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Investigations on Heat Loss in Solar Tower Receivers with Wind Speed Variation

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Abstract: The performance of the Solar Tower Receiver (STR) affects significantly the efficiency of the entire solar power generation system and minimizing the heat loss of the STR plays a dominant role in increasing its performance. Unlike the other thermal losses the convective heat loss in STR has direct relation with wind conditions. In this study a Simulation tool ANSYS® FLUENT® was used to determine the convection heat loss in both cavity and externalSTR at wind speed varies from(2) to (10) m/s. A fixed tilt angle ($\theta = 90°$) for the cavity receiver is adopted. The results show that the convection heat loss in both receivers increases with increase of wind speed. The absolute values are considerably lower in the case of the cavity with comparison to the external type. Furthermore, the radiative heat loss in the external and the cavity receivers is investigated. The results show that for the same absorbed area, the radiation loss in the cavity is lower by almost (80%) than the radiation loss in the external.

Keywords: Solar Tower Receiver, Central Receiver System, Heat loss, CFD

1. Introduction

Concentrated Solar Power technology implementation is growing fast. In 2013 , (2.136) GW are operating, (2.477) GW under construction and (10.135) GW are announced mainly in the USA followed by Spain and China and about (17) GW of CSP projects are under development worldwide [1]. Central Receiver Systems (CRS) arevery promising from the point of view of cost produced electricity [2].TheCRS is composed of the following main components: the heliostats, the receiver, the power block and thermal storage and balance of plant components allow high temperatures which lead to high efficiency of the power conversion system [3], easy integration in fossil plants for hybrid operation in a wide variety of options. It has the potential for generating electricity with high annual capacity factors from(0.4) to (0.8) through the use of thermal storage [3], and great potential for cost reduction and efficiency improvements [4]. Table (1) shows some characteristics of CRS.

Table 1. *Characteristics of solar thermal central receiver systems [5] .*

Typical Size	10-200 MWel
Operating Temperature	
• Rankine	565 ˚C
• Brayton	800 ˚C
Annual Capacity Factor	20 – 77%*
Peak Efficiency	16 – 23%*
Annual Net Efficiency	12 – 20%*
Available Storage technologies	Nitrate salt for molten salt receivers Ceramic bed for air receivers
Hybrid designs	Yes

Since the early$1980s$ the tower technology has attracted worldwide a lot of interest and numerous pilot projects have been successfully tested inUSA, Spain, and France. However, the first commercial Concentrating Solar Thermal Power Plants using large heliostat field and a solar receiver placed on the top of a tower are PS10, PS20, and Gemasolar in Spain [6]. The Ivanpah and Crescent Dunes installations in the USA pass the(100) MW_{el} threshold in 2013and 2014. In the case of power towers, incident sunrays are tracked by large mirrored collectors (heliostats) which concentrate the energy flux towards radiative /convective heat exchangers, called solar receivers, where energy is transferred to a working thermal fluid. After energy collection by this solar subsystem comprised of optical concentrator, and solar receiver, the conversion of thermal energy to electricity has many similarities with the one from fossil-fueled thermal power plants [3].

As the thermal receiver plays a very important role of transferring the solar heat to the engine, and heat loss of the thermal receiver can significantly reduce the efficiency and consequently the cost effectiveness of the system, it is important to assess and subsequently improve the thermal performance of the thermal receiver [7]. Therefore, the research aiming to minimize heat losses of existing receivers and developing of new designs is of great interest.

There are different types of receivers that can be classified into four groups depending on their functionality and geometric configuration. The four groups are external receivers, volumetric receivers, the cavity receivers and the particle receivers [1]. The present study will focus only on external (outer surface) and cavity type receivers.

