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Cavity Losses Estimation in CSP Applications

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Abstract. Estimations of convection and radiation cavity losses in two common CSP applications have been analyzed; a cavity in a solar tower plant for high temperature (800 K) and in a down facing cavity in a Fresnel configuration for medium temperature (350 K) applications. An analysis regarding the effect of the configuration, geometry and the presence of wind has been also carried out.

INTRODUCTION

Heat losses can be the bottleneck in any CSP application. A description has been carried out, involving how to estimate convection and radiation cavity losses in terms of appropriate correlations [1,2], validate them by comparison to full 3D CFD calculations and relate to previous results. The goal is to develop a real time computing scheme to be included in a geometric optimization tool that will be described elsewhere. Additionally, the numerical model provides a way to analyze the effect of wind intensity and direction. The proposed method is then applied to a cavity in a solar tower plant (800 K) and to a down-facing concept cavity in a CSP Fresnel configuration for medium temperature applications (350 K). For this last application, two secondary reflector geometries have been studied in order to estimate convection cavity losses.

METHODOLOGY

The estimation of the convective losses depends essentially on the ability to simulate the heating of the air inside the cavity and estimate the transfer of mass and energy across the aperture by the combined influence of flow over the aperture and buoyancy. In this process, a part of the cavity experiences stagnation condition and the boundary separating this area from the remaining of the cavity can be approximated by a horizontal plane passing through the topmost point of the cavity aperture. It is called the stagnation zone boundary and can be modeled as an adiabatic surface. The zone below is usually called convective zone where strong convective currents are observed. This description is confirmed by our CFD simulations in the absence of wind. Unless the aperture is in a horizontal plane (facing-down cavity), the experimental and numerical evidence shows that natural convection plays a significant role in the heat loss process

Based on extensive experimental results with different geometries and conditions, universality properties and analogies, a functional form (correlation) for a relevant quantity is proposed in [2]. This correlation has been used in this specific setup by comparing with 2D and 3D CFD calculations with ANSYS-Fluent. The natural convection and radiation losses from open cavities for the convective losses of the Nusselt number taking the aperture diameter as characteristic dimension is given [2] by:

$$Nu = 0.122(Ra)^{0.31}\left(\frac{T_w}{T_a}\right)^{0.066}(1+cos\theta)^{0.38} \tag{1}$$

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where Ra is the Rayleigh number, T_a the ambient temperature, T_w the wall temperature and θ the angle of the aperture. The total heat loss due to convection is given by Q=h A $(T_w - T_a)$ were h is the convective heat transfer coefficient, with A the area of the aperture and k the air conductivity.

In [1] another correlation is obtained for the heat losses due to radiation. They are independent of the inclination and proportional to the area of the aperture of the cavity and the wall temperature to the fourth power. The conclusion is that the designer has no other way to reduce the radiation losses than minimizing the aperture area or try to keep the wall temperature as low as possible.

Full 3D calculation of the convective heat losses in the cavity has been estimated with ANSYS-Fluent. Two models has been used for solar tower cavity: a 3D coupled CHT (Conjugate Heat Transfer) thermal-fluid model for HTF(Heat Transfer Fluid) and wall temperatures calculation, and a 3D coupled CHT thermal-fluid model for ambient air. In **FIGURE 1** is shown the used mesh for the last model mentioned, where air ambient is inside the control volume. In order to reduce complexity, the receiver panel is simplified in this second model as a flat surface and made with detail in the first model, where obtained film coefficient of the flat surface is then applied on every external surface of the receiver panel.

