PERFORMANCE ANALYSIS OF PARABOLIC TROUGH COLLECTOR TUBE WITH INTERNAL INTERMITTENT FINS

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Abstract - *In this study, a numerical simulation of turbulent flow and heat transfer in PTC receiver tubes with internal intermittent fins have been investigated. The results show that inclusion of intermittent fins in PTC pipes results in 60 to 140 percentage increase heat transfer in lower turbulent region. The efficiency is found to be 40 to 80 percentage greater than that of plain tube when considering pressure drop. As the turbulence is increased, the heat transfer efficiency of the pipes with fins of higher pitch is found to be greater than that of fins with lower pitch. From this simulation, it is found that intermittent fins 1d pitch has maximum efficiency up to a Reynolds number of 100000. In higher turbulence region, the efficiency of pipe with intermittent fins of relatively higher pitches has higher efficiency.*

Key Words: Fins, Monte Carlo Ray Tracing, Parabolic Trough Collector

1.INTRODUCTION

Energy production and management are essential problems of our society because of the existence of many environmental problems such as air pollution, global warming, climate change and water pollution. The conventional energy sources, such as fossil fuels (oil, coal and natural gas), will be depleted in some decades and creates high carbon dioxide emissions. Thus, the use of renewable energy sources seems to be one of the most promising solutions in order to produce clean energy. Solar energy is one of the largest renewable energy resources. Solar energy is the most promising renewable technology because this energy source is able to be used in numerous applications and it has great availability. The most usual applications which exploit the solar energy are space heating/cooling, industrial heat production and electricity production. Power generation systems based on parabolic solar collectors is well established and commercialized all over the world in the past two decades. 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. The aim of the current study is to examine the thermal performance of solar parabolic trough collector with internal intermittent fins attached to it. The numerical simulation is implemented by using Computational Fluid Dynamics (CFD). The effect of different internally finned geometries is analysed using computational tool.

1.1 Literature Review

Ghasemi et. al. [1] numerically investigated the performance of parabolic solar collector with three segmental rings. This numerical simulation is implemented for a constant distance between three segmental rings, the results show that use of three segmental rings in tubular solar receiver enhances the Nusselt number and system performance. By decreasing the inner diameter of three segmental rings, the Nusselt number increases, but with considering the pressure loss, thermal performance decreases. Suresh et. al. [2] conducted an experimental investigation on the convective heat transfer and friction factor characteristics in circular tube with spiralled rod inserts (pitch = 15 mm, 30 mm) under turbulent flow with constant heat flux is carried out with distilled water and Al_2O_3/water Nano fluids. It is found that (i) heat transfer enhancement is caused by suspending nanoparticles and becomes more pronounced with the increase of the particle volume concentration (ii) the Nusselt number for spiralled rod inserts under turbulent flow showed an increase of about 10–48% compared to the Nusselt numbers obtained with plain tube (iii) the isothermal pressure drop for turbulent flow with spiralled rod inserts were found to be between 2 and 8% higher than the plain tube.

Cheng et. al. [3] calculated the solar energy flux distribution on the outer wall of the inner absorber tube of a parabolic solar collector receiver by adopting the Monte Carlo Ray-Trace Method (MCRT Method). It is revealed that the non-uniformity of the solar energy flux distribution is very large. Huang et. al. [4] conducted a numerical simulation on the fully developed turbulent flow and heat transfer in the inner tube with and without helical fins, protrusions and dimples. The results show that the receiver tubes with dimples have superior performance of heat transfer augmentation compared with that with protrusions or helical fins. Bellos et. al. [5] investigated the utilization of internally finned absorbers in LS-2 PTC module for various operating conditions. Twelve different longitudinal fins are tested and compared with the smooth case. The analysis is performed with SolidWorks Flow Simulation, using a validated model by literature results. Generally, it is proved that both greater length and thickness lead to higher thermal enhancement and to higher pressure losses. Various methods are presented for evaluating together the thermal efficiency or Nusselt number enhancement versus the increase in pressure drop or in the friction factor. Taking into consideration four different criteria, the absorber with 10 mm fin length and 2 mm fin thickness is found to be the overall optimum case. For this case, the thermal efficiency is enhanced about 0.82%, the Nusselt number increase 65.8%, while the friction factor and the pressure losses are about the double compared to the smooth case.

