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HEAT TRANSFER ENHANCEMENT ANALYSIS OF SOLAR PARABOLIC TROUGH COLLECTOR TUBE WITH PIN FINS

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Abstract - *The non-uniform heat flux around the periphery of the receiver results in a large temperature gradient. Heat transfer enhancement methods were used to reduce the temperature gradient across the tube. This study deals with the numerical simulation of turbulent flow with heat transfer in Parabolic Trough Collector receiver tubes with staggered pin fin arrangement. It was found that addition of the fins resulted in better performance than the plain tube. The effect of varying the height of the fin was also investigated. Increase in the height of the fins resulted in increase in heat transfer as well as the pressure drop. Combining the effect of increase in both the parameters the finned receiver tube with fin height equal to 12 mm showed the best performance.*

Key Words: Parabolic Trough Collector, Fins, Temperature gradient, Pressure drop.

1. INTRODUCTION

With the rise in global population and industrialization in developing countries, the demand for energy has reached unimaginable heights. Currently the use of fossil fuels is in such a rate that the nature would take one million years to produce the fossil fuel consumed by us in a year. It is expected that within some decades, all the known oil reserves will be exhausted if we continue the current rate of energy consumption with the assumption of no population growth. Apart from the problem of increase in energy demand, the energy harnessing and utilization of the energy resources lead to the environmental damage and cause many environmental problems such as air pollution, water pollution, global warming, climate change, etc. At the current scenario, we cannot abandon any of the existing energy sources. We must bring necessary changes to reduce their environmental impact and new sources of energy must be added, mainly the renewable ones.

Solar energy is a clean source of renewable energy that is harnessable from everywhere in the world. It is a limitless source of power, as it comes from the sun. Any location where the sunlight hit the surface of the Earth can be treated as a potential location to generate the solar power. So solar energy is an ideal source of energy for those areas which are far located from the electric grid. The solar energy can be converted either to useful heat or to electricity, which make use of various applications from water heaters, solar cells, solar absorption refrigeration system, solar stills, solar dryers, solar cooling systems and to electricity production with the concentrating solar power plants.

The concentrating solar power and photovoltaic are the two main solar energy utilization technologies. Concentrating Solar Power (CSP) technologies use various mirror configurations to concentrate the sunlight onto a receiver and convert it into heat. The heat can then be used for the production of hot water or generation of steam to drive a turbine to produce electric power. Thermal energy storage systems are integrated with Concentrating Solar Power plants to generate electricity during cloudy periods or for hours before the sunrise or after the sunset. When the sunlight is diffused, the thermal energy of the sunlight is low. But when it is concentrated it can reach high temperatures. There are mainly four types of CSP technologies are used to concentrate the sunlight. They are Parabolic Trough Collector (PTC), Linear Fresnel Reflector, Parabolic Dish Collector and Solar Power Tower. Among these CSP technologies, the Parabolic Trough Collector is the most mature and widely applied concentration technology, with the best commercial future. Parabolic Trough Collector uses large mirrors shaped like parabola that are connected in long lines to track the sun's movement throughout the day. When the sunlight strikes on this parabolic mirror, it will get reflected from the mirror and the curved shape sends most of the sunlight onto a receiver pipe i.e. filled with a heat transfer fluid. The bottom area of the receiver tube will be subjected to the concentrated rays reflected from the parabolic mirror, while the top portion of the receiver is subjected to the direct sunlight. . Cheng et al. [1] calculated the solar energy flux distribution around the periphery of PTC by Monte Carlo Ray Tracing method (MCRT). It was found that the distribution of the solar energy around the tube is highly non-uniform. The bottom portion of the tube will be having a higher temperature compared to the bottom portion of the tube. This leads to a temperature gradient along the periphery of the receiver tube due to the non-uniform heat flux. This temperature gradient can lead to thermal strain and thereby thermal deformation of the receiver tube and the glass envelop surrounding the tube and can lead to the rupture of the glass cover [2].