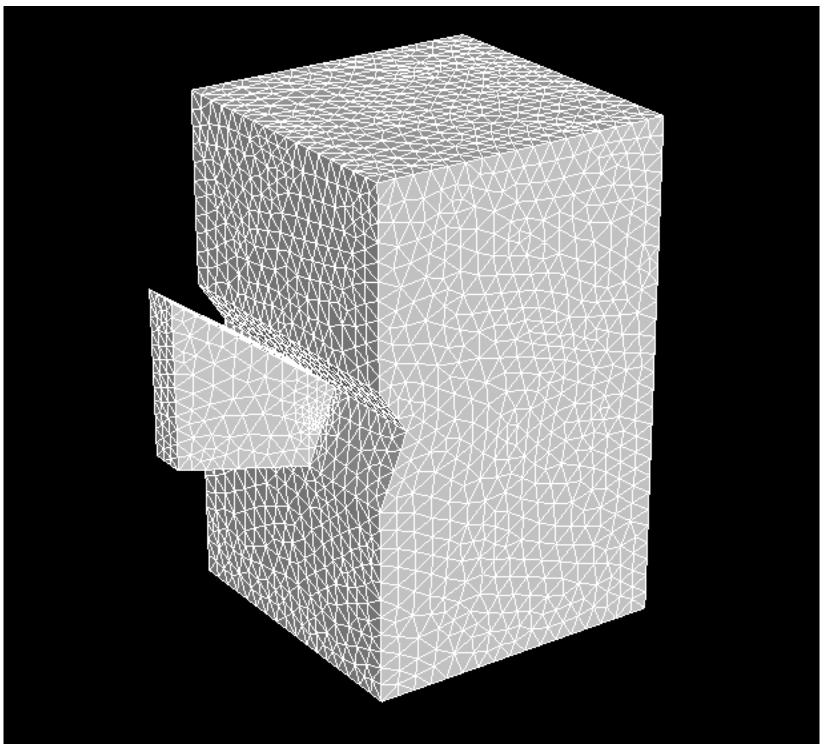

FIGURE 1. Mesh for control volume involving external ambient air, composed by the interior of the cavity and a large ambient parallelepiped (10x7x7 m).

The problem is solved by iterating heat transfer coefficient of every wall of the cavity and their wall temperatures, between the mentioned models. Further details of the first model cannot be disclosed due to confidentiality constraints. The second model is solved by means of a k-epsilon turbulent model with enhanced wall treatment. In this matter, result comparison has been done between k-epsilon and k-omega turbulent models and differences of 15% in terms of energy losses have been found, therefore, comparison with experiments is highly needed. The boundary conditions are fixed temperature at the interior walls and adiabatic walls for the tower. For the floor, sky and outside walls, a pressure outlet boundary condition with constant reduced pressure has been found to be the best option, and a non-stationary calculation with refined mesh at the entrance of the cavity is also needed. The convergence criterion is the formation of the convection cell and the global balance of power between the cavity and the ambient.

For heat losses calculation in Fresnel cavity, a 3D thermal-fluid model and a 2D fluid model have been developed. In the 3D model, **FIGURES 2,3**, saturated water is used as HTF and is heated from 198 to 200ºC inside of a simplified evacuated receiver. Radiation enclosures have been declared between the Absorber Tube (AbTu) and the Glass Seal (GlSe), between the GlSe and the inner surfaces of the Secondary Reflector (SeRe), and radiation to the ambient from the outer surfaces of the SeRe. Incident radiation profiles have been imposed at the outer surface of the AbTu and inner surfaces of the SeRe, these profiles have been obtained by modeling the entire Fresnel plant in an optical software based on Montecarlo method. Convection boundary conditions have been established at the GlSe, inner and outer surfaces of the SeRe. These surfaces have been divided in different parts according to their geometry. Two types of SeRe's have been studied, shown in **FIGURE 4**. For cylindrical reflector geometry, parameter L_3 has been modified in order to study surface convection heat transfer coefficients of every surface. For trapezoidal reflector geometry, parameters L_2 and h have been modified. The problem has been solved by iterating these coefficients and exchanging solid temperatures between the models described.

