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NUMERICAL SIMULATION OF HEAT TRANSFER CHARACTERISTICS IN THE ABSORBER TUBE OF PARABOLIC TROUGH COLLECTOR WITH INTERNAL FLOW OBSTRUCTIONS

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ABSTRACT

Absorber tube performance enhancement by using passive techniques is one of the major topics of research in the field of solar thermal power engineering. Earlier studies on ducts revealed that passive augmentation techniques have shown considerable enhancement in heat transfer. Experiments were conducted earlier on Parabolic Trough Collector (PTC) with plain absorber tube without inserts for different flow rate conditions. It was observed that fluid flow rate with 85 kg/hr has shown higher temperature difference for solar flux condition of 850 W/m^2. In the present study, numerical analysis using CFD is performed by using inserts of different cross-section inside the fluid flow path of an absorber tube to study the flow characteristics. The fluid flow and heat transfer phenomena through a 3-dimensional absorber tube with varying heat source are obtained by solving the fundamental governing equations namely: conservation of mass, momentum and energy. Turbulence is modeled using SST k-ω model of closure. The heat transfer and pressure drop is calculated from the studies conducted for a mass flow rate of 85 kg/hr using the ANSYS CFX 12.1 software. The result of the numerical analysis is validated with the experiments carried out with the parabolic trough collector. The numerical study is carried out with triangle, inverted triangle and semi-circular inserts and compared with that of plain absorber tube. It has been found that triangle insertion gives optimized results with respect to uniform heat transfer which reduces the thermal fatigue even the pressure drop is relatively high compared to the plain absorber tube without insertion.

Keywords: absorber tube, flow obstruction, triangular turbulator, turbulence heat transfer.

INTRODUCTION

In India, progress of solar thermal research is fast booming over the last decade. The success of solar concentrators in India, especially for community cooking applications has given motivation to use these technologies for other applications too. Currently, as per the recent report released by MNRE Report, solar power plants are being planned to setup at a few locations throughout the country [1]. Parabolic collector trough (PTC) technology has been widely used in many parts of the world for power generation activities because of its matured technology. Higher thermodynamic efficiency can be achieved for such type of concentrators. Efforts have been made to increase the absorber characteristics, reducing the heat loss from the collector, extending heat transfer ratio, increasing the fluid conduit aspect ratio and optimizing the design parameters. Absorber tubes of the PTC system receive the concentrated energy from the reflective mirrors and deliver it to the flowing fluid inside. Mafizul hug and Michael K. Jensen *et al*. [2, 3] has revealed that the heat transfer coefficient of the internally finned tube large in the entry region and then the enhancement in heat transfer in the fully developed region due the effects of the internal fins. R. Forristall and Sane *et al*. [4, 5] done the modelling and the heat transfer analysis of a parabolic solar trough collector using fluent and suggested many concepts towards the performance enhancement of the of an absorber tube of a cylindrical solar parabolic trough collector. Earlier many researchers had done numerous simulations, involving both experimental as well as numerical analysis, and gave a conclusion that heat enhancement is possible with the use of passive augmentation techniques. Insertion of passive elements in the fluid flow path has been proved to be a low cost method of improving the heat transfer performance, since the flow characteristics are performance, since the flow characteristics are changed Abu-Khader *et al*. [6]. The performance of a Solar Collector system is largely dependent on the radiation which provides us with an altogether different set of problems because of the variable nature of the Solar Radiation values, when compared to conventional heat exchanger systems. Various radiation models have been suggested by different authors based on the method of estimating the diffused radiation components. Qiu-Wang Wang *et al*. [7] investigated the effect of different fin profiles on the absorber tube flow characteristics of an internally finned tube. O. García-Valladares and N. Velázquez [8] conducted numerical analysis to determine whether the use of concentric circular heat exchangers improved the functioning of parabolic trough solar collector. Munoz and Abanades *et al*. [9] mentioned the heterogeneous nature of flux distribution on the absorber wall and indicated how the use of internally finned tubes would results in homogenized temperature distribution on the absorber wall. The use of these fins also resulted in decrease in outer surface temperature and thermal losses. Matthew Roesle *et al*. [10] conducted numerical analysis for heat loss from the absorber tube of a parabolic trough solar collector. Sekhar *et al*. [11] performed experimental simulation to study the heat transfer enhancement using Al_2O_3 nanofluids in a plain tube with twisted tape inserts. Thundil *et al*. [12] had earlier performed experimental and numerical analysis to study the performance of the absorber tube for a concentric tube insertion configuration and it was observed that there is slight increase in outlet temperature with a huge penalty in pressure drop. This study is intended to estimate the heat transfer performance under uniform heat flux boundary condition in the turbulent flow range for various cross-sections of insertions using CFD analysis.