1.2 Problem Definition

Many researches have been conducted in PTC tube to increase its heat transfer characteristics and thermo hydraulic performance by adding heat transfer area and turbulent generators into the pipe. Many of the researches concentrated on any one of these methods. Intermittent fins are a Novel idea whereby which we can increase the heat transfer in flow channels by its increased surface area(fins) and numerous turbulence generation points (due to the intermittent placement of fins). The main aim of this project to study the thermo-hydraulic performance of PTC tube with internal intermittent fins. The dimensions of fin are taken from bellos et. al. [5] are 10mm in length and 2 mm in thickness. The simulation is conducted for PTC tube with

1. intermittent fins with pitch spacing of 0.5d, 1d, 1.5d, 2d where d- internal diameter of tube
2. Reynolds number of 30000, 50000, 100000, 150000 and 200000

2. NUMERICAL MODEL

2.1 Geometry

In this study, the solar collector under consideration is LS-2 module. The tube in the module is made of stainless steel. The concentration ratio of the module is equal to 22.74 and aperture area is 39m^2. Only evacuated tube part is considered for thermo-hydraulic performance evaluation. The tube is made up of stainless steel and have 66 mm inner diameter (d) and 70 mm outer diameter (D). The length of the LS-2 model is 7.8 meter. For the sake of the analysis, only 2 meter of length of the pipe is considered.

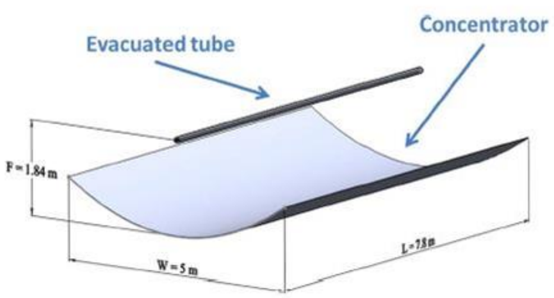

Fig -1: LS-2 PTC Module

The tube of the PTC is modified in order to create internal intermittent fins. The internal fins are placed at the lower section of the tube as lower portion of the tube gets more solar radiation than the upper portion. The cross section of fins is rectangular with dimensions r= 10 and t= 2. The length of the fin is taken to be l= 0.5d. The parameter which is varying in this analysis is pitch length between two consecutive fins. Total 5 geometries are considered for analysis (Plain Tube, Pitch p-0.5d, 1d, 1.5d & 2d).

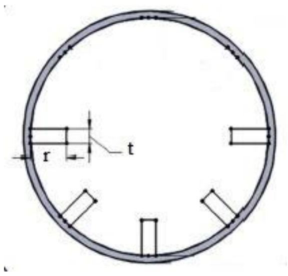

Fig -2: Rectangular Fins inside the Tube

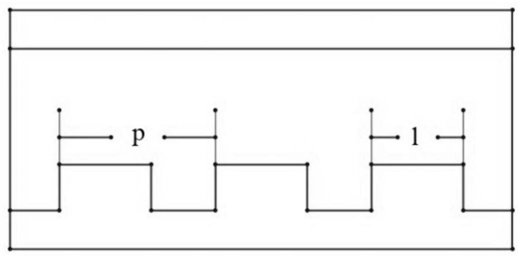

Fig -3: Intermittent Fins inside the Tube

2.2 Governing Equations

Using Ansys Fluent 16.0, governing equations for flow conditions and heat transfer were numerically solved based on following assumptions: steady, incompressible and turbulent flow with constant thermo-physical properties of the fluid. The resulting governing equations are:

