Thermo-Fluidic Analysis of Finned Receiver for Concentrator Solar Power Applications

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***Abstract*—this paper analyses the effects of using internally finned receiver for parabolic trough or Fresnel concentrator. A new 3-D model of a receiver with rectangular insert is proposed, the model approach the fluid dynamic interaction with the heat transfer inside the receiver, the solution of the physical problem with the K-Ɛ turbulent model is obtained by COMSOL Multiphysics, the Tube inlet is defined as a mass flow entrance for the Syltherm 800 which is used as a heat transfer fluid in the present model. The Thermo-Fluidic study investigate for several solar radiation for the rectangular fin, and for various mass flow rates, the chosen geometry is studied and compared to experimental data , the analysis shows a heat transfer enhancement without any effects for the fluid flow inside the receiver**

Keywords; CFD ; CSP; K-Ɛ; COMSOL; solar absorber

I. Introduction

Energy has always been very important for human societies; human behavior is strongly induced by its availability, price and resources [1], fossil energy resources are so limited and unclean (global warming….) [2] this why the whole world must work to assure the production of energy. Nine Solar Electric Generation Systems (SEGS) built in southern California between 1984 and 1990 continue to produce 14-80 [MWe] of utility-scale electric power each from solar thermal energy input. The systems collect energy using one of synthetics heat transfer fluids pumped through absorber tubes in the focal line of parabolic trough collectors who's calculated. The heated fluid provides the thermal resource to drive a Rankine steam power cycle [3].

In Algeria, Hassi R'Mel hybrid power plant Fig.1 is the first gas-solar power plant in Algeria; it is located in the region Tilghemt, 25 km north of Hassi R'Mel largest gas field in Africa. it started in July 2011, With a capacity of 150 mega-watts (MW),The power plant will combine an array of parabolic trough concentrating solar power of 30 MW, in an area of 180,000 m2 and in conjunction with a gas turbine power plant of 120 megawatts. Other plants were already scheduled; two stations on the same principle are scheduled for 2013, in eastern Algeria, the solar power plant Maghaïr in the state of El-Oued and to the west the solar power plant Naama in the state of El-Bayad[4].

Figure 1. Solar Field Of Hassi R'Mel Hybrid Power Plant.

One of the major disadvantage of CSP plants is there efficiency (1), it's (the electricity produced divided by the annual solar radiation in the field) between 11% and 17% [4] comparing to the gas turbine power plant (between 30% and 40%). This paper presents CFD study using the commercial code Comsol to enhance the solar field efficiency by enhancing the heat transfer among the solar absorber that has been applied for solar trough collectors by Ravi Kumar (2009) [5]. Internally finned tubes have not been studied since the author's knowledge for solar trough collectors, but the advantages of this design for enhance heat transfer has been documented in a great amount of papers [6–12]. The estimation of the thermal behavior with our model has been qualified and validated with its application to the analysis of the experimental data obtained in the Sandia laboratory [13]

$$\eta_{t-e} = \eta_{\text{field}} \cdot \eta_{\text{cycle}} - \frac{W_{\text{P}}}{A_{\text{F}} \cdot \text{DNI}} \tag{1}$$

Where:

η is the efficiency

W is the losses of power

A apparture

DNI direct normal irradiation

p,f parasitc, field

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II. MATHEMATICAL MODEL

There is no analytic solution for fluid flow and heat transfer equations which make the work within solar absorber very difficult and it's unavoidable to use numerical approximation

Continuity equation:

$$\nabla \cdot (\rho \mathbf{u}) = 0 \tag{2}$$

Conservation of momentum equation:

$$\rho(\mathbf{u} \cdot \nabla)\mathbf{u} =$$
$$\nabla \cdot \left[-p\mathbf{I} + (\mu + \mu_T)\left(\nabla \mathbf{u} + (\nabla \mathbf{u})^T\right) - \frac{2}{3}(\mu + \mu_T)(\nabla \cdot \mathbf{u})\mathbf{I} - \frac{2}{3}\rho k \mathbf{I} \right] + \mathbf{F} \tag{3}$$

Conservation of energy equation:

$$\rho C_p \mathbf{u} \cdot \nabla T = \nabla \cdot (k \nabla T) + Q + Q_{vh} + W_p \tag{4}$$