2D fluid problem is solved by means of a k-epsilon turbulent model with enhanced wall treatment in order to achieve better results near walls [7]. For every value of the parameters exposed, an unique set of incident radiation profiles has been used on each geometry, i.e. the selected range does not change significantly the radiation profiles and remain constant for each geometry.

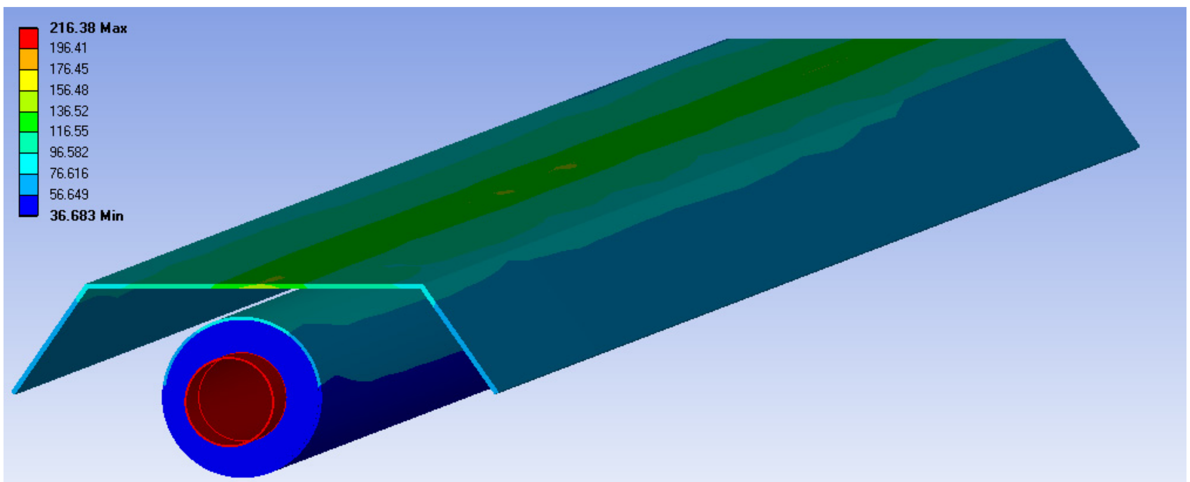

FIGURE 2. Temperature field (ºC) for 3D thermal-fluid model with trapezoidal geometry. AbTu reaches temperatures above HTF temperature (210ºC) due to the incident radiation.