Mass Conservation

$$(\nabla . V) = 0$$

Momentum Conservation

$$\rho_{nf}(\nabla . V)V = -\nabla P + \mu_{nf}\nabla^2 V$$

Energy Conservation

$$\rho_{nf}C_{pnf}(V.\nabla)T = k_{nf}\nabla^2 T$$

PTC utilizes only the solar beam irradiation (Gb) and thus the available solar irradiation on the collector aperture (Qs) is calculated as:

$$Qs = Aa \cdot Gb,$$

The useful heat (Qu) captured by the fluid is calculated according to the energy balance in the fluid volume:

$$Qu = m \cdot cp \cdot (Tout - Tin),$$

The heat transfer coefficient (h) between absorber and fluid can be calculated by below given equation. In this equation, the useful heat (Qu), the receiver mean temperature (Tr), as well as the mean fluid temperature (Tfm) are known by the computational tool.

$$h = \frac{Qu}{\pi DriL(Tr - Tfm)}$$

The mean fluid temperature can be estimated as:

$$Tfm = \frac{Tout + Tin}{2}$$

The Nusselt number (Nu) is calculated using the heat transfer coefficient:

$$Nu = \frac{hDri}{K}$$

The friction factor (f) is calculated according to Eq. if the pressure losses along the absorber tube are known

$$f = \frac{2D\Delta P}{\rho V^2 L}$$

The theoretical value for the friction factor can be estimated according to the below given equation. This equation is valid only for the smooth absorber case and it can be used only for the validation of the developed model

$$f\,th = [0.79 \cdot \ln(Re) - 1.64]^{-2}$$

It is important to state that for all the examined cases of this study, the Reynolds number is over the typical limit of 2300 and thus the flow is assumed to be turbulent. The theoretical Nusselt number for these flow conditions can be estimated for circular tubes as

$$\text{Nusselt Number } Nu = \frac{\frac{f}{8} Re\, Pr}{1.07 + 12.7\left(\frac{f^{\frac{1}{2}}}{8}\right)(Pr^{\frac{2}{3}} - 1)}$$

Performance Evaluation Index is used for comparing the performance and efficiency of the models. This index is modified in order to compare the fin cases under equivalent operating conditions. The most commonly used thermal enhancement index is the evaluation of the cases under identical pumping work. In this case, the thermal enhancement index PEC is given as

$$PEC = \frac{\frac{Nu}{Nu0}}{\frac{f}{f0}^{\frac{1}{3}}}$$

3. CFD SIMULATION

In this study, the examined models of PTCs (Plain Tube, Pitch p-0.5d, 1d, 1.5d & 2d) were designed in Ansys and it was simulated in its CFD program, Fluent. This tool is able to perform simultaneous thermal and hydraulic analysis. In order to conduct the CFD analysis, the user has to select a great number of parameters in the program environment. Below, the main parameters are described with details in order to make clear the way that the simulations have been performed. It is essential to state that the present simulation tool has been also used in a great variety of literature studies; many of them are about solar collectors

3.1 Boundary Conditions

The solar beam irradiation (Gb) is selected to be 1000 W/m^2 for all the examined cases. Giving a typical value for solar intensity is a reasonable technique for focusing on the thermal comparison of the examined finned absorbers. The incident angle (θ) is selected to be zero in all the cases in order to give the emphasis in the thermal analysis of the cases. Many experimental studies were conducted by the researchers to find out the pattern of heat flux around the tube of LS-2 collector and almost all results show a pattern like in the figure 4. Figure 4 shows the profile of Local Concentration Ratio around tube with respect to its circumferential angle. One of the most common technique used to find the profile is Monte Carlo Ray Tracing method which accurately predicts the LCR around tube. For this study, we have taken the Ray profile from the analysis of Cheng et al. [3]

