this paper a deep review of the existing natural convection correlations for solar cavity receivers for central receiver systems was performed. These correlations were implemented in an existing model comparing the results obtained. It was found that the correlations introduced by Leibfried and Ortjohann [5], Clausing et al. [4], and Siebers and Kraabel [3] results in almost identical performance characteristics. The correlation presented by Clausing [2] resulted in similar efficiency to the previous 3 models while the one by Paitoonsurikarn and Lovegrove [6] seems to overestimate the convective heat losses for the cavity studied. Finally Clausing [2] is chosen as the preferred correlation due to the possibility of calculating individual heat losses for each surface.

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