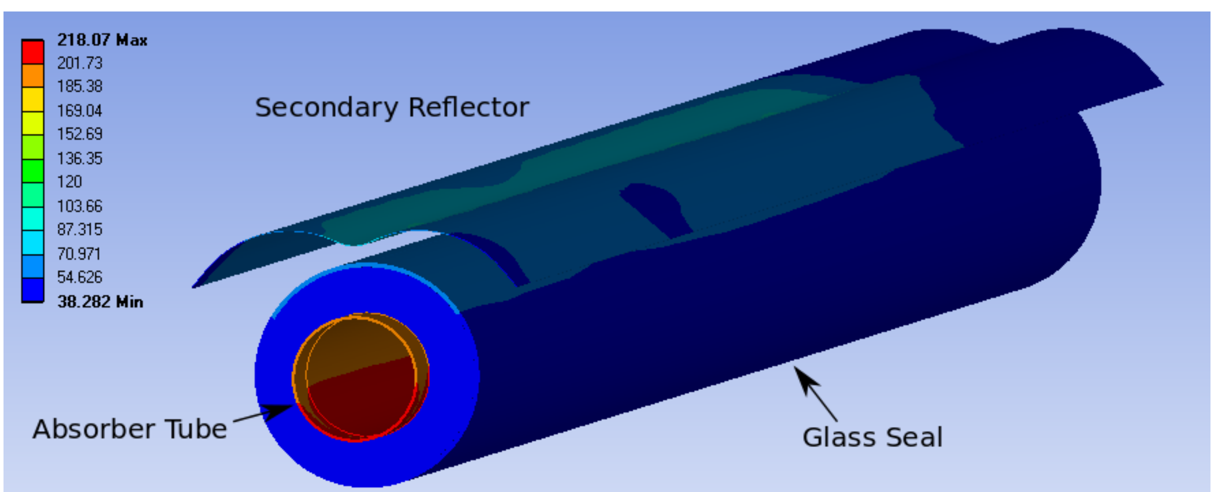

FIGURE 3. Temperature field (ºC) for 3D thermal-fluid model with cylindrical geometry. AbTu reaches temperatures above HTF temperature (210ºC) due to the incident radiation.

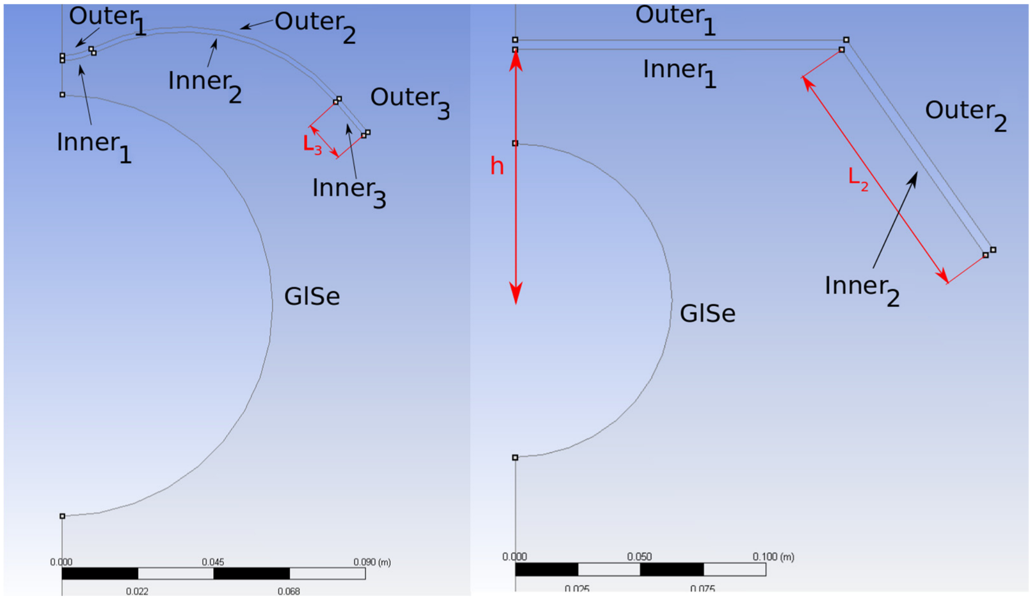

FIGURE 4. Geometry description and parameters to be changed for cylindrical geometry (left) and trapezoidal geometry (right). L_3 represents the length of lines Inner_3 and Outer_3 without any more changes, h is the distance between the center of the receiver and the horizontal surface of the SeRe, and L_2 is assigned to the length of lines Inner_2 and Outer_2.

RESULTS FOR SOLAR TOWER PLANTS

In CERSOL project [3], modelization and optimization of the geometry of a cavity for a Solar Tower Plant have been developed. In **FIGURE 5** the best fitting to the CFD 2D (circles) and 3D (squares) calculations for the convection losses is shown. The details will be published elsewhere but to give an estimation of the size of the cavity, the diameter of the aperture was 2 m and the length of the cavity was 4 m. From this investigation, it has been concluded that under operating conditions the wall temperature of the cavity should be below 850 K to get total losses under the 30 % threshold. Wind load has also been modeled with appropriate boundary conditions. The study shows that even a moderate lateral wind can dramatically modify the heat exchange process; the stagnation cell is significantly reduced and the convection increases. These results are in agreement with the results of [4].