The essential feature of the receiver in power tower plants is to absorb the maximum amount of the concentrated solar irradiance and transfer it to the working fluid or materials heat with minimum losses. In external type, the receiver is exposed to the environment and receives the solar irradiation from the heliostats. Being exposed to the environment, the performance of the external receivers can be affected by the environment conditions such as wind speed and ambient temperature. The radiation and convection heat transfer from the hot surface temperature of the receiver to the environment will reduce its efficiency. The cavity receiver is not exposed to the environment to the same extent as external type. However, still the heat loss in cavity receiver remains an issue to be solved and quantified. To date, some publications concerning the heat loss of the STR are available. Clausing (1981) presented an analytical model which enables the estimation of convective losses from cavity receivers and his model indicated that the influence of the wind on the convective loss at normal operating conditions is minimal [8]. Stine and McDonaled (1989) stated that the convective heat loss depends on the air temperature within the cavity, the inclination of the cavity and external wind conditions [9]. Leibfried and Ortiohann (1995) reported that the radiation loss is dependent on the cavity wall temperature, the shape factors and emissivity/absorptivity of the receiver walls, while conduction is dependent on the receiver temperature and the insulation material [10]. Sendhil Kumara (2007) investigated the approximate estimation of the natural convection heat loss from actual geometry of a cavity receiver by varying the inclinations of the receiver from 0° to 90° and also investigated the effect of operating temperature [11]. Prakash M. (2008) studied the convective and radiative heat losses of a cavity receiver taking in consideration different wind speed and direction, input temperature and receiver inclination angel [12]. Shuang and Xiao (2010)obtained that the conduction and radiation in cavity receivers can readily be determined analytically, on the other hand, the determination of convection heat loss is rather difficult due to the complexity of the temperature and velocity fields in and around the cavity and usually rely on semi-empirical models [7]. Qiang and Zhifen (2011) presented that wind conditions can obviously affect the thermal losses and the value reaches its maximum when the wind blows from the side of the receiver ($\theta = 90°$)[13]. Figure (1)shows the tilt angle in a cavity receiver

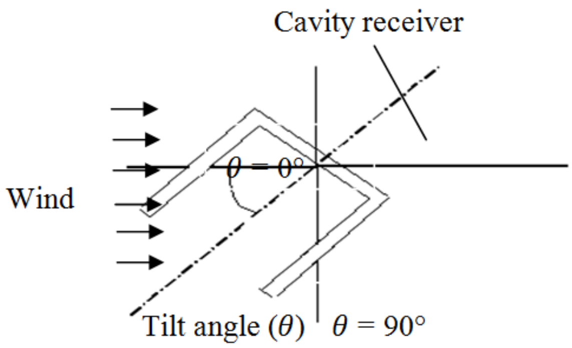

Fig. 1. *A sketch illustrates the tilt angle.*

The objective of this study is to investigate the convection heat loss characteristics in external and cavity receivers with a fixed tilt angle of (90°) and varying wind speed, using Computational Fluid Dynamics (CFD) simulation ANSYS® FLUENT®

2. Mathematical Model

Thermal losses in STR are mainly consists of three losses as following: conduction loss ($\dot{Q}_{loss,cond}$) ,radiation loss ($\dot{Q}_{loss,rad}$) , and convection loss ($\dot{Q}_{loss,conv}$) .The conduction heat loss occurs because of the heat-conduction through the receiver body, and it can be minimized by using insulated material with low thermal conductivity. In this model an assumption was made that all surfaces are adiabatic surfaces and therefore, heat flux passes through them is negligible. The radiation heat loss is caused by infrared radiation that emits from the receiver wall to the environment. In the other hand the convective heat loss occurs because of the convective heat exchange between the receiver walls and the air flowing along the receiver wall. The radiation and convection heat losses will be considered in this study.

2.1. Radiation Heat Loss

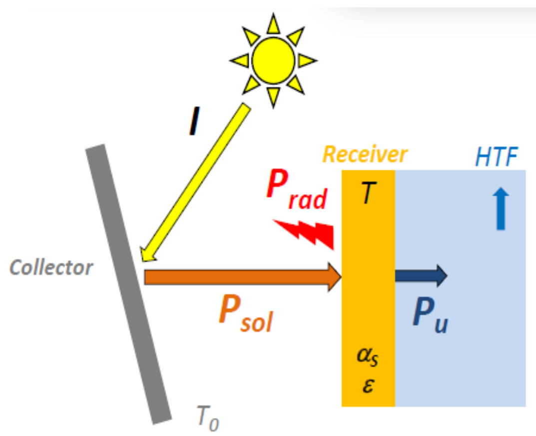

Fig. 2. *Energy balance of a solar absorber [14].*

The radiation loss ($\dot{Q}_{loss,rad}$) is caused by infrared radiation, which emits from the receiver walls to the environment. The radiative loss is dependent on wall temperature, the shape factors and emissivity/absorptivity of the receiver walls and independent of the inclination [9]. The figure (2) shows an energy balance of a solar absorber.