EXPERIMENTAL SETUP

A prototype PTC system is installed at the terrace of SMBS building, VIT University Vellore campus as shown in Figure-1. The dimensions of the same parabolic trough collector and absorber tube are considered for the present analysis as shown in Table-1. Absorber tube material used is made of copper.

Figure-1. Solar parabolic trough collector.

Table-1. Dimensions of parabolic trough collector.

Width of collector	(W)	0.91 m
Length of collector	(L)	1.50 m
Outer diameter of absorber tube	(D_o)	0.0125 m
Inner diameter of absorber tube	(D_i)	0.0115 m
Reflectvity of receiver	(K)	0.85
Rim angle	($Ø_r$)	90^0
Focal length	(f)	0.2275 m
Concentration ratio	(C)	22.85

Material properties of the tube are considered as mentioned below:
Density = 8978 kg/m^3
Specific heat capacity = 381 J/kgK
Thermal conductivity = 387.6 W/mK
Water is taken as the working fluid in the analysis and the properties are shown in Table-2.

Table-2. Properties of the working fluid (water).

Properties of fluid (water)@ 20^0C		Value
Density of fluid	(ρ)	998.2 kg/m^3
Specific heat capacity	(C_p)	4182 J/kg K
Thermal conductivity	(k)	0.600 W/m K
Dynamic viscosity	(μ)	0.001003 kg/m-s

METHODOLOGY OF CFD SIMULATION

The numerical analysis using CFD is carried out with plain absorber tube and tube inserted with different insertion cross sections are as shown in Figures 2-4, respectively. The fluid flow simulation is accomplished using commercial CFD software Fluent 12.1. Meshing of the Model has been done using ICEM CFD software V 12.1. Some assumptions have been made during the conduct of experimental simulation and are listed as follows:

It has been assumed that the tube will be subjected to two boundary conditions- upper part of 270^0 will be exposed to ambient direct radiation and lower part 90^0 will be exposed with concentrated radiation. The upper part of the tube is assumed to receive a constant direct beam radiation of 870.5 W/m^2 as measured manually at VIT University on 19th January 2013 using Pyrheliometer an d the remaining 90^0 receive constant concentrated radiation of 28266.3 W/m^2 calculated as per the equation reported by Cheng *et al*. [8];

(1) The flux distribution is assumed to be uniform over the surface of the tube;
(2) The fluid flow is assumed to be fully developed;

Assumptions (1) and (2) have been arrived at by considering information available in existing literature [8, 9, 11]. The fluid flow has been considered a s 85 kg/hr and the temperature of fluid at inlet of the absorber tube has been considered based on an experimental values. Mesh generation of the 3-D model of the absorber tube is then done using the pre-processor ICEM CFD meshing tool. The entire model is discretized using hexahedral mesh elements which are accurate and involve less solver effort and requires high skill of meshing algorithm compared to tetrahedral elements. The hexahedral meshing of the fluid domains of semi-circular insertion and inverted triangle insertion are shown in Figures 4 and 5, respectively. The Y^+ value is maintained less than 1 near the bottom wall surface where the heat flux is higher. This Y+ value is required to capture the turbulence physics exerted by the SST K - ω model of closure. Fine control on the hexahedral mesh near the wall surface allows capturing the thermal and velocity boundary layers accurately. The discredited model is checked to have a minimum angle of 79.6 and a minimum determinant quality of 0.9. The inverted triangle insertion fluid domain is meshed with 196455, 354712 and 477945 hexahedral elements and it has been found that the numerical solution