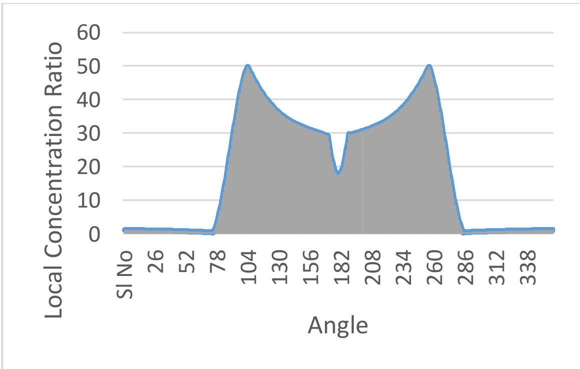

Fig -4: Flux profile for LS-2 collector

3.2 Mesh and Grid Independency

To ensure the accuracy of the numerical results, a careful check of the grid dependence of the numerical solutions has been carried out by considering four grid systems with large number of grid points, i.e., 432557 cells, 863532 cells, 1639754 cells and 2230456 cells. The averaged Pressure drop on these four grid systems are listed in Table 2. It is found that the relative deviations of Pressure drop between grid 3 and grid 4 is only 0.015%. Thus, to save computer resources and keeping a balance between computational economy and prediction accuracy, the grid system of 1639754 cells is chosen.

Table 1: Grid Independency

Sl. No.	Elements	Pressure Drop	Baseline Difference	Aspect Ratio
1	432557	91.42	6.12	4.5
2	863532	93.05	4.49	4.505
3	1639754	97.545	.015	4.5
4	2230456	97.56	0	4.56

3.3 Validation

The model validation is done based on comparing current numerical results for the Nusselt number and friction factor of base fluid flowing in the receiver tube of collector with Pethukhov experimental correlations. The Pethukhov correlation for the fully developed turbulent flow in circular tubes is given by

Nusselt Number $Nu = \dfrac{\frac{f}{8} Re\, Pr}{1.07+12.7\left(\frac{f^{\frac{1}{2}}}{8}\right)(Pr^{\frac{2}{3}}-1)}$

Friction Factor $f = (0.79\ln Re - 1.64)^{-2}$

Validation is conducted by applying a uniform heat over the boundary of the tube. The analysis provides inlet outlet temperature, solid boundary temperature and pressure drop. From the data Nusselt number and Friction factor can be found out. It is then compared with theoretical data as given by Pethukhov equation. From Fig. 5 and 6, it can be seen that the present CFD results agree well with the correlations used for comparison, with a maximum difference of 11% and 13.5% for Nusselt number and friction factor, respectively.

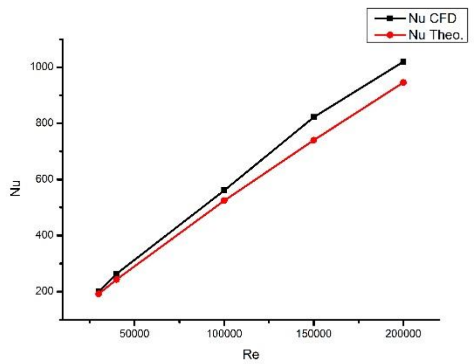

Fig -5: Validation of Nusselt number

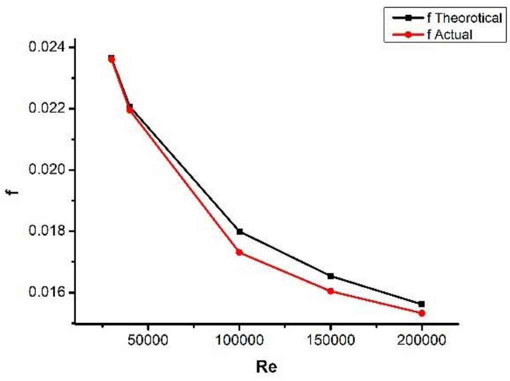

Fig -6: Validation of friction factor

4. RESULTS

4.1 Evaluation of Nusselt Number and Friction Factor

The variation of Nu (Nusselt number) with increase in Re (Reynolds number) is shown in figure 7. From the figure it can be inferred that Nusselt number is greatly enhanced by the addition of the intermittent fins. The pipe with pitch of 1d has the maximum Nusselt number and the pipe with 0.5d fin pitch spacing has the lowest Nusselt number in the lower turbulent region (Re up to 100000). This is mainly because, the turbulence generation points (obstructions) in the 1d pitched pipe is higher than that of the rest of the pipes. Since the fins are closer and there is an actual space between two fins, it results in higher turbulence and thus higher Nusselt number.