The total infrared energy can be calculated by the Stefan-Boltzmann law as:

$$E = \sigma T^4 \quad (1)$$

$$T^4 = (T_{Abs}^4 - T_{Amb}^4) \quad (2)$$

The solar power to the receiver ($\dot{Q}_{Sol}$) can be calculated by applying an energy balance on the solar absorber, as following:

$$\dot{Q}_{Sol} = \dot{Q}_{HTM} + \dot{Q}_{loss,rad} + \dot{Q}_{loss,conv} + \dot{Q}_{refl} \quad (3)$$

Where

($\dot{Q}_{HTM}$) is the power transferred to the heat transfer medium.

($\dot{Q}_{refl}$) is the reflected power which happens when the incident rays of visible light that come from the heliostats are reflected at the wall of the receiver.

Rearrangement yields:

$$\dot{Q}_{HTM} = \dot{Q}_{Sol} - \dot{Q}_{loss,rad} - \dot{Q}_{loss,conv} - \dot{Q}_{refl} \quad (4)$$

With

$$\dot{Q}_{Sol} = C \cdot A_{Abs} \cdot I \cdot \eta_{optical} \quad (5)$$

$$\dot{Q}_{loss,rad} = \varepsilon \cdot \sigma \cdot A_{Abs} \cdot T^4 \quad (6)$$

$$\dot{Q}_{loss,conv} = h \cdot A_{abs} \cdot (T_{Abs} - T_{Amb}) \quad (7)$$

$$\dot{Q}_{refl} = (1-\alpha) \cdot \dot{Q}_{Sol} \quad (8)$$

The energy efficiency of the system can be calculated

$$\eta_{sys} = \frac{\dot{Q}_{HTM}}{\dot{Q}_{Sol}} = \frac{\dot{Q}_{Sol} - \dot{Q}_{loss,rad} - \dot{Q}_{loss,conv} - \dot{Q}_{refl}}{\dot{Q}_{Sol}} \quad (9)$$

The convective heat loss ($\dot{Q}_{loss,conv}$) will be illustrated more in details later in this paper.

The radiative heat loss $(\dot{Q}_{loss,rad})$ in the case of the external receiver can be calculated as described above in equation (6):

$$\dot{Q}_{loss,rad,external} = \varepsilon \cdot \sigma \cdot A_{Abs} \cdot T^4 \quad (10)$$

The radiative heat loss $(\dot{Q}_{loss,rad})$in the case of the cavity receiver can be calculated using the following equation [15]:

$$\dot{Q}_{loss,rad,cavity} = \varepsilon_{eff}.\sigma.A_{Ape}.T^4 \quad (11)$$

The aperture area of the cavity receiver(A_{Ape})in the model is equal to (10.89) m^2.

The effective emissivity (ε_{eff}) of the cavity can be derived using the following equation [15]:

$$\varepsilon_{eff} = \frac{1}{1+(\frac{1-\varepsilon}{\varepsilon})\frac{A_{Ape}}{A_{Abs}}} \quad (12)$$

