Key Aspects of a Novel Undulated Receiver for Parabolic Trough Collectors

Yassine Demagh[1] and Ahmed A. Hachicha[2]

[1] Laboratoire d'Etude des Systèmes Energétiques Industriels (LESEI), Université de Batna 2, Algérie.

[2] Sustainable and Renewable Energy Engineering, University of Sharjah, United Arab Emirates

Abstract

This study proposes the replacement of the conventional straight absorber by the newly designed longitudinally undulated. Numerical results revealed that the new absorber could dethrone the former for several reasons: Among others, it allows a more homogeneous distribution of concentrated solar radiations on its outer surface (Monte Carlo Ray Tracking results); unlike other techniques which improve the inner heat transfer by increasing simultaneously the load of the absorber and the pressure drop within it, the proposed curved absorber is going to generate in a natural way, without any additional mechanical components, vortices within the main streaming which allowed to increase the heat transfer coefficient of about 63 % with an increase of the pressure drop penalty of about 60 %. On the other hand, it allows a drastic reduction of the size of the solar collector field. All these facts lead to decreasing the wall temperature gradient bellow 40 K. Results are obtained for the Syltherm 800 Reynolds number range 2.5×10^4 to 12.3×10^4 and a fluid inlet temperature of 450 K.

Keywords: Parabolic trough collector (PTC), size reduction, undulated pipe, heat transfer enhancement.

1. Introduction

Parabolic trough collectors (PTCs) are the most promising technology for electricity generation and process heating application. The main option to drive the cost of PTC technology down is to reduce the size of the solar field (Price et al. 2002). One of the ways to achieve this goal is the improvement of the thermal performances of the solar absorber by passive techniques by adding additional mechanical components to the absorber pipe (Ghadirijafarbeigloo et al., 2014, Mwesigye et al., 2014); nevertheless, these techniques produce a significant pressure drops penalty. On the other hand, since the work of (Demagh et al., 2015), the heat transfer improvement should be achieved without any additional mechanical parts. In the present study, 3D steady-turbulent simulations are carried out to investigate the scenario where the conventional straight absorber is replaced by the novel undulated absorber (Fig. 1(a)) proposed by (Demagh et al., 2015) and highlight its thermal performances enhancement and effects on the solar collector sizes.

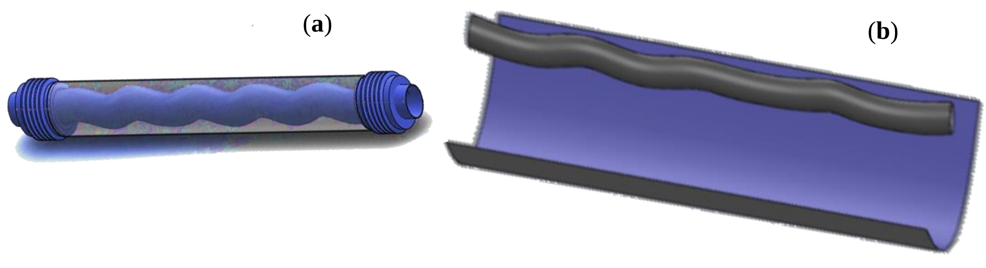

Fig. 1: (a) The novel undulated PTC receiver. (b) A PTC module with the novel undulated absorber.

doi:10.18086/swc.2017.04.02 Available at http://proceedings.ises.org

2. The numerical modelling and results

Fig. 1(b) shows the novel absorber mounted on a PTC module; its main characteristics schematized in Fig. 2 are summarized in Tab. 1.