In the simulation the fluid is considered a fully developed turbulent flow, for this we will use two more equations

K equation:

$$P_k = \mu_T \left[\nabla \mathbf{u} : \left(\nabla \mathbf{u} + (\nabla \mathbf{u})^T\right) - \frac{2}{3}(\nabla \cdot \mathbf{u})^2 \right] - \frac{2}{3}\rho k \nabla \cdot \mathbf{u} \tag{5}$$

$$\rho(\mathbf{u} \cdot \nabla)k = \nabla \cdot \left[\left(\mu + \frac{\mu_T}{\sigma_k}\right) \nabla k \right] + P_k - \rho \epsilon \tag{6}$$

Ɛ equation:

$$\rho(\mathbf{u} \cdot \nabla)\epsilon = \nabla \cdot \left[\left(\mu + \frac{\mu_T}{\sigma_\epsilon}\right) \nabla \epsilon \right] + C_{\epsilon 1}\frac{\epsilon}{k}P_k - C_{\epsilon 2}\rho\frac{\epsilon^2}{k}, \quad \epsilon = \mathrm{ep} \tag{7}$$

$$\mu_T = \rho C_\mu \frac{k^2}{\epsilon} \tag{8}$$

Heat losses based on LS-2 collector test:

Thermal losses = $673.17 + 1.09\,T + 8.15e\text{-}9\,T^4$ (9)

Where:

u velocity

μ Viscosity

ρ density

C calorific capacity

T temperature

P pressure

F external force

III. NUMERICAL VALIDATION

To validate the numerical simulation the first part of this study is dedicated for the comparison between numerical and experimental data.

Table 1 represent the absorber tube parameter, Table 2 represent HTF (heat transfert fluid) Syltherm 800 technical data used in the solar field of real CSP plant as in KRAMER JUNCTION CAL,USA.

TABLE I. PARAMETERS OF THE SOLAR ABSORBER

Parameter	Absorber configuration
Length of the receiver (m)	2.0
Inner diameter of the receiver (m)	0.054
Outer diameter of the receiver (m)	0.06
Absorber material	STAINLESS STEEL

TABLE II. SYLTHERM 800 TECHNICAL DATA @ 25°C

Parameter	HTF configuration
Viscosity	9.1 MPA.S
Auto ignition Point	385°C
Flash Point	160°C
Freeze Point	-60°C
Density	936 KG/M^3
Heat of Combustion	28,659 KJ/KG
Thermal conductivity	0.1012 W/M K
Heat capacity at constant pressure	1916 J/ KG. K

The simulation convergence trend is in Fig.2 and it shows that the obtained result has an error of estimation of 10^{-3}, the simulation can reach an error of 10^{-6} but it will take a lot of time and memory space to have those results because the number of freedom degree is 37794 for the smooth tube with a refine mesh in the interaction between solid-HTF and a diagonal symmetry. the freedom degree number can reach 40843 for the finned tube.

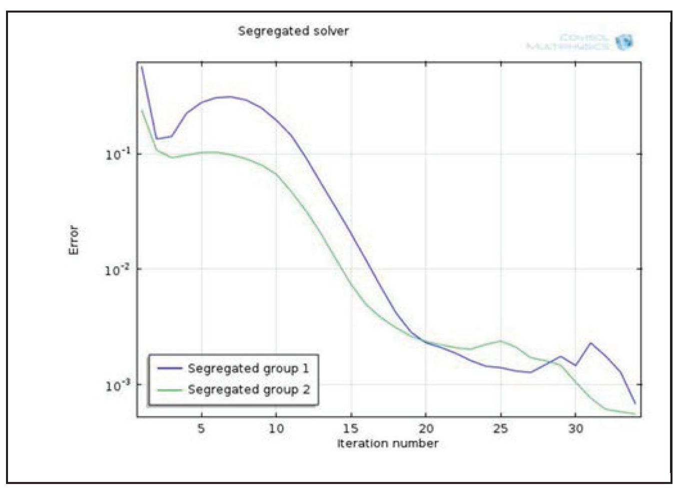

Figure 2. Convergence plot

The comparison between numerical and experimental data is shown in Table 3. The table shows a slight error between the two results we can attribute this error to the numerical calculation that is based on an iterative simulation with a predefined error. And also there is always an error in any experiment which leads us to conclude that the simulation process is accurate.