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obtained do not vary significantly beyond 354712 hexahedral elements. Similar grid independence study is carried out for the geometry considered in this analysis.

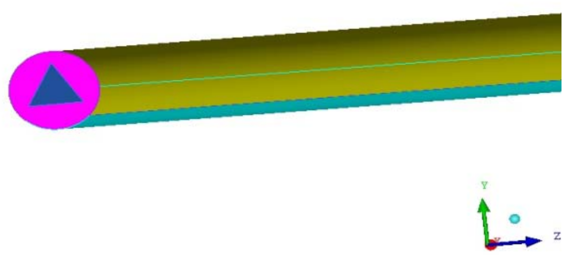

Figure-2. Triangular insertion.

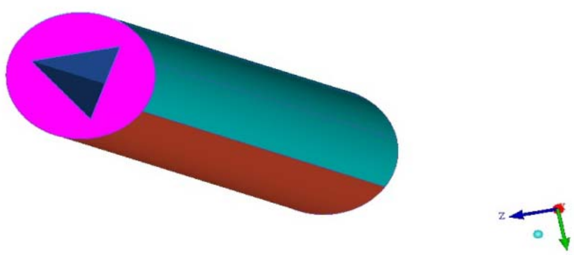

Figure-3. Inverted triangular insertion.

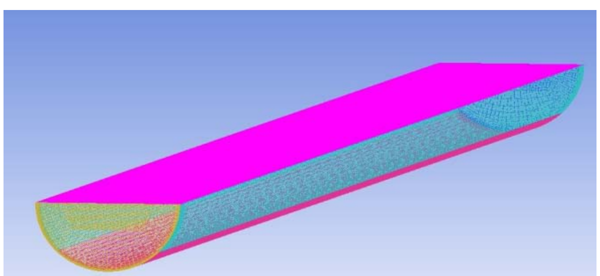

Figure-4. Semicircular insertion.

The following assumptions are made for CFD analysis:

a. Steady state heat transfer is considered so that the heat flux at the wall does not change.
b. The contact thermal resistance between the wall and the fluid is not considered.
c. Thermal conductivity of the absorber tube material is uniform and constant.
d. The radiation heat transfer from the absorber tube is neglected.

Pre-processing

The fluid flow is in turbulent and in steady condition. So the governing equations for continuity, momentum, energy and standard k-ε turbulence model can be expressed as follows:

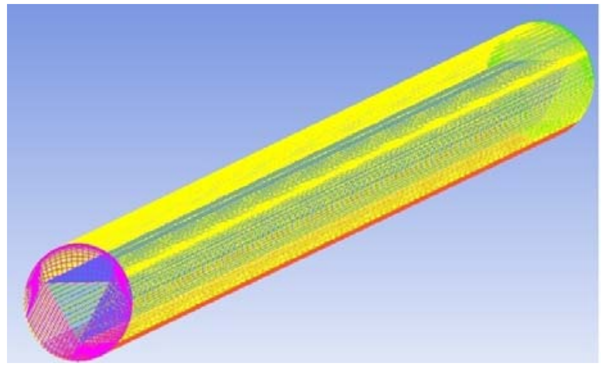

Figure-5. Hexahedral Mesh of the absorber tube with insertion.