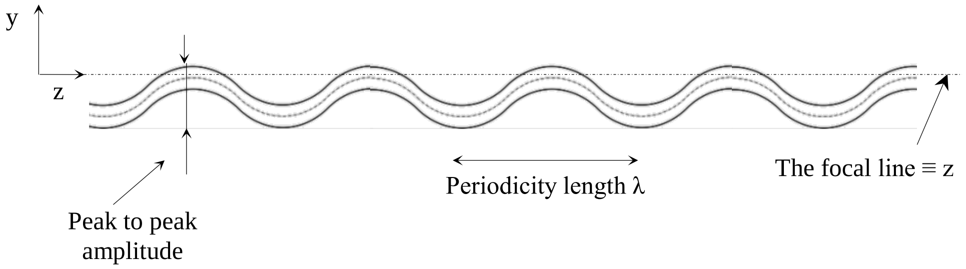

Fig. 2: Main characteristics of the undulated absorber.

Tab. 1: Absorber pipe c haracteristics.

Description	
Outer pipe diameter	0.070 m
Inner pipe diameter	0.066 m
Total length assumed in this study	0.975 m
Periodicity length	0.195 m
Peak to peak amplitude	0.020 m

The periodicity length is not the critical point, but the amplitude is (Demagh et al., 2015). The smaller the length, the more the transfer is efficient. For actual technical considerations of manufacturing, the selected periodicity length of 195 mm could be easy to achieve on a stainless steel pipe with inner/outer diameters of 66/70 mm.

2.1. Modelling and grid independence tests

(Di Piazza and Ciofalo, 2010) compared numerical results (the friction factor and Nusselt number) of different turbulence models with experimental data of curved pipes. The authors concluded that the SST k–ω eddy-diffusivity model gives the best agreement, but requires several computational grid nodes compared to k–ε model. Thus, by means of the CFD code (FLUENT 6.3), the k–ω based (SST) model was adopted to give accurate predictions of the onset and the amount of secondary flow, produced as a result of the curved shape.

The pipe absorber was meshed using tetrahedral elements with a structured mesh into the wall and an unstructured (tetra/mixed) non-uniform grids within the fluid medium, as shown in Fig. 3. During the meshing process, additional nodes are placed inside the viscous sub-layer to ensure the satisfaction of the $y^{+} < 1$ requirement at the first grid point close to the wall.

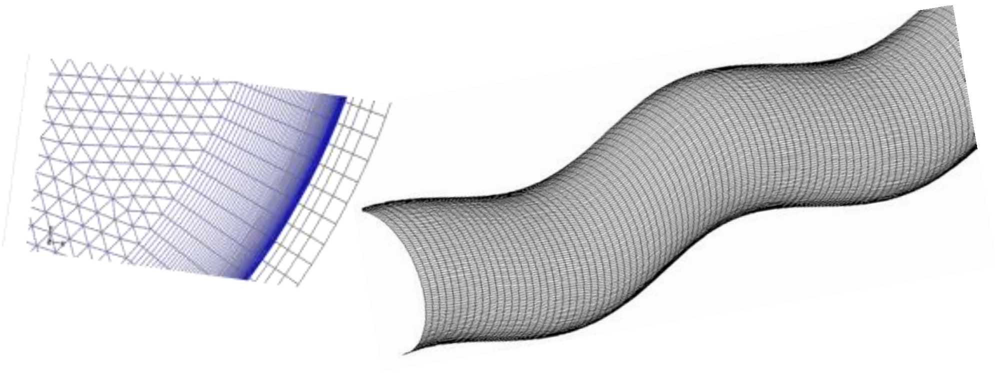

Fig. 3: The meshing generated by GAMBIT.

The grid independence was conducted in the analysis by adopting different grid sizes, and a sample of these tests is summarised in Table 2. The solution is considered mesh independent when the variations of the predicted mean heat transfer coefficient (h) and pressure drop (ΔP) are less than 1.5%. It is found that results for the global grid of 2,910,119 cells is not affected by the refinement and is therefore used in the present study.

Table 2- Grid independence test.