2.2. Comparison of Radiative Losses (External VS. Cavity)

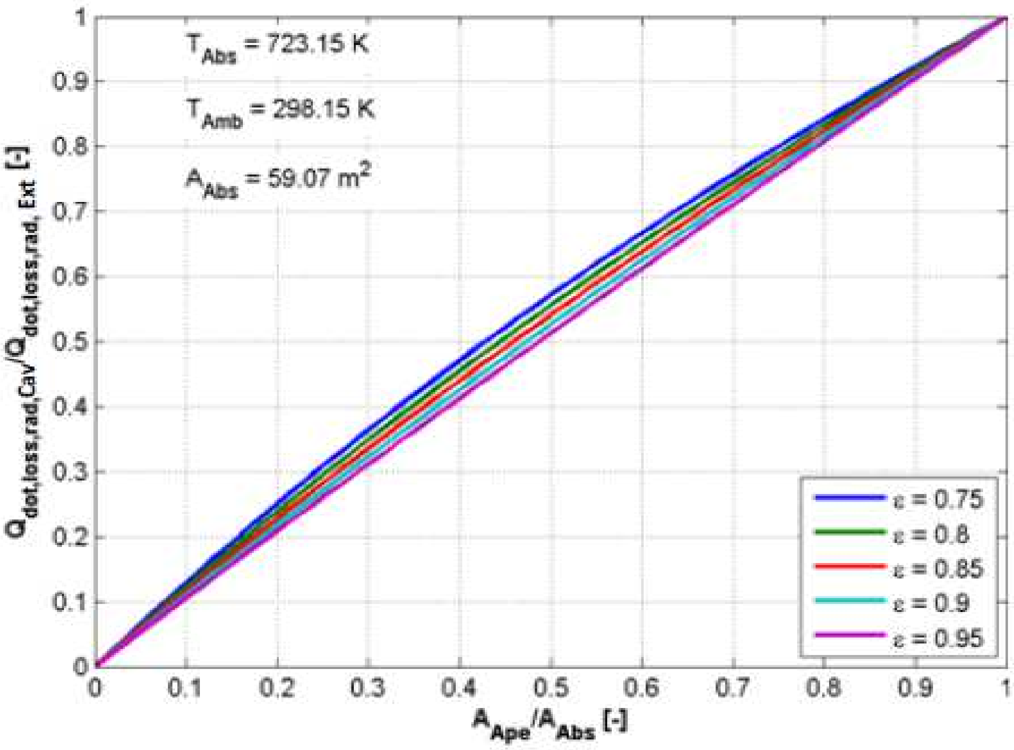

Fig. 3. *Ratio of radiative losses for external and cavity type receivers versus the ratio of aperture to absorber area.*

Even though the areas of the absorbers in both receivers in our model are equal ($59.07\ m^2$); however, the radiation loss in both receivers is not the same. As described above in questions (10) and (11) the radiation lossin the cavity receiver depends on the aperture area of the cavity not on the absorbed area as the case of the external receiver. Therefore, the radiation loss associated with the external type receiver is much higher than the radiation loss associated with the cavity type receiver. Figure (3) below illustrates the ratio of radiative losses for both receivers versus the ratio of the aperture to the absorber area.

2.3. Convection Heat Loss

The convective heat loss is the only heat loss in the receiver that has direct relation with the wind conditions. Since there is an existing empirical formula to calculate the heat loss in a turbulent flow over a flat plate as shown in equations below [15], it appears reason to do the simulation for the receiver as a flat plate with the same hot surface area, thickness, and insulation material and to compare the results with the empirical results. This allows validating the simulation model.

The average laminar Nusselt number $Nu_{1,lam}$ for a flat plate of the length l is calculated by equation (13):

$$Nu_{1,lam} = 0.664\,\sqrt{Re_1}\,\sqrt[3]{Pr} \tag{13}$$

With $Nu_{1,lam} = \frac{hl}{\lambda}$ and $Re_1 = \frac{ul}{\nu}$

$$Re < 10^5$$

The average turbulent Nusselt number $Nu_{1,turb}$ for a flat plate of the length l is calculated by equation(14):

$$Nu_{1,turb} = (0.037 Re_1^{0.8} Pr) \Big/ \left(1 + 2.443 Re_1^{-0.1}\left(Pr^{-\frac{2}{3}} - 1\right)\right) \tag{14}$$

$$5 \times 10^5 < Re < 10^7$$

$$Nu_{1,0} = \sqrt{\left(Nu_{1,lam}^2 + Nu_{1,turb}^2\right)} \tag{15}$$

Where $Nu_{1,0} = \frac{hl}{\lambda}$

$$10^1 < Re < 10^{10}$$

The convection heat transfer coefficient can be calculated by:

$$h = \left(Nu_{1,0}\lambda / l\right) \tag{16}$$

Knowing the convection heat transfer coefficient, the convection heat loss can be estimated by equation(17):

$$\dot{Q}_{loss,conv} = A_{Hot}.h.(T_{Hot} - T_{Amb}) \tag{17}$$

3. Physical Model

The physical model of the receivers is shown in the Fig. 4.There are few assumptions made in all simulation cases as following:

1. Hot surface area of all receivers is $(59.07)\ m^2$.
2. Thickness of the flat plate and cavity receivers is $(0.35)\ m$.
3. Insulation material is Rockwool.
4. Surrounding temperature is $(25)\ ^\circ C$.
5. Temperature of hot surface is $(450)\ ^\circ C$.
6. Height of the receiver is $(70)\ m$ above ground.