TABLE III. VALIDATION OF NUMERICAL SIMULATION

Parameter	DNI (W/m^2)	Flow rate (l/min)	T in (°C)	T out (°C)
LS-2	933.7	47.7	102.2	124
Simulation	933.7	47.7	102.2	120

IV. NUMERICAL SIMULATION

In this part, we will compare between smooth and internally finned tube the study will be done for two mass flow rates (inlet velocity)

A. *Smooth absorber with mass flow of 47.7 l/min*

The results obtained for the three major aspects for the solar absorber are in Fig.3-5

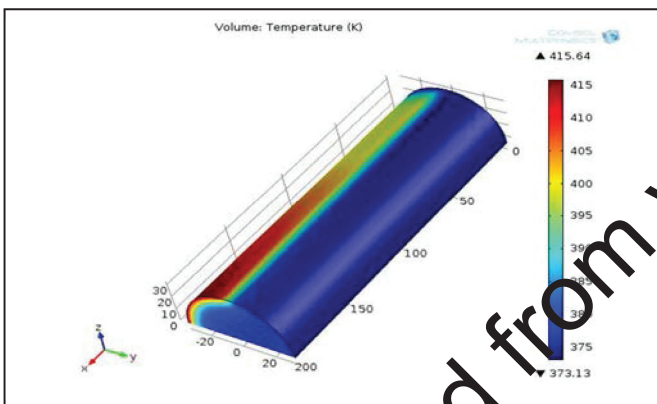

Figure 3. Temperature distribution for the smooth tube

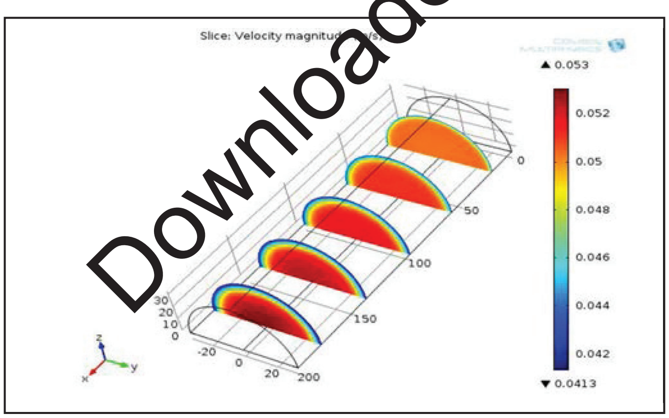

Figure 4. velocity distribution for the smooth tube

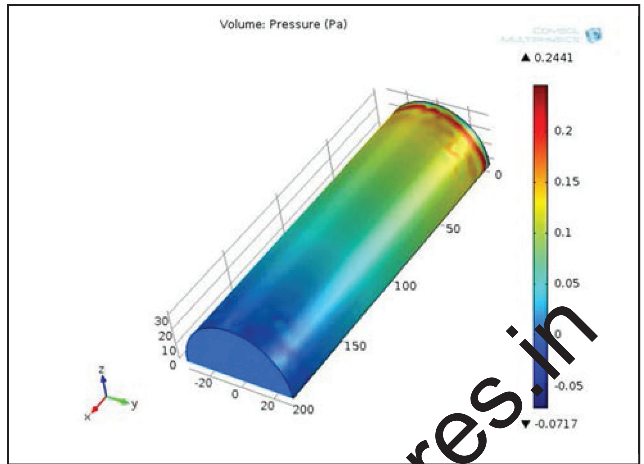

Figure 5. Pressure distribution for the smooth tube

The velocity distribution is as expected to be the magnitude of the velocity increase from the center to the outlet to reach 0m/s at the interaction Fluid-Solid.

The pressure distribution shows (0.23 Pa) that there is a slight pressure loss due to viscosity and the stainless steel.

The temperature distribution show there is no uniformity in heat transfer inside the absorber tube from here we will use an internally finned tube.

B. *Internally finned absorber with a mass flow of 47.7 l/min*

The velocity magnitude, the pressure and the temperature distribution are shown in Fig.6-8 for an absorber with a 2mm fin.