Global grid	% change of ΔP	% change of h
2,300,000	//	//
2,437,600	-1.6%	-0.45
2,910,119	-4.4%	2.4
3,256,400	1.5%	0.55

2.2. Results and interpretation

The Monte Carlo Ray Tracing method (MCRT) provided the non-uniform two-dimensional (2D) of the heat flux density distribution (***q***) on the outer surface of PTC straight absorbers (He et al., 2011). The direct normal irradiance equals $933.7 W/m^2$; the parabolic trough rim angle used was 70°, aperture width 5 m and focal length 1.84 m. By means of the free code-source Tonatiuh, exploiting its validated results (Blanco et al., 2009), the reconstituted 3D heat flux density distribution on the outer wall of the undulated PTC absorber pipe as established by (Demagh et al., 2015) is shown in Fig. 4. With regard to the focal line (Fig. 2), the y-location of cross-section centres changes periodically along the undulated absorber, which is at the origin of the 3D nature of the (***q***) relating to the 2D nature of the conventional straight absorber where the y-location of the cross-section centres remain unchanged (He et al., 2011).

Using the built-in curve fitting functions in Microsoft Excel, a UDF was written and compiled under Fluent GUI to set up the thermal boundary condition on the outer absorber pipe surface. The Heat transfer fluid (HTF) was the Syltherm 800 and its proprieties were considered as a temperature-depending.

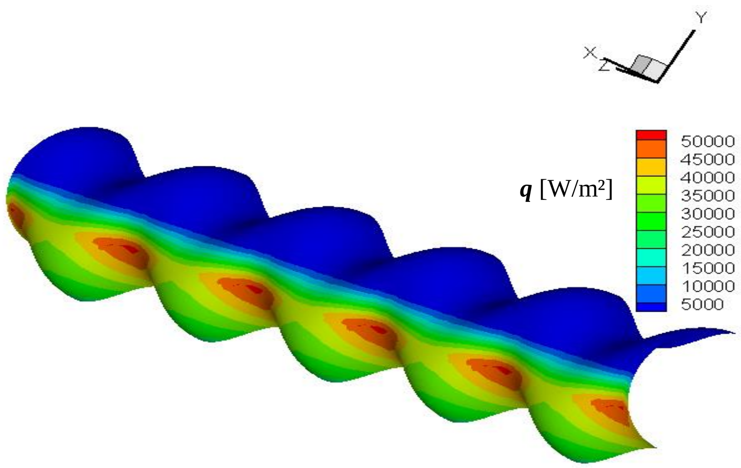

Fig. 4: Contours of the heat flux density distribution *q* [W/m²] on the outer face of the undulated absorber for a direct solar irradiance of 933.7 W/m² (Demagh et al., 2015).

The numerical results for the average heat transfer coefficient and the pressure drop obtained with Syltherm 800 flowing through the undulated PTC absorber are reported in Fig. 5 for $T_{in} = 450\,K$ and the Reynolds number range 2.5×10^4 to 12.3×10^4. It is evident that the heat transfer coefficients of the undulated absorber are larger than that of the conventional straight absorber obtained by Gnielinski's correlation (Incropera et al., 2007), about a 63% increase.