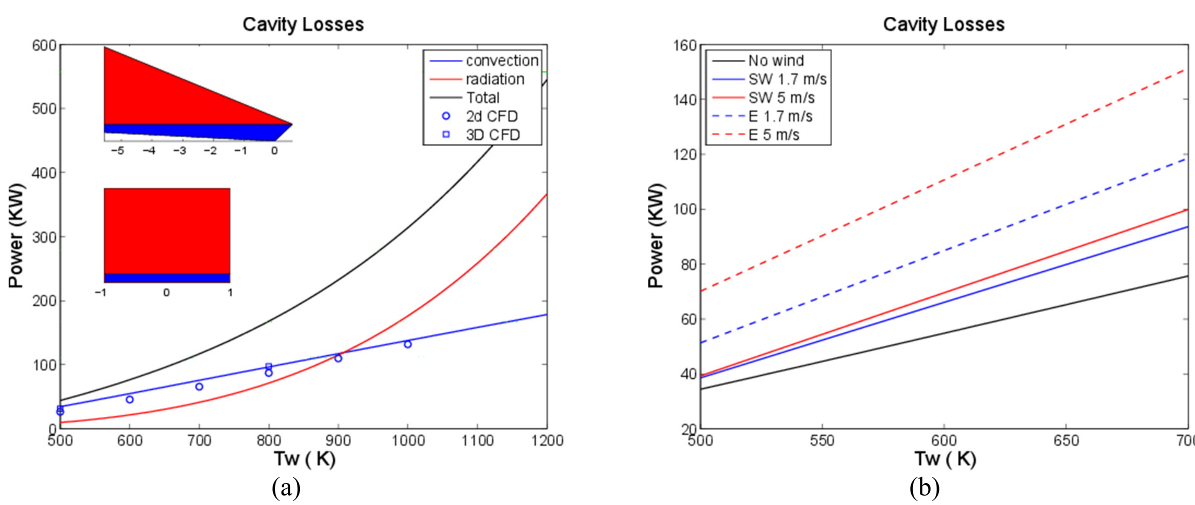

FIGURE 5. (a) Cavity losses validation of the correlation (blue) and radiation losses (red). The black curve represents the total losses. Blue circles (squares) are CFD computed energy losses inferred from a 2D (3D). Inset display the lateral and back geometry of the cavity where blue is the convection zone and the red is the stagnation zone. (b) Cavity convection losses as a function of T_w for different wind velocities and direction.