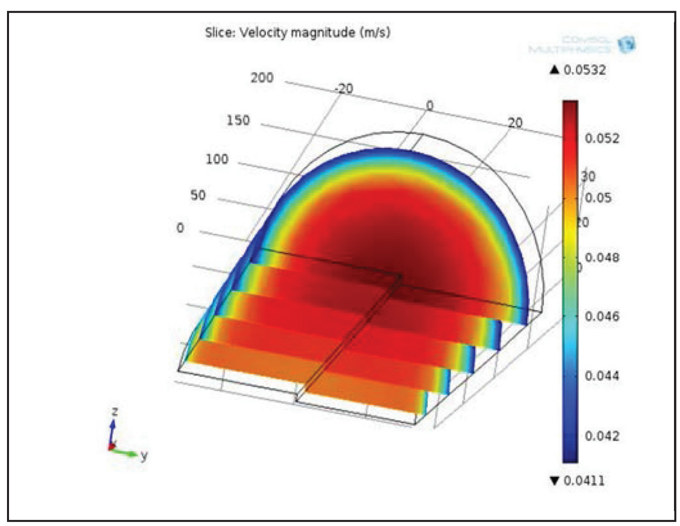

Figure 6. velocity distribution (finned tube 47.7 l/min)

The velocity can reach 50mm/s and it is not affected by the presence of the fin because of the high velocity at the center.

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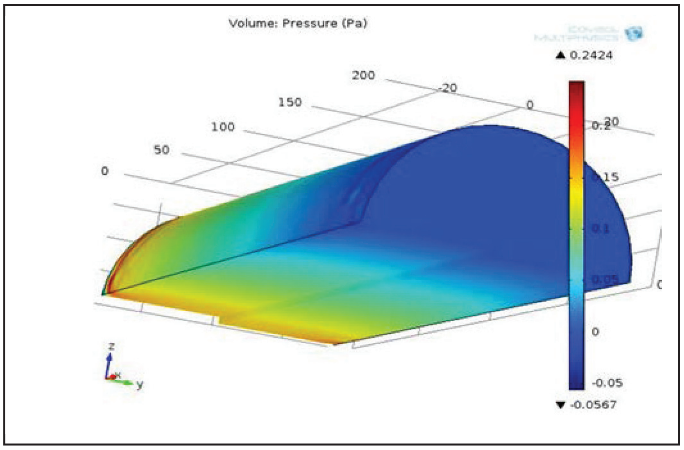

Figure 7. Pressure distrubution (finned tube 47.7 l/min)

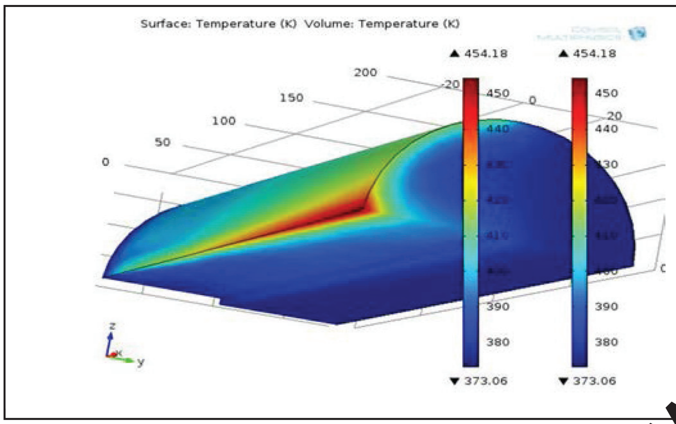

Figure 8. Temperature distrubution (finned tube 47.7 l/min)

After the simulation, we remark that the pressure losses for a finned tube and a mass flow of 47.7 l/min is considered so small or negligated and the fin enhance the heat transfer inside the absorber tube.