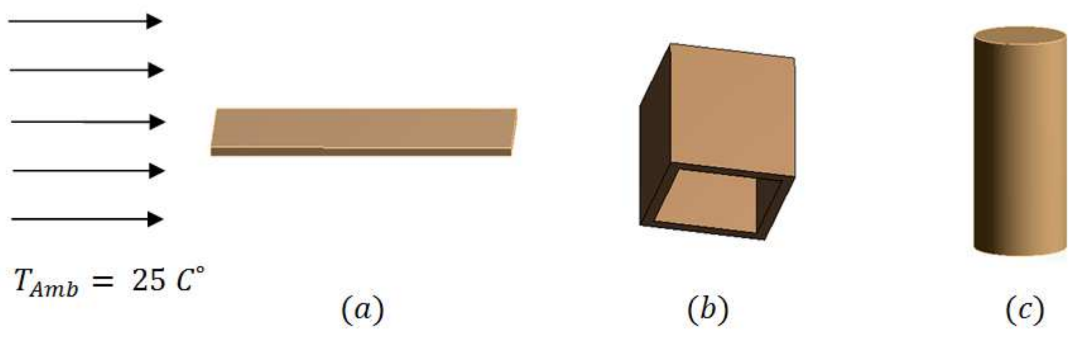

Fig. 4. *Three dimensional models of Central Receiver System.(a) Flat plate receive (b) Cavity receiver with cube geometry. (c) External receiver with cylindrical geometry.*

All receivers are facing the wind with a tilt angle of(90°). The area of the hot surfaces in all receivers is assumed to be equal and the dimensions are as following: in the flat plate (a) is $(16 \times 3.692)\ m$ as length and width respectively. In the cube receiver (b) the length of the inner cube is $(3.3)\ m$, the height is $(3.65)\ m$ and the roof area is $(3.3 \times 3.3)\ m$. In the cylindrical receiver (c) the outer diameter is $(3.3)\ m$ and the height is$(5.7)\ m$. The imaginary wind tunnel used in the model is external type. Each one of the receivers will be centered in the tunnel in order to apply the boundary conditions such as temperature and wind speed. The cross section of the tunnel is $(50 \times 50)\ m$ with$(50)\ m$ length.

4. Velocity Distribution

Fig. $5(a, b)$.Show the velocity distributions in cavity and external receivers with the same wind speed of $(10)\ m/s$. It should be mentioned that the flow is from left to right. The cavity is supposed to be suspended from a cantilever type receiver. The impact of tower and cantilever on the flow field across the cavity is neglected. It is obvious that the wind speed along the hot surface inside the cavity is lower than in the wind speed around the hot surface of an external receiver. That is because of the air recirculation which reduces the heat transfer coefficient and as a result reduces the convection heat loss.