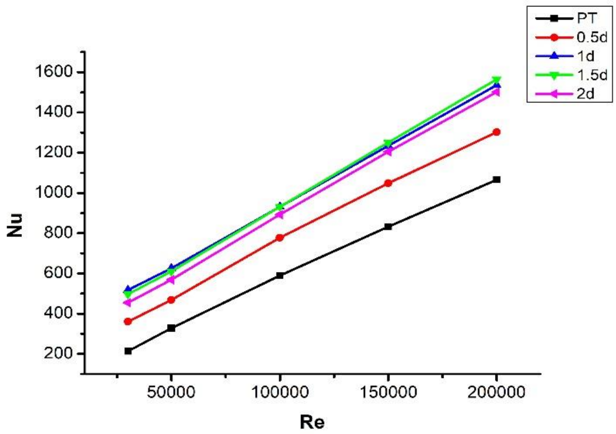

Fig -7: Nu vs Re graph

Also heat transfer area of pipe with 1d pitched fins is higher than the pipes with 1.5d and 2d pitch. The pipe with 0.5d pitched fins has the highest heat transfer area. Since it has no turbulence generation points, the Nusselt number increase is low compared to the rest of the models. However, in the higher turbulence region (Re>100000), the pipe with the fin pitch of 1.5d has a greater Nusselt number compared to 1d. From the figure, it can be noticed that the slope of 1.5d pitched pipe is greater than that of 1d pitched pipe and 2d pitched pipe has greater slope than that of 1.5d pitched pipe. It is clear from the graph that as the pitch of the intermittent fins increases, the Nusselt number increases in the higher turbulence region and decreases in the lower turbulent region.

From the figure 8, it is evident that as the pitch decreases, the friction factor Increases. Maximum friction factor is for 1d pipe as it has higher number of obstructions to the flow than that of the lower pitched fins. The friction factor is least for the 0.5d pitched pipe as it didn't have much obstruction to the flow. The pattern is consistent throughout the entire Reynolds number range.

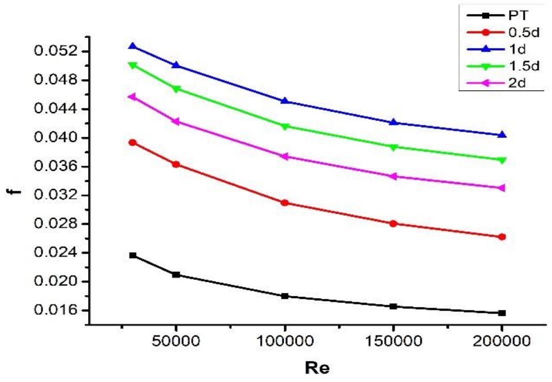

Fig -8: Friction factor vs Re

4.2 Evaluation of Nusselt Number ratio and Friction Factor ratio

The graph of Nusselt number ratio vs Re is shown in Fig 9. From the figure, it can be inferred that the 1d pipe has the maximum increase in Nusselt number in the lower turbulence region. For a Reynolds number of 30000, the Nusselt number of the 1d pitched pipe was 2.42 times as that of plain tube. So Nusselt number increased by around 142%. It was 132% in the 1.5d pitched pipe and 112% for 2d pitched pipe. 0.5d has the least increase in Nusselt number, 68%. The heat transfer augmentation in 0.5d pitched pipe was only done by the use of extended surfaces whereas in the other cases, it was done by a combination of extended surfaces and turbulence generation points.

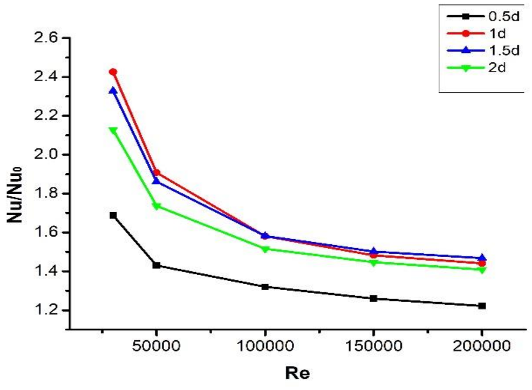

Fig -9: Nusselt number ratio vs Re

As the Reynolds number increases, the Nusselt number ratio decreases and the decrease is more pronounced for the pipes with lower pitched fins. So the downward slope of 1d fins is higher than that of 1.5d fins so that the two graphs meet at Re = 100000. Above Re =100000, the Nusslet number ratio of 1.5d is higher that of 1d even though former does have less turbulent generation points and heat transfer area. The same can be said in the case of 1.5d and 2d also. So it can be inferred from the graph that as the pitch increases the Nusslet number ratio decreases in the lower turbulence region and increases in the higher turbulence region.