C. Internally finned absorber with a mass flow of 66.7 l/min

The velocity magnitude, the pressure and the temperature distribution for a mass flow of 66.7 l/min are shown in Fig.9-11

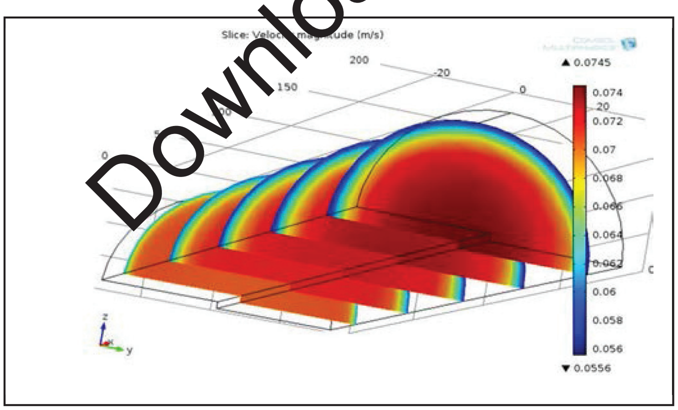

Figure 9. Velocity distribution (finned tube 66.7 l/min)

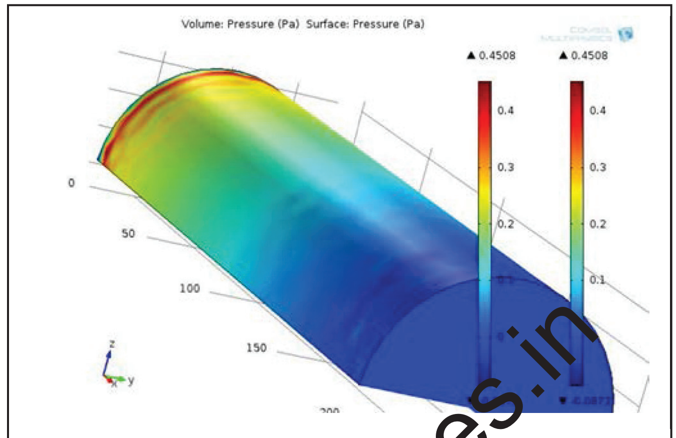

Figure 10. Pressure distrubution (finned tube 66.7 l/min)

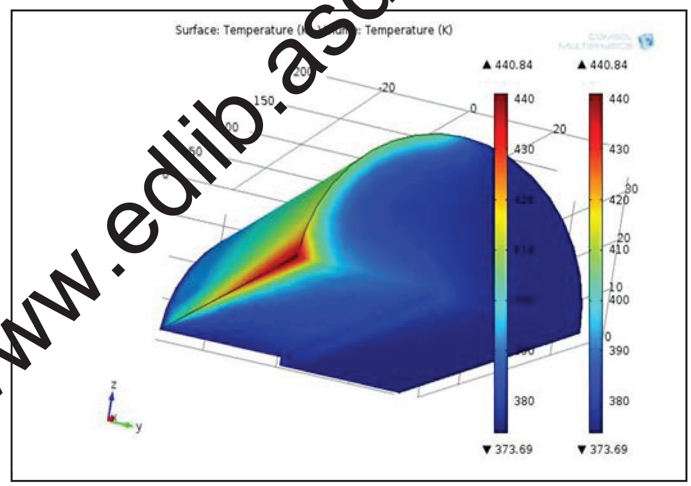

Figure 11. Temperature distrubution (finned tube 66.7 l/min)

For a mass flow of 66.7 l/min the pressure losses is doubled from 0.24 Pa to 0.45 Pa wich mean that we need to optimase any CSP plant to work with a mass flow rate as small as possible to evoid energy losses due to the pumping.

Increasing the mass flow rate will cause a decrease in heat transfer because we minimize the contact time with the stainless steel absorber.

V. CONCLUSION

This paper has analyzed the effect of the utilization of internal finned tubes for the design of parabolic trough collectors with computational fluid dynamics tools. The heat transfer and the fluid flow characteristics was investigated, The performance of the receiver has been improved by insertion of intern fin with pressure as a penalty. Our approach is validated and we can resume:

- For mass flow rate of 47.7 L/min the best absorber is the finned tube that will enhance heat transfer with a slight pressure losses in the order of 0.004 Pa for each absorber.

- For mass flow rate of 66.7 L/min the heat transfer factor is reduced and the pressure losses increases with 0.24 Pa for each absorber. And this lead us to tell that we need to optimize each CSP plant to work with the slowest mass flow rate without increasing the solar field aperture.
- The use of finned tube will increase the plant efficiency by increasing the thermal gain to the HTF. With this efficiency the Levelized Cost of energy will be reduced.

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