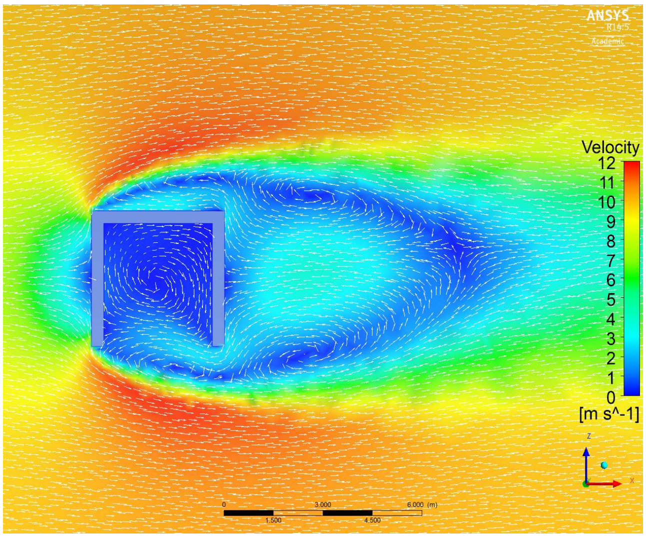

Fig. 5(a). *Velocity distribution around and in the cavity receiver.*

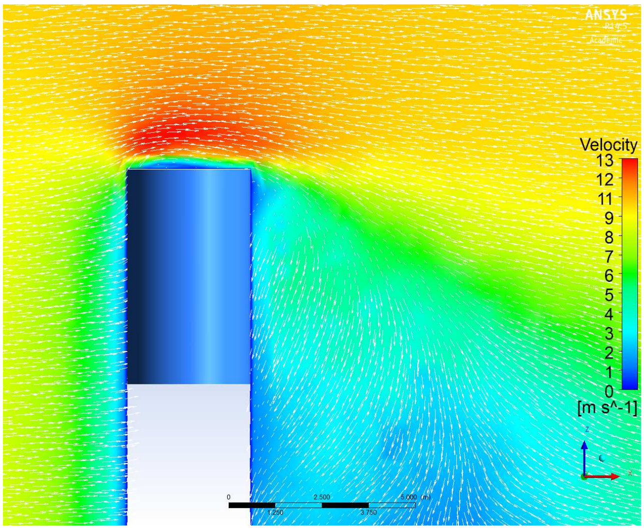

Fig. 5(b). *Velocity distribution around the external receiver.*

5. Calculation of Convective Heat Transfer Coefficient

In the flat plate receiver the convection heat transfer coefficient can be calculated theoretically by the above equations and the result shown in table (2)

Table 2. *Convective heat transfer coefficient with different wind speed.*

Wind speed (m)	Nu	$h(W/m^2\ K)$	$\dot{Q}_{loss,conv}(MW)$
2	3400.71	5.58	0.1400
3	4649.02	7.62	0.1919
4	5809.09	9.53	0.2392
5	6907.98	11.33	0.2844
6	7960.60	13.06	0.3278
7	8976.33	14.72	0.3696
8	9961.58	16.34	0.4101
9	10920.96	17.91	0.4496
10	11857.97	19.45	0.4882

Figure (6) shows the calculated convective heat transfer coefficient increases approximately in early with increase of wind speed.

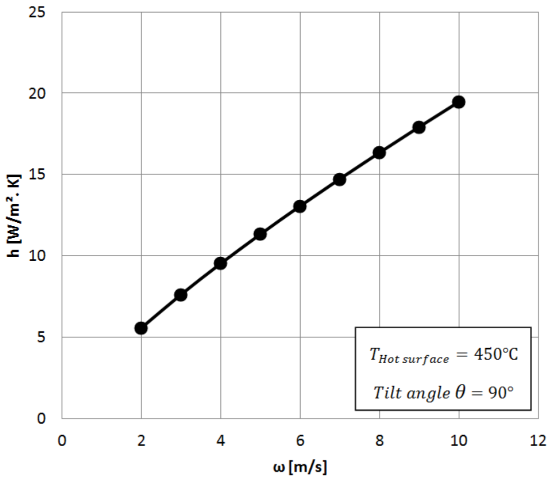

Fig. 6. *The calculated convective heat transfer coefficient of the flat plate receiver verus wind speed.*

In order to examine the accuracy of the model, the result of the analytically calculated convective heat transfer coefficient was compared to the numerical result from the model (FLUENT®) with $(450)\ ^\circ C$ temperature of the hot surface, $(90°)$ title angle and different wind speed varying from (2) to $(10)\ m/s$.Based on Figure (7) there is a good agreement between the analytical and numerical results for the convective heat coefficient.

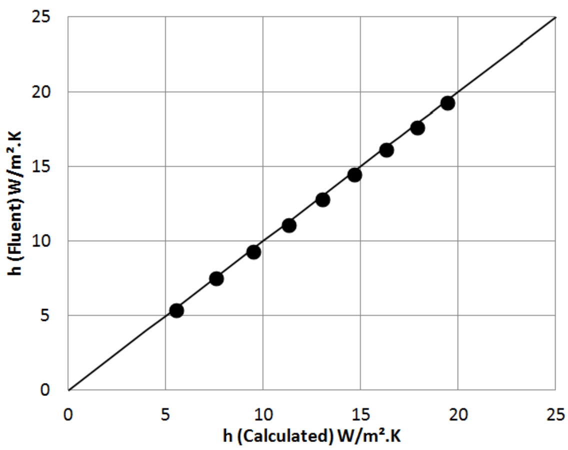

Fig. 7. *Comparison between the analytical and numerical heat transfer coefficient for the Flat Plate receiver.*

To compare the convective heat transfer coefficient values of the cavity receiver (cube geometry) and the external receiver (cylindrical geometry), it is necessary to apply the same boundary conditions on both receivers. Figure (8) shows a comparison between the convective heat transfer coefficients of all receivers with different wind speed.