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i}\left[\left[(\mu_t + \mu)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right]\right]$$

$$-\left[\left(\frac{2}{3}\right)(\mu_t + \mu)\frac{\partial u_l}{\partial x_l}\delta_{ij})\right] + \rho g_i \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left[\left(\frac{\mu}{pr} + \frac{\mu_t}{\sigma T}\right)\frac{\partial T}{\partial x_i}\right] + Sr \tag{3}$$

K equation:

$$\frac{\partial}{\partial x_i}(\rho u_i k) = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu T}{\sigma k}\right)\frac{\partial k}{\partial x_i}\right] + Gk - \rho\varepsilon \tag{4}$$

ε equation:

$$\frac{\partial}{\partial x_i}(\rho u_i \varepsilon) = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu T}{\sigma\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_i}\right] + \left[\frac{\varepsilon}{k(C_1 Gk - C_2\rho\varepsilon)}\right] \tag{5}$$

Where the turbulent viscosity μ_t and the production rate Gk are given by:

$$u_t = \frac{C_\mu(\rho)(k^2)}{\varepsilon} \tag{6}$$

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$$Gk = \mu_t * \left(\frac{\partial u_i}{\partial x_j \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)} \right) \qquad (7)$$

In order to improve the quality of the analysis second-order upwind analysis has been done so as to obtain more accurate results. The SIMPLE algorithm has been used to ensure coupling between velocity and pressure.

RESULTS AND DISCUSSIONS

Experiments were conducted in the month of January 2013, on the prototype PTC system with water as working fluid using plain absorber tube. The experimental setup used in the earlier analysis is a closed circuit configuration. The parameters measured during the experimentation are inlet temperature, outlet temperature, ambient temperature, inlet mass flow rate and the pyrheliometer reading. The beam radiation values were continuously measured and the variation of the radiation during the duration of the experiment is within the allowable limit (±50 W/m^2). The results of the experiment conducted for different flow rates confirm that for water mass flow rate of 85 kg/hr the temperature difference between outlet and inlet fluid temperature is maximum as shown in Table-3. Hence, using the same flow parameters CFD simulations have been carried out for absorber tube with insertion of different cross section to analyze the flow characteristics. The insertions having the cross sections of triangular, inverted triangle and semicircular are considered in the analysis. The boundary conditions assumed for the model are ambient temperature as 307.157 K and pressure at outlet is 101325 Pa, respectively. The values of the boundary conditions like operating temperature, mass flow rate of water are taken from the earlier experimental work [13]. Other boundary conditions like density, specific heat, thermal conductivity and other material properties are considered as constants throughout the analysis. Initially the absorber tube without insertion is numerically solved and validated against the same boundary conditions as that of the experimental setup and a very good correlation is obtained between the numerically predicted results and the experimental setup as reported in our previous work [13]. upon gaining confidence with validation the CFD tool is further extended to analyze numerically absorber tube with triangle, inverted-triangle and semi-circle insertions. The discretized fluid domain is imported to ANSYS CFX 12.1 pre processor and the boundary conditions and turbulence model are given. The top and bottom of the absorber tube surfaces are given constant wall fluxes of 870.5 W/m^2 and 28226.3 W/m^2, respectively. In the case of the model with the insertion, the wall of the insertion is assumed to be adiabatic. The water inlet temperature is set at a temperature of 307.15 K which is obtained from experimental data. The solver is made to have a convergence of 10E-06 for mass, momentum and turbulence and set at 10E-08 for energy.

Table-3. Temperature data of experimental set up.

Mass flow rate (kg/hr)	**Average inlet temperature** (°C)	**Average outlet temperature** (°C)
85	34	38.6
63	35.8	40
33	37.4	41.4

The heat transfer takes place from the wall surface of the absorber tube to the fluid. The inlet of the absorber tube is given Mass Flow Inlet Boundary Condition with the mass flow rate set at 0.023611kg/s, as used in the experimental analysis. The results of the CFD analysis for plain tube is compared with the experimental results and found to be in good agreement with a deviation of less than 5%, thus validating the present CFD analysis.