Heat transfer enhancement methods are used to reduce the temperature gradient. The enhancement methods involve the use of nanofluids, inserts and addition of fins, etc. Mwesigye et al. [3] numerically studied the heat transfer enhancement in a parabolic trough receiver tube using twisted tape inserts. The maximum rise in heat transfer was 236% and the reduction in circumferential temperature was 76% on comparing with the plain receiver tube. The rise in friction factor was 21.8 times compared to that of a plain receiver tube. Kaloudis et al. [4] used Syltherm 800/ Al_2O_3 as the heat transfer fluid in the parabolic receiver tube for a numerical study and it was found that the two phase approach give more accurate results than the single phase approach. The presence of nanoparticles cause an enhancement in the heat transfer and for an Al_2O_3 concentration of 4%, the rise in efficiency of the receiver was found to 10%. Ghasemi et al. [5] numerically investigated the heat transfer characteristics of a solar parabolic trough receiver 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 the receiver tube 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.

Benabderrahmane et al. [6] numerically investigated the heat transfer in the receiver tube of PTC with longitudinal fins and nanofluid. The Nusselt number is responsive to the type of nanoparticle used. Nusselt number varies between 1.3 to 1.8 times and friction factor varies between 1.6 to 1.85 times that of the plain tube. The metallic nanoparticle show higher heat transfer enhancement than other nanoparticles. Higher heat transfer enhancement can be obtained by combining the fins and the nanofluids. Bellos et al. [7] conducted a numerical study to find the optimum number of internal fins in the receiver tube of PTC. The fins were rectangular in shape with 10 mm height and 2 mm width. Various number of fins were placed at various locations of the receiver tube and its performance was evaluated considering the rise in Nusselt number and friction factor. Higher number of fins lead to high performance and the fins should be located at the lower part of the receiver tube, where the higher temperature is observed. The use of fins at the upper part of the receiver tube does not offer significant increase of the performance. Gong et al. [8] numerically investigated the effect of pin fins on the receiver tube of PTC. The average Nusselt number was increased up to 9% and overall heat transfer performance factor was increased up to 12% that of the plain tube.

The addition of fins are regarded as one of the best method, which increases the heat transfer by increasing the surface area in contact with the fluid flow and it increases the effective conductivity of fluid and hence an increase in the Nusselt number. It leads to the increase in the thermal efficiency of the collector. There were only few researches which adopted the addition of pin fins to the receiver tube of PTC. Providing intermittent pin fin arrays is an effective way to increase the turbulence and can achieve the increase in heat transfer. But the addition of this pin fins lead to relatively higher pressure losses and higher friction factor and this leads to the requirement of higher pumping power. Therefore an increase in heat transfer with the addition of fins can be obtained only through the penalty of higher pressure losses. So it is necessary to evaluate the performance of the finned tube by taking it account of rise in both heat transfer and pressure drop.

The main aim of this study is to calculate the thermo-hydraulic performance of the receiver tube of PTC with pin fins arranged in a staggered manner.

2. MODEL DESCRIPTION

2.1. Physical Model

The solar collector selected for this study is the LS-2 module. The geometrical parameters of the collector used in this study are listed in Table 1. For the analysis, only the receiver tube of inner diameter (D) 66 mm and 1m length is selected and it is made of stainless steel. Geometric modelling of the tube was done in ANSYS DesignModeler. The receiver tube is modified with internal pin fin arrays. The pin fins are to be placed on the higher temperature part. So the pin fins are limited only to the inner lower circumference of the tube. The pin fins are cylindrical in shape with a diameter (d) of 4 mm and height (h) of 2 mm. The sketches of the absorber tube with pin fins used for the simulation are shown in Fig. 1. The geometrical dimensions illustrated are inner diameter of receiver tube (66 mm), outer diameter of receiver tube (70 mm), axial distance between two pin fins (p), pin fin height (h), pin fin diameter (d).

Table 1 Geometrical parameters of LS-2 PTC module

Model Dimensions	Values
Length (L)	5 m
Width (W)	5 m
Focal length (F)	1.84 m
Concentration ratio	22.74
Area of aperture	39 m^2
Receiver tube outer radius	70 mm
Receiver tube inner radius	66 mm
Cover outer radius	115 mm
Cover inner radius	109 mm

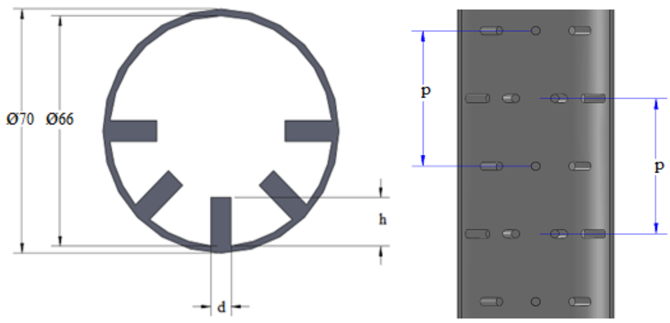

Fig -1: Sketches of receiver tube with pin fins

The parameter which is varying in this analysis is the height of the fin. The fin diameter is taken to be 4 mm and the axial distance between fins is taken as p = 0.5D. Total 8 geometries are considered for the analysis (Plain Tube (PT), pin fin height h - 0.5d, 1d, 1.5d, 2d, 2.5d, 3d, 3.5d).