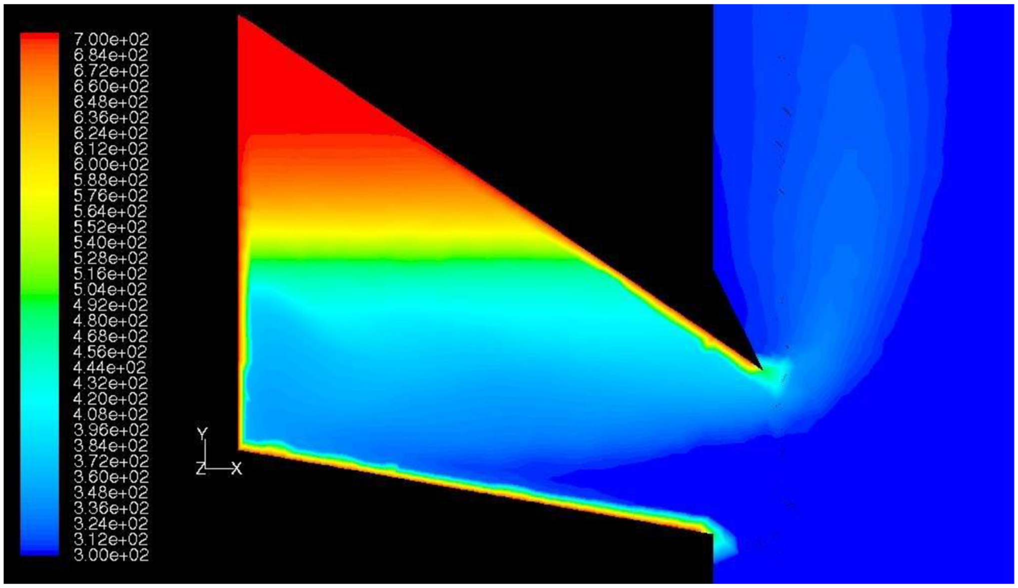

FIGURE 6. Temperature contour plot along the symmetry plane with a well defined stagnation zone (red) and the characteristic thermal plume heading up outside the cavity.

RESULTS FOR MEDIUM TEMPERATURE CONCEPT FRESNEL PLANTS

For the two-concept design of SeRe for medium temperature applications, temperature field, convection losses and film coefficient of every surface have been obtained. From **FIGURE 7** can be concluded that both thermal plumes are achieved and surface temperatures are greater at the trapezoidal SeRe than cylindrical due to, on one hand, to the input radiation on the SeRe coming from the primary reflectors, and on the other hand, although the stagnation zone is not achieved in these models, air temperature around the linear receiver is higher and the incident radiation at the trapezoidal SeRe is also higher due to optical performance.

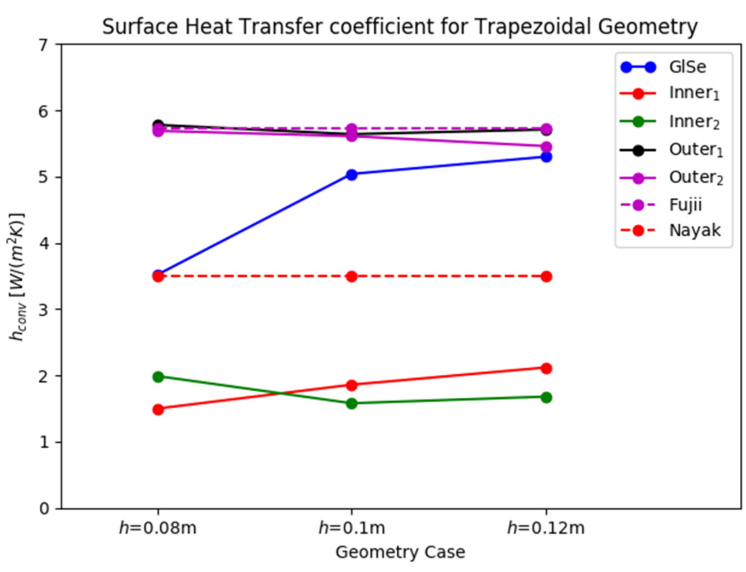

FIGURE 7. Surface heat transfer coefficient evolution for different values of parameter h for trapezoidal geometry, compared to Fujii [5] and Nayak [2].

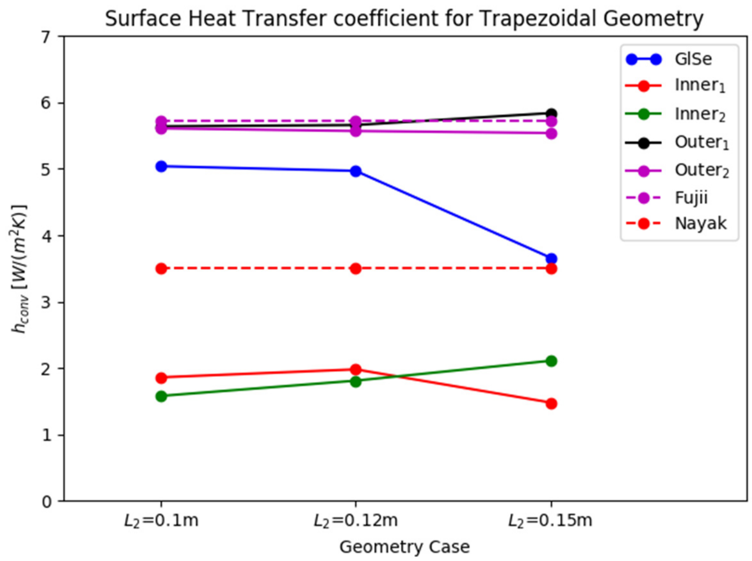

FIGURE 8. Surface heat transfer coefficient evolution for different values of parameter L_2 for trapezoidal geometry, compared to Fujii [5] and Nayak [2].