The wall temperature distribution along the length of the absorber tube for plain tube and tube with insertions is shown in Figures 6-9, respectively. It can be observed from Figure-6 that the radial temperature distribution between the top and bottom layers of the tube is not uniform from the inlet to outlet section. Also the wall temperature is maximum near the inlet region, on the wall surface receiving concentrated radiation. The wall temperature decreases along the axis of the tube on the same surface due to heat transfer between the wall surface and the fluid. The wall temperature is lower in the case of the absorber tube with insertion than that of the absorber tube without insertion.

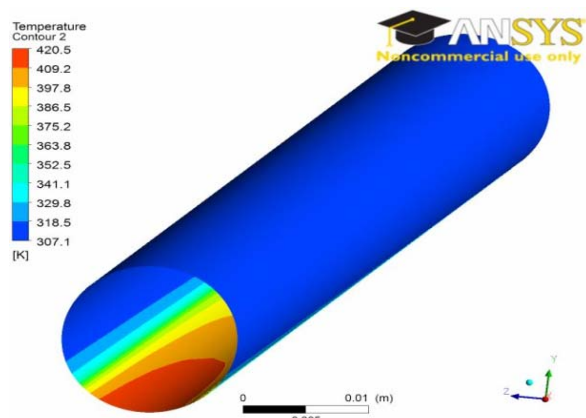

Figure-6. Wall temperature distribution of absorber tube without insertion.

The Figures 10-12 show more uniform temperature distribution between the top and bottom layers of the tube along its entire length for the absorber tube with triangle insertion compared to plain tube and tube with other type of insertions. The maximum temperature in case of plain tube is 420.5 K and minimum is 307.15 K

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and hence there is no uniform temperature distribution as shown in Figure-10. Because of this huge temperature difference the thermal loading will be more and hence the life time of the plain tube would be lesser compared to other type of insertions. Even though the temperature difference in case of semi-circular insertion is low, but the heat absorbing capacity is relatively low compared to plain and other type of absorber tubes with insertions and the maximum temperature of the wall is limited to around 335 K as shown in Figure-9. The pressure drop of the semi-circular insertion is 274 Pa as shown in Figure-17 which is relatively higher for any type of absorber tube considered in this analysis. The pressure drop for the triangular and inverted triangle are almost the same with 147 Pa which is relatively higher compared to plain absorber tube with 48 Pa as seen in the Figures 14 to 16. Even though the pressure drop with triangular insertion is high compared to plain absorber tube it is preferred because of uniform temperature distribution. Compared to triangular and inverted triangular insertions absorber tube both have the same pressure drop and heat gain characteristics, but the thermal loading is more uniform for triangular insertion compared to inverted triangle absorber tube.

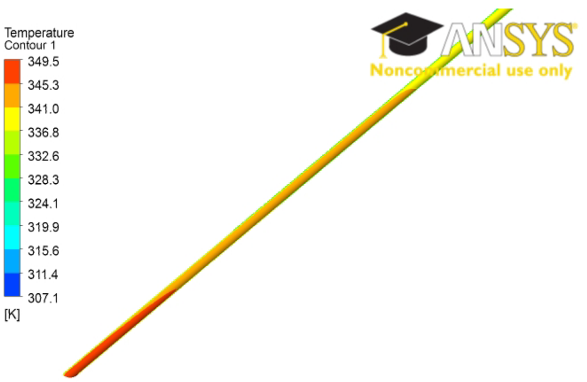

Figure-7. Wall temperature distribution on the absorber tube with triangular insertion.

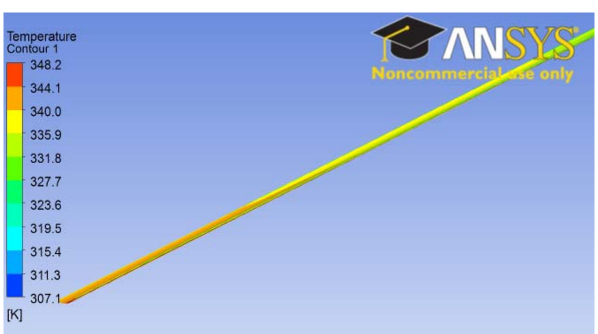

Figure-8. Wall temperature distribution of absorber tube with inverted triangular insertion.