2.2. Governing Equations

The governing equations are solved using Ansys Fluent 16.2. The following assumptions are made to solve the governing equations of the flow conditions and the heat transfer: steady, incompressible and turbulent flow with constant thermo-physical properties for water.

Mass conservation equation

$(\nabla.V)=0$

Momentum conservation equation

$\rho(\nabla.V)V= -\nabla P+ \mu\nabla^2 V$

Energy conservation equation

$\rho C(V.\nabla)T=k\nabla^2 T$

The useful heat is calculated from the outlet temperature (To) from the simulation, inlet temperature (Ti), mass flow rate ($\dot{m}$) and specific heat capacity (C_p).

$Q_u= \dot{m}C_p (T_o-T_i)$

The solar irradiation available on the aperture of PTC (Q_a) is the product of solar beam irradiation (G_b) and collector area (A_c).

$Q_a= G_b.A_c$

The heat transfer coefficient (h) between the receiver tube and the fluid is calculated using useful heat, collector dimensions, as length (L) and receiver tube inner diameter (D), mean fluid temperature (T_f) and mean receiver temperature (T_r).

$$h = \frac{Q_u}{(\pi DL)(T_r - T_f)}$$

The mean fluid temperature (T_f) can be estimated as:

$$T_f = \frac{T_o + T_i}{2}$$

The Nusselt number can be obtained using the heat transfer coefficient:

$$Nu = \frac{hD}{k}$$

The friction factor is calculated according to the following equation using pressure drop (ΔP) from simulation, fluid density (ρ), tube length (L), mean fluid velocity (V) and inner diameter of the receiver tube (D).

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

The theoretical value of the friction factor can be calculated according to the following equation. This equation is valid only for smooth receiver tube and it can be used only for the validation of developed model.

$f_{th} = (0.79\ln(Re)-1.64)^{-2}$

$$Nu = \frac{\frac{f}{8} Re\, Pr}{1.07 + 12.7\left(\frac{f}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)}$$

To compare the performances of the finned tube working under same operating conditions, the thermal enhancement index is used. In this case the thermal enhancement index is given as Performance Evaluation Criteria (PEC). The index evaluates the different cases under the identical pumping work. The Performance Evaluation Criteria is defined as:

$$PEC = \frac{(Nu/Nu_0)}{(f/f_0)^{(1/3)}}$$

The term Nu/Nu_0 is the ratio of Nusselt number of the given case to the Nusselt number of smooth tube case. And f/f_0 is the ratio of friction factor of the given case to the friction factor of the smooth tube case.

2.3. Boundary Conditions

The heat flux distribution around the tube is non-uniform with a lower value of heat flux at the top periphery and a higher value of heat flux at the bottom periphery of the receiver tube. The pattern of heat flux around the receiver tube is as shown in Fig. 2. Monte Carlo Ray Tracing method is used to find the LCR around the tube. The ray profile is taken from Cheng et al. [1]. The solar beam irradiation (G_b) is taken to be 1000 W/m². The temperature of the heat transfer fluid (water) at inlet is taken to be 300 K. Inlet condition is taken as velocity inlet and outlet condition is made to be pressure outlet.

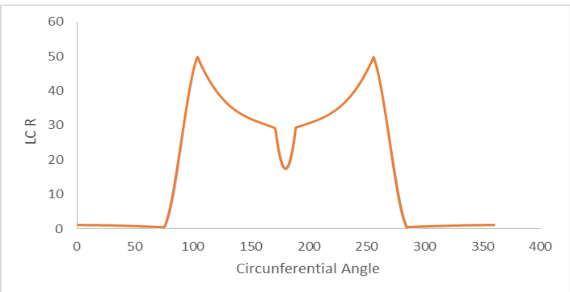

Fig -2: Profile of heat flux around receiver tube

3. NUMERICAL IMPLEMENTATION

The designed models were simulated in CFD program, Fluent. The Fluent is able to perform simultaneous flow and thermal analysis. The governing equations are discretized by Finite Volume Method (FVM). Among the different turbulence models available for analysis, the Shear Stress Transport k-ω model (SST k-ω) is found to give the better results for this problem. The coupling between pressure and velocity is based on SIMPLEC algorithm. The convective terms in momentum and energy equations are discretized with second upwind scheme.