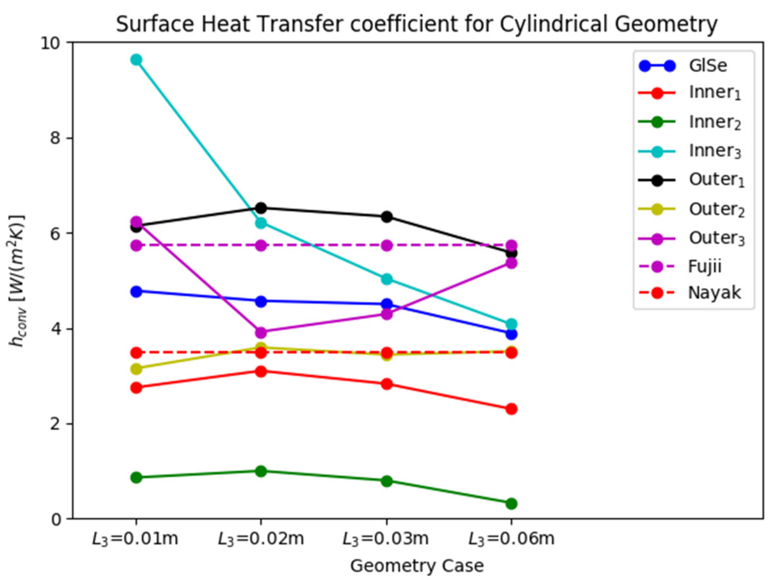

FIGURE 9. Surface heat transfer coefficient evolution for different values of parameter L_3 for cylindrical geometry, compared to Fujii [5] and Nayak [2].

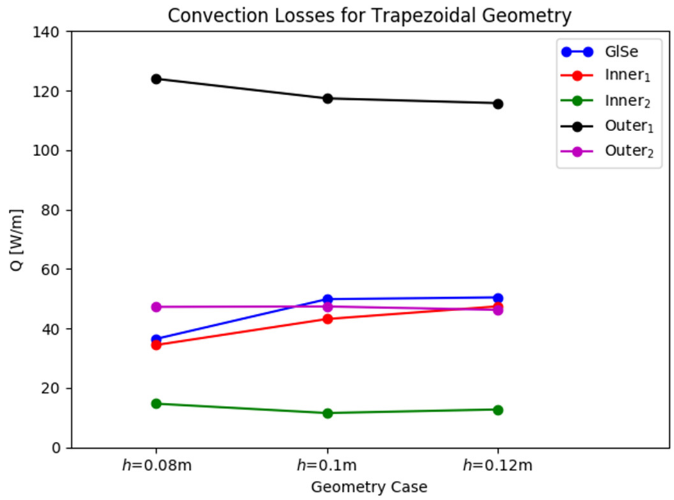

FIGURE 10. Convection losses evolution for different values of parameter h for trapezoidal geometry. Q is calculated from the entire circumference for GlSe, i.e., symmetry is undone.