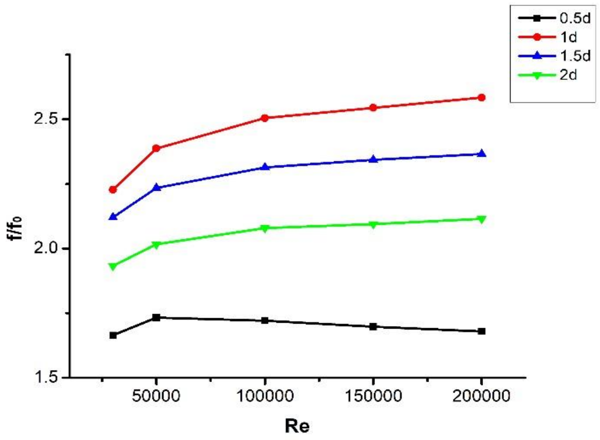

Fig -10: Friction factor ratio vs Re

The friction factor ratio vs Re graph is shown in fig. The ratio remained almost constant in 0.5d. It might be due to the fact that the 0.5d doesn't have any turbulence generation points. In 1d, 1.5d and 2d the friction factor ratio is found to be increasing as the Reynolds number increases. The maximum increase in friction factor was in the 1d with 122% increase at a Reynolds number of the 30000. The increase was 112%, 93% and 66% in cases 1.5d, 2d and 0.5d respectively. The slope of the graph was higher in the case of 1d than that of 1.5d and 2d.

4.3 Evaluation of PEC

The PEC vs Re graph is shown in fig 11. As evident from previous results, the performance 1d is higher than that of the rest of the pipes in lower turbulent region. Up to a Reynolds number of 100000, fins spaced at a pitch length of 1d shows maximum efficiency. Above 100000, the efficiency of 1.5d becomes greater than that of 1d. From the graph it can be inferred that the downward slope of the graph is inversely proportional to pitch distance for all cases except 0.5d. Since 0.5d (straight fin) doesn't have any turbulence generators, the slope remains very low throughout the entire Reynolds number range.

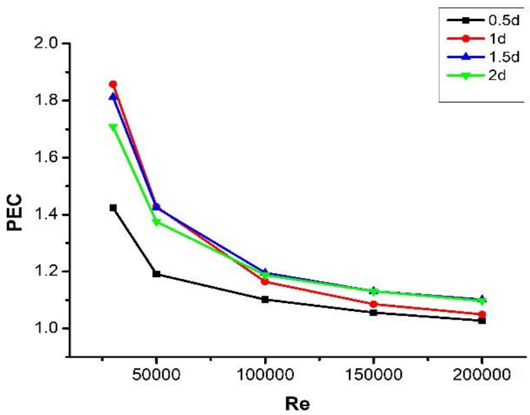

Fig -11: PEC vs Re

5. CONCLUSION

In this study, a numerical simulation of turbulent flow and heat transfer in PTC receiver tubes with internal intermittent fins have been investigated. The results show that inclusion of intermittent fins in pipes greatly improves heat transfer. Nusselt number and its percentage increase was higher for pipes with intermittent fins with a pitch of 1d in the lower turbulence region (less than 100000 Re). This is due to the higher transfer area and higher turbulence generation points in such tubes. The friction factor also was higher in the case of pipes with intermittent fins with pitch of 1d. But as the turbulence is increased, the heat transfer efficiency of the pipes with intermittent fins with a pitch of 1.5 d is found to be greater than that of fins with a pitch of 1d. In general, as the fin spacing is lowered, the efficiency is found to be greater in lower turbulent region and lesser in higher turbulent region.

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