Figures 10-13 show the temperature distribution at the outlet section for plain as well as tube with different insertions. It can be observed from Figure-10 that the bottom layers of the tube experience high thermal loading compared to the top layers due to which the tube may easily fail due to non-uniform thermal loading and the product life could be too short. If tube with insertions are considered the thermal loading is almost uniform except for inverted triangular insertion where the bottom layers experience some non uniformity. But, the magnitude of non uniformity is minimized. Also, it can be observed from Figures 11 and 12 that the average outlet temperature is almost constant.

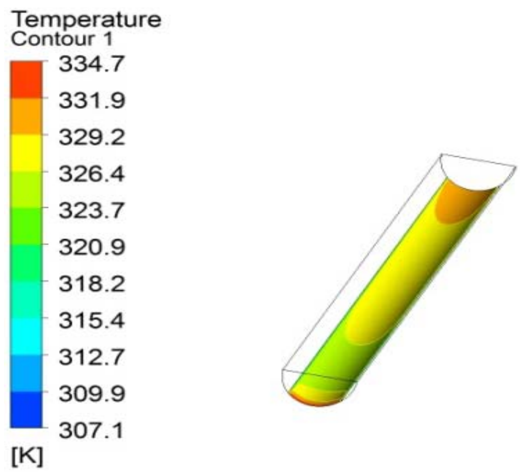

Figure-9. Wall temperature distribution of absorber tube with semicircular insertion.

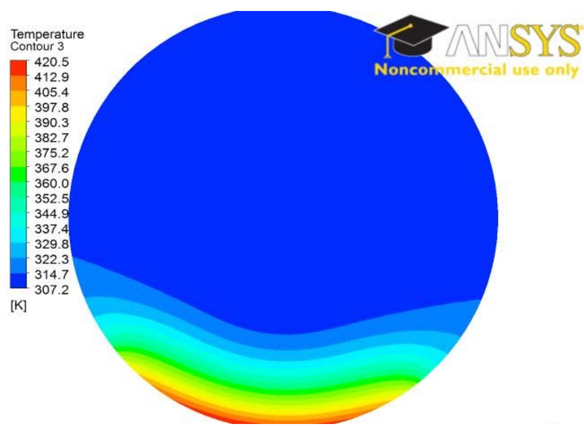

Figure-10. Outlet temperature distribution for absorber tube without insertion.

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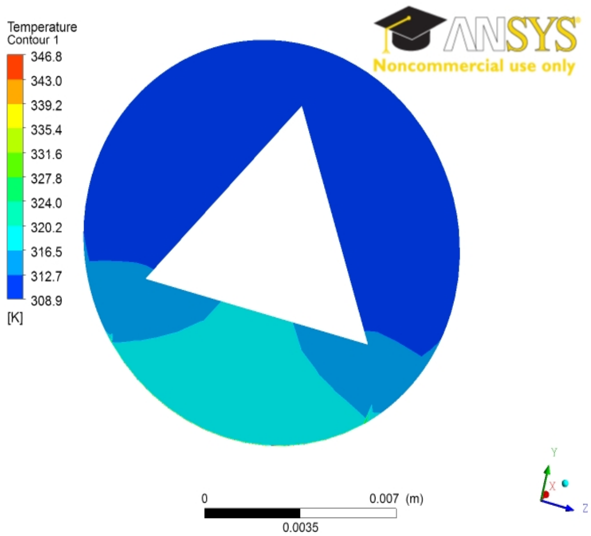

Figure-11. Outlet temperature distribution for absorber tube with Triangular insertion.