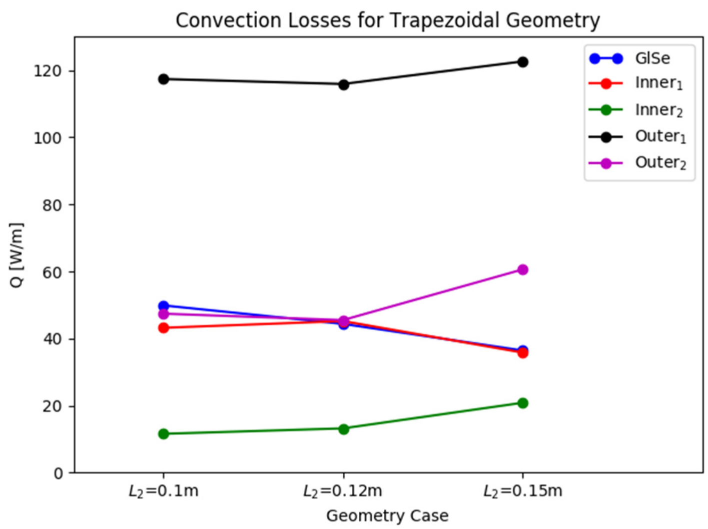

FIGURE 11. Convection losses evolution for different values of parameter L_2 for trapezoidal geometry. Q is calculated from the entire circumference for GlSe, i.e., symmetry is undone.

On one hand, from trapezoidal geometry, GlSe h_{conv} decreases when the receiver is more embraced by the SeRe, i.e., when h gets smaller or L_2 higher, near-stagnation zone expands and, therefore, GlSe h_{conv} and convection losses decrease. Nayak correlation for inner surfaces is not valid in this model due to the presence of the linear receiver. However, Fujii correlation for tilted flat surfaces represents very well the natural convection at outer surfaces of the SeRe.

On the other hand, results from cylindrical geometry show similar behaviors: as L_3 parameter increases, the receiver is more embraced and its coefficient starts decreasing. However, unlike trapezoidal geometry, stagnation zone is not so well settled and, therefore, convection losses do not decrease for every higher length of L_3 parameter. The incident radiation profile of trapezoidal geometry causes higher solid temperatures than cylindrical geometry, therefore, although both GlSe h_{conv} have similar values, convection losses are greater when the SeRe has trapezoidal morphology.

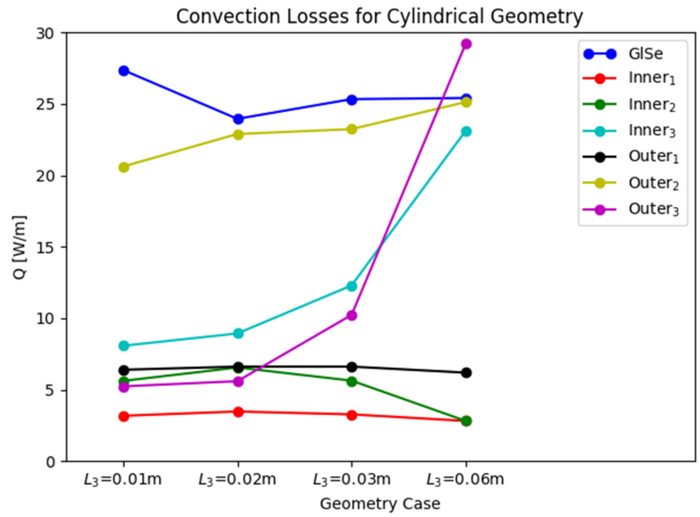

FIGURE 12. Convection losses evolution for different values of parameter L_3 for cylindrical geometry. Q is calculated from the entire circumference for GlSe, i.e., symmetry is undone.

CONCLUSIONS

This paper has shown some ways of estimate thermal cavity losses in CSP applications. A central tower cavity thermal problem has been solved by means of two 3D thermal-fluid-mechanical and thermal-fluid models, and Nayak correlation has been validated for this kind of cavity. A concept Fresnel cavity for medium temperature applications has been solved by means of two models, 3D thermal-fluid-mechanical and 2D thermal-fluid, and no correlation can be fully validated. Experimental results are needed from this matter in order to obtain a new correlation for convection and radiation losses in this new concept of Fresnel cavities for medium temperature applications.

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