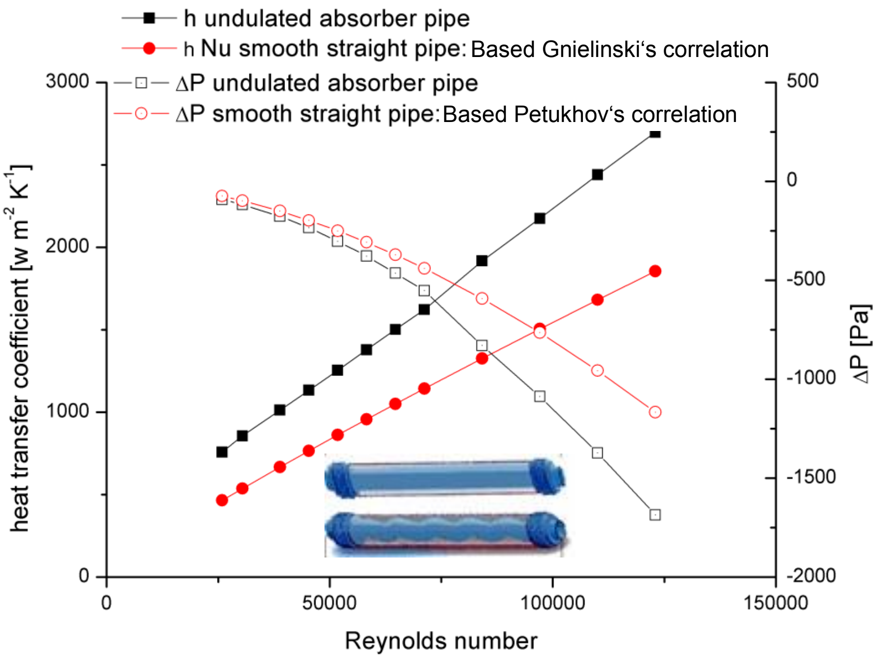

Fig. 5: The convective coefficients and pressure drops vs. Reynolds number. Correlation from (Incropera et al., 2007)

Also, as expected, the enhancement of the heat transfer is accompanied by an increasing of the fluid pressure drops, about 60 %. The better the distribution of ***q*** and the enhancement heat transfer cause the reduction of the circumferential temperature gradient ΔT bellow 40 K, as shown in Fig. 6.

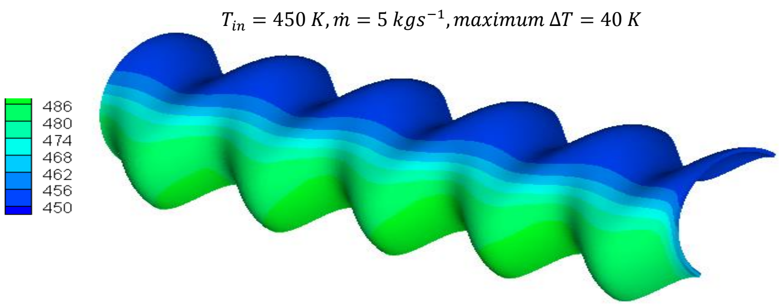

Fig. 6: Contour of the circumferential temperature gradient Δ*T*.

It is well known that curved configurations induce a secondary flow within it due to the centrifugal force, generated by curvatures of the pipe, as the fluid flows. The secondary streaming significantly enhances the heat transfer rate, causing a better mixture of fluid by the disturbance of the boundary layer. Vortices are identified in the bends, as shown in Fig. 7, where the y-velocity colours are synonymous of the intensification of vortices. Outside the bend-planes, the vortices are absent. On the other hand, compared to the conventional straight pipe absorber, the increase of the straight length of the undulated pipe absorber is insignificant, about 2.547 % for a peak amplitude of 10 mm (in this study), and the increase of the geometric concentration ratio will be the same. Thus, the improvement of heat transfer is mainly due to the existence of secondary flows rather than the increase of the heat exchange surface.

Fig. 8 shows the longitudinal change of the local Nusselt number along the second, third and fourth period for an inlet HTF temperature of $450K$ at Reynolds numbers of $\mathrm{Re} \simeq 64740$, corresponding to an HTF mass flow rate of $5\ kg \cdot s^{-1}$.