The variation in magnitude of temperature distribution around the insertion could be due to the inside convection currents occurring during the fluid flow. The temperature distribution at the outlet for semicircular insertion shown in Figuire-13 confirms that the bottom fluid layers of the tube have very low temperature as compared to other insertions. This is because the low heat flux in the top surface of the absorber tube is zero and hence the total heat source to the semicircle insertion type is much lesser than the other configurations. It can also be observed from Table-4 that considerable difference in outlet temperatures is not observed for plain tube and tube with insertions.

Table-4. Temperature Details of Plain absorber tube and with various insertions.

S. No.	Type of absorber tube	T_{in} (K)	T_{out} (K)
1	Plain tube	307.157	312.328
2	Triangle insertion	307.157	312.313
3	Inverted triangle	307.157	312.321
4	Semi-circle	307.157	312.006

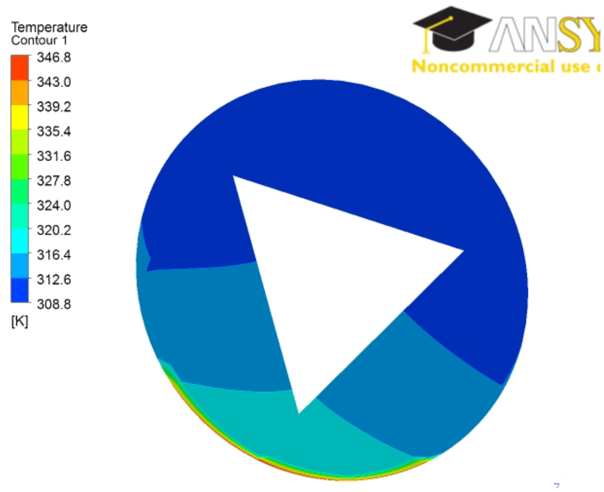

Figure-12. Outlet temperature distribution for absorber tube with inverted Triangular insertion.

CONCLUSIONS

1) The 3-dimensional numerical analysis is able to predict the fluid flow and heat transfer characteristics through the absorber tubes with and without insertions.
2) No appreciable changes in temperature are obtained between plain absorber tubes and absorber tubes with different types of insertions.
3) The thermal stresses on the absorber tube with insertion is lower than that on the absorber tube without insertion and it is found that triangular insertion is better compared to other types of insertions.

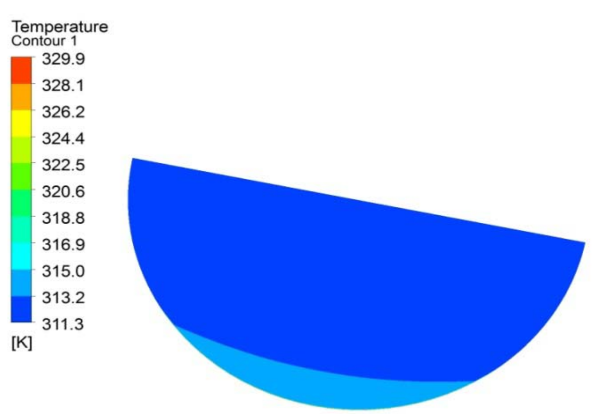

Figure-13. Outlet Temperature Distribution for Absorber Tube with semi circular insertion.

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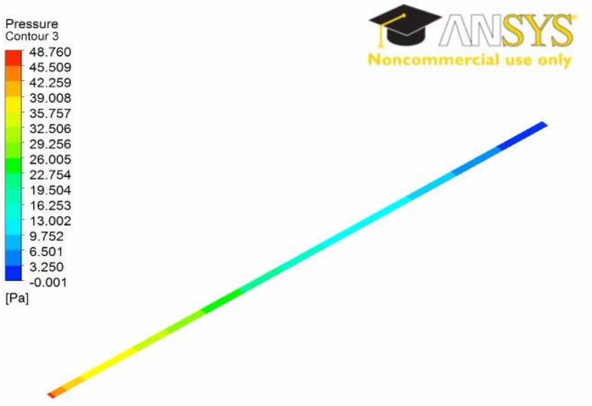

Figure-14. Pressure drop profile of the absorber tube without insertion.