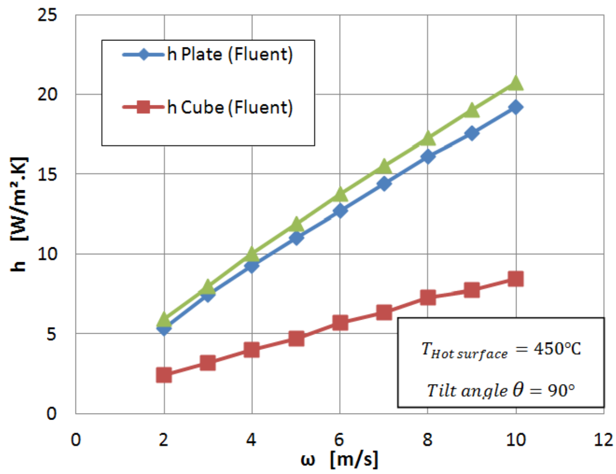

Fig. 8. The convective heat transfer coefficient of three receivers at different wind speed.

6. Results and Discussion

The convective heat loss of all receivers with different wind speed, fixed tilt angle (90°) and fixed hot surface temperature (450) $^{\circ}C$ is shown in figure (9). The convective heat lossshows for all types of receivers nearly linear dependence on wind speed. The cavity receiver is suffering the smallest convective heat loss of all the receivers examined.

For the same conditions applied on the cavity and external ("outside") receivers, the radiative heat loss in the cavity receiver reduced by almost (80) % with comparison to the radiative heat loss in the external receiver.

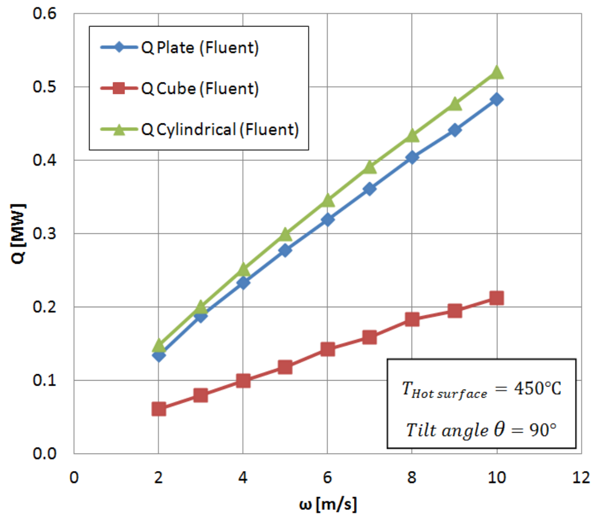

Fig. 9. The convective heat loss of all receivers at different wind speed.

Table 3. Shows the numerical result of the convective heat loss in all receivers.

Wind speed $[m/s]$	$\dot{Q}_{loss,conv}$ Plate $[MW](Fluent)$	$\dot{Q}_{loss,conv}$ Cube $[MW](Fluent)$	$\dot{Q}_{loss,conv}$ Cylindrical $[MW](Fluent)$
2	0.1341	0.0614	0.1485
3	0.1875	0.0801	0.2002
4	0.2327	0.0998	0.2515
5	0.2767	0.1184	0.2992
6	0.3196	0.1429	0.3455
7	0.3612	0.1589	0.3905
8	0.4044	0.1827	0.4344
9	0.4415	0.1944	0.4775
10	0.4829	0.2123	0.5200

7. Conclusions

The effects of wind speed variation on the convective heat loss of both cavity receiver and external outer surface receiver at a tilt angle ($\theta = 90°$) is studied. The following conclusions can be drawn from study:

First, the convective heat loss is increasing almost linearly with the increase of the wind speed and the highest convective heat loss is obtained at highest wind speed. Second, the convection heat loss associated with using cavity receiver is much lower than the loss associated with using external receivers. For the analyzed geometry the convective heat loss in the cavity receiver was approximately half of the convective heat loss in the external 'outside' receiver.