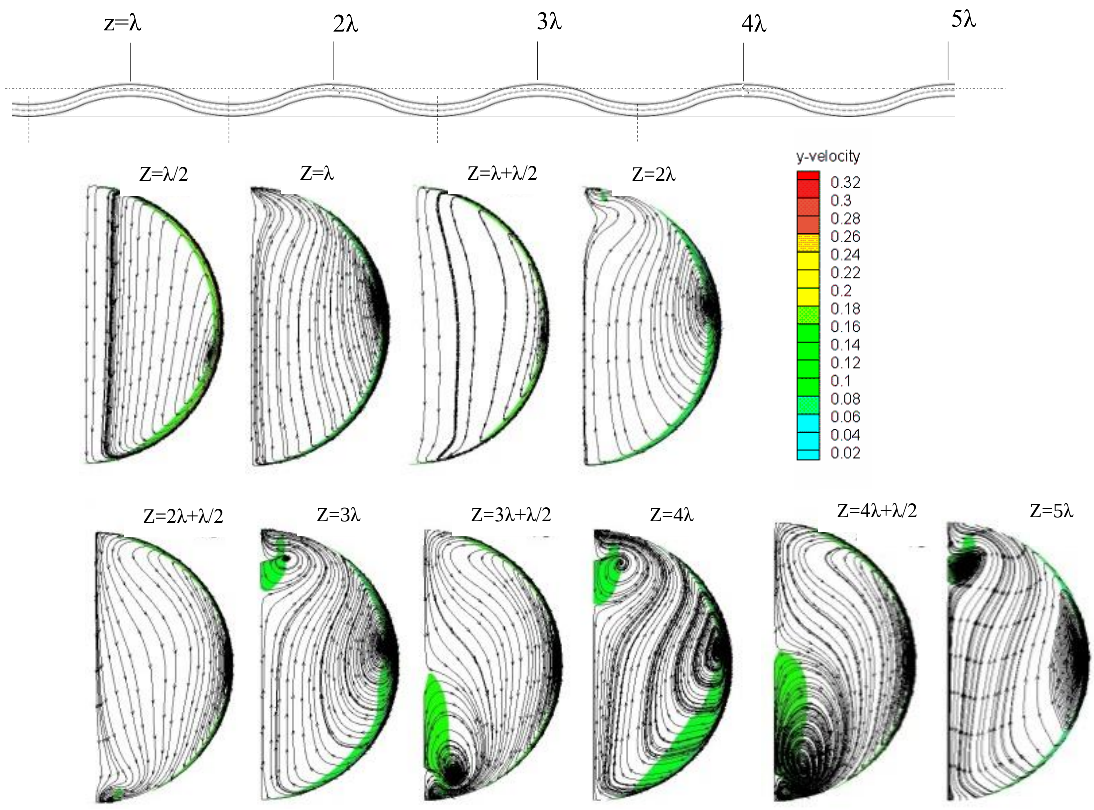

Fig. 6: The Secondary flow configuration. Colours refer to y-velocity magnitude [m/s].

The curve possesses a sinusoidal shape, the local Nusselt number increases up to a location at midway between the uppermost and bottommost bends. From this location, it starts decreasing, exceeding the bottommost bend, up to next midway location of the next periodic segment and increases again, and so on periodically along the entire length of the absorber.

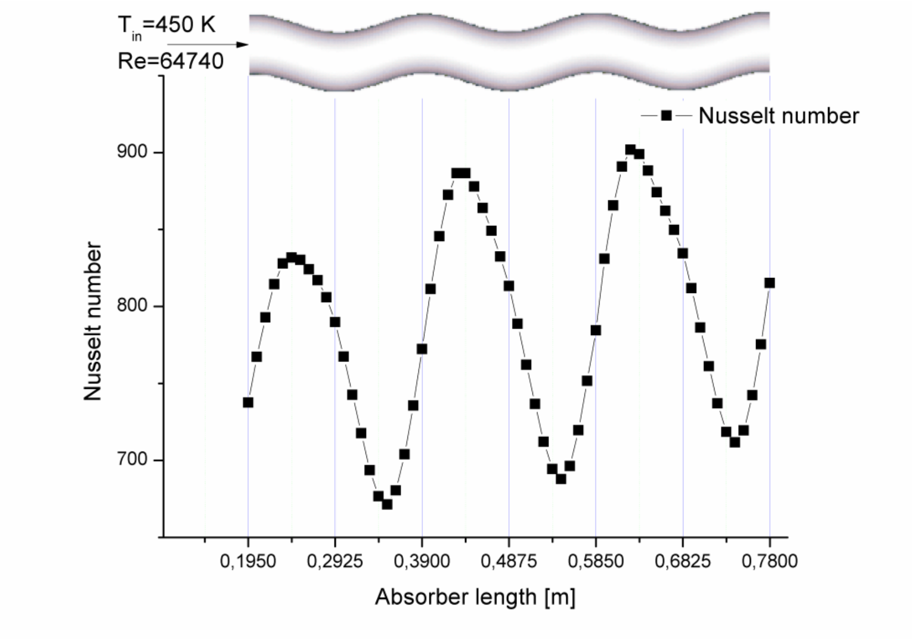

Fig. 8: Change of the local Nusselt number along the absorber length.