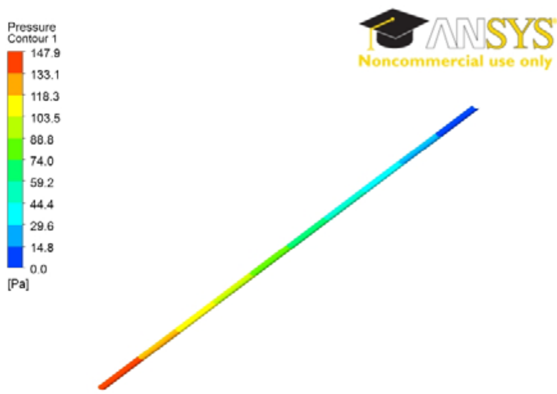

Figure-15. Pressure drop profile of the absorber tube with inverted triangular insertion.

4) Even though the pressure drop in the absorber tube with triangular insertion is 147 Pa while that in the absorber tube without insertion is 48 Pa, the absorber tube with triangular insertion is preferred due to more uniform temperature distribution which leads to better life time of the product.

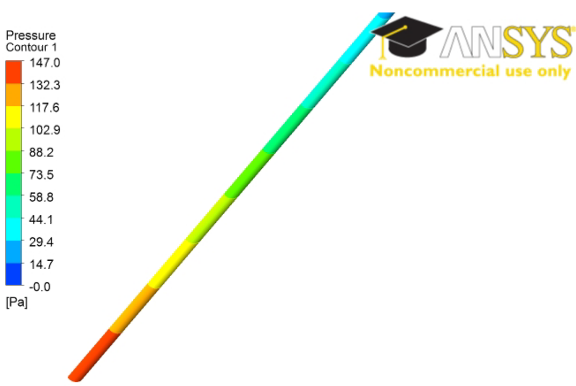

Figure-16. Pressure drop profile of the absorber tube with triangular insertion.

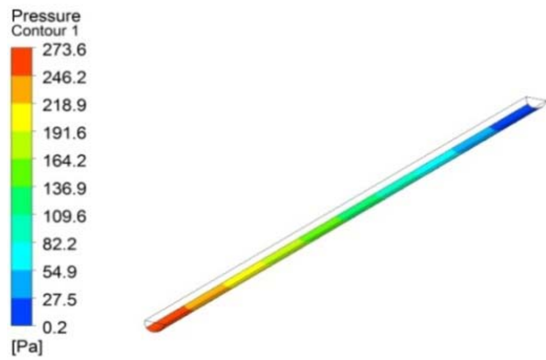

Figure-17. Pressure drop profile of the absorber tube with semi circular insertion.

5) The transient numerical study can be further extended to establish the real physics taking place in the absorber tubes before it reaches equilibrium which would give more insight for optimizing other types of insertions.

Nomenclature

I_b Direct normal irradiance (W/m^2)
W Width of the parabolic trough collector (m)
L Length of the absorber tube (m)
D Diameter of the absorber tube (m)
m Mass flow rate (kg/s)
T Temperature (K)
u, v, w x, y, z velocity components (m/s)
C Concentration ratio
K Reflectivity of collector system
x, y, z Cartesian co-ordinate system

Greek symbols		
Μ	Dynamic viscosity	(kg/m-s)
μ_t	Turbulent viscosity	(kg/m-s)
Κ	Thermal conductivity	(W/mK)
Ε	Turbulent dissipation rate	
σ_T	Turbulent Prandtl number	
Ν	Kinematic viscosity	(m^2/s)
Ρ	Density	(kg/m^3)

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