Furthermore, regardless of the wind speed and the tilt angle, the radiative heat loss in the cavity receiver is almost (80)% less than the radiative heat loss in the external receiver for the same absorbed area.

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Nomenclature

h : Convective heat transfer coefficient $(W/m2K)$
ω: Wind speed (m/s)
ℓ: Characteristic length (m)
ν : Kinematic viscosity $(m2/s)$

λ : Thermal conductivity ($W/m.K$)
T_{Hot}: Hot surface temperature($^{\circ}C$)
T_{Amb}: Ambient temperature ($^{\circ}C$)
Nu: Nusselt number
Re : Reynolds number
Pr: Prandtl number
E : Infrared energy (W)
C: Concentration factor
A_{Hot} : Hot surface area of the absorber (m^2)
A_{Abs}: Receiver's absorbed area (m^2)
A_{Ape}: Receiver's aperture area (m^2)
I: Irradiance flux (W/m^2)
ε: Emissivity
α : Absorptivity
σ: Stefan-Boltzmann constant
$\eta_{optical}$: Optical efficiency of the receiver mirrors
η_{sys}: System efficiency

References

[1] Behar, O., Khellaf, A., and Mohammedi, K.,"A review of studies on central receiver solar thermal power plants". In: Renewable and Sustainable Energy Reviews, Algeria, pp. 12-39, 2013.

[2] An Overview of CSP in Europe, North Africa and the Middle East, CSP Today, October; 2008.

[3] IRENA, 2012, "Concentrating Solar Power. Cost analysis series". In: Renewable Energy Technologies.

[4] Romero, Manuel.,Zarza, E., 2007, "Concentrating solar thermal power. Energy conversion". In: Taylor & Francis Group.

[5] Romero, M., Buck, R., Pacheco, J., "An Update on Solar Central Receiver Systems, Projects, and Technologies". In: Solar Energy Engineering, Madrid, Spain, pp. 98-108, 2002.

[6] Antonio, L., A vila-Marın., "Volumetric receivers in Solar Thermal Power Plants withCentral Receiver System technology: A review“. In: Solar Energy, Madrid, Spain, pp. 891-910, 2011.

[7] Shuang-Ying Wu., Lan, Xiao., "Convection heat loss from cavity receiver in parabolic dish solar thermal power system: A review". In: Solar Energy, Chine, pp. 1342-1355, 2010.

[8] Clausing, A.M., "An analysis of convective losses from cavity solar central receivers". In: Solar Energy, USA, pp. 295-300, 1981.

[9] Stine, W.B., McDonald, C.G., 1989, "Cavity receiver convective heat loss". In:Proceedings of International Solar Energy Society Solar World Congress,Japan, pp. 1318–1322.

[10] Leibfried, U., Ortjohann, J., "Convective heat loss from upward anddownward-facingcavity solar receivers: measurements and calculations". In: Solar Energy Engineering, pp. 75–84, 1995.

[11] Sendhil Kumar, N., Reddy, K.S., "Numerical investigation of naturalconvection heat loss in modified cavity receiver for fuzzy focal solar dish concentrator". In: Solar Energy, pp. 846–855, 2007.

[12] Prakash, M., Kedare, S.B., Nayak, J.K., "Investigations on heat losses from a solar cavity receiver". In: Solar Energy, India, pp. 157–170, 2008.

[13] Qiang, Yu.,Zhifeng, Wang., "Simulation and analysis of the central cavity receiver's performanceof solar thermal power tower plant". In: Solar Energy, Chine, pp. 164-174, 2011.

[14] A. Soum-Glaude, I. Bousquet, M. Bichotte, S. Quoizola, L. Thomas, G.Flamant, 2013,"Optical characterization and modeling of coatings intended as high temperature solar selective absorbers". In: Solar PACES 2013, Las Vegas.

[15] Springer, 2010, "VDI Heat Atlas, Second Edition". Heidelberg, Germany.