All rising branches of the curve occur on the upper half of the absorber pipe where the heat flux density distribution and wall temperature are moderate, see Figs 4 and 6. The HTF temperature will approximates the wall temperature, which it is not heated by the concentrated solar radiation and the local Nusselt number increases. On the other half of the absorber pipe, it's quite the opposite phenomenon, the Nusselt number decreases given that the temperature gap between the HTF and wall increases due to the concentrated solar radiation on this half (Fig. 4) heating the wall of the tube (see Fig. 6). It should be emphasized here that the local Nusselt number seems to respond to the heat flux density distribution to the flow dynamics (acceleration or deceleration in various branches or even the vortex in bends). The general trend of the local Nusselt number is increasing due to the mixture of fluid with each pass through the bends where the vortices occur.

3. Conclusion

For the range of Reynolds numbers, the inlet HTF temperatures and the geometrical parameters considered, it is established that the heat transfer rate may be increased by up to 63% compared with a straight pipe, while the pressure drop increased by less than 60%; the secondary streaming (reversed flow) contribute considerably to the overall heat transfer enhancement. The circumferential temperature difference of the absorber was decreased below 40 K for almost all the range of the mass flow rates and will significantly reduce the thermal stress on the absorber pipe. The local Nusselt number possesses a sinusoidal shape along the pipe absorber, and seems to respond to the heat flux density distribution to the flow dynamics (acceleration or deceleration in various branches or even the vortex in bends). With the 2.547 % increase of the straight length of the pipe absorber, the improvement of heat transfer is mainly due to the existence of secondary flows rather than the increase of the heat exchange surface. The final conclusion is that to achieve the same HTF temperature rise that the conventional straight absorber the length of the undulated absorber would be reduced due to the improvement in the heat transfer rate.

4. References

Blanco, M.J., Mutuberria, A., Garcia, P., Gastesi, R., Martin V., 2009. Preliminary validation of Tonatiuh. SOLARPACES Symposium, Berlin, Germany.

Demagh, Y., Bordja, I., Kabar Y., Benmoussa H., 2015. A design method of an S-curved parabolic trough collector absorber with a three dimensional heat flux density distribution. Sol Energy 122, 873–884.

Di Piazza, I., Ciofalo, M., 2010. Numerical prediction of turbulent flow and heat transfer in helically coiled pipes. Int. J. Therm. Sci. 49, 653–663.

Ghadirijafarbeigloo, S., Zamzamian, A.H., Yaghoubi, M., 2014. 3-D numerical simulation of heat transfer and turbulent flow in a receiver tube of solar parabolic trough concentrator with louvered twisted-tape inserts. Energy Proc. 49, 373–380.

He, Y.L., Xiao, J., Cheng, Z.D., Tao, Y.B., 2011. A MCRT and FVM coupled simulation method for energy conversion process in parabolic trough solar collector, Renew Energy 36, 976–985.

Incropera, F.P., Dewitt, D.P., Bergman, T.L., Lavine, A.S., 2007. Fundamentals of Heat and Mass Transfer. Sixth edition, J. Wiley & Sons.

Mwesigye, A., Bello-Ochende, T., Meyer, J.P., 2014. Heat transfer and thermodynamic performance of a parabolic trough receiver with centrally placed perforated plate inserts. Applied Energy 136, 989–1003.

Price H., Lüpfert E., Kearney D., Zarza E., Cohen G., Gee R., Mahoney R., 2002. Advances in parabolic trough solar power technology. J. Sol. Energy Eng. 124, 109–125.