3.1. Mesh and grid independency

Polyhedral mesh was generated all over the model. Grid independence study was conducted by considering six different mesh sizes: 528066, 1057741, 1626452, 2141631, 2367517 and 2640726 for discretizing the entire domain of the receiver tube. For each case, the values of pressure drop was obtained. When the cell count was increased from 2.3 million to 2.6 million, the variation in pressure drop is only 0.02 Pa and it is clear that if the cell count is increased further, it would not have any effect on the output and it will increase the computational time. So to save the computer resources and to keep a balance between prediction accuracy and computational economy, the grid system of 2640726 cells is chosen for the study.

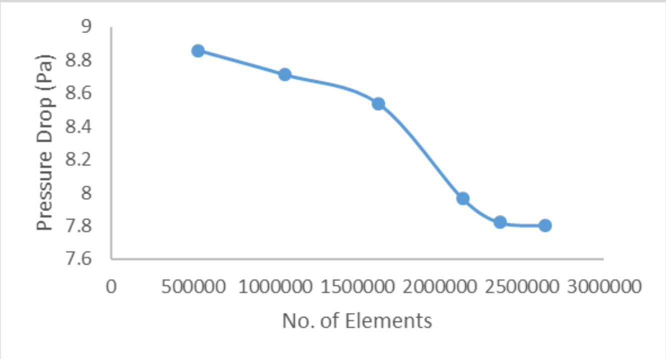

Fig -3: Comparison of pressure drop for different grids

3.2. Validation

For the validation, a uniform heat flux is applied around the boundary of the tube. And inlet temperature, outlet temperature, boundary temperature and pressure drop were obtained from the analysis. Using these values the Nusselt number and friction factor were calculated. And these values were then compared with the theoretical value obtained from the Petukhov equation. Fig. 4 and Fig. 5 indicates the comparison of the CFD results with the results given by the correlation. And it is clear that the CFD results agree well with the results obtained using the correlations.

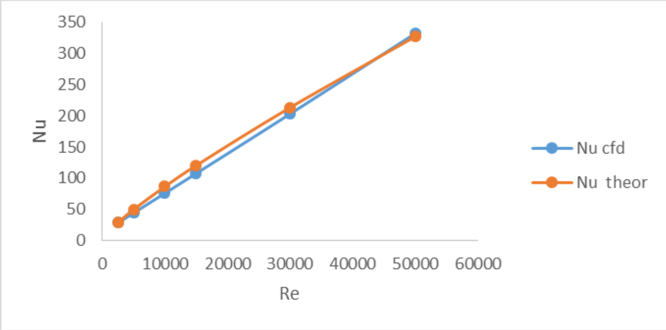

Fig -4: Validation of the model for Nusselt number

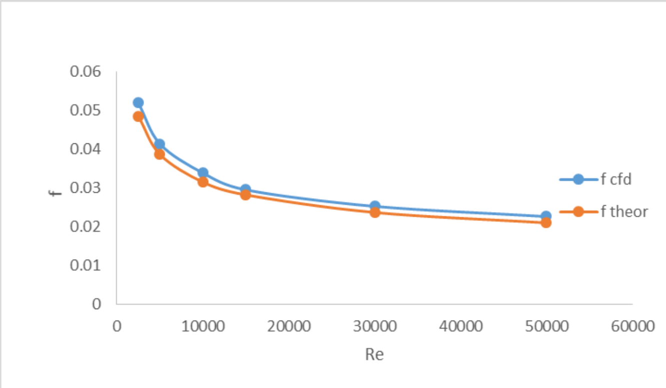

Fig -5: Validation of the model for friction factor

4. RESULTS

4.1. Nusselt number and friction factor

When the flow rate is low i.e. at low Reynolds number, the heat transfer coefficient also will be low and consequently receiver temperature will be high. Therefore the thermal losses will be high. So in order to achieve better thermal performance the receiver tubes are operated at higher Reynolds number (Re). And to evaluate the thermal performance, Nusselt number (Nu) is used.

Fig. 6 shows the change in Nusselt number with respect to the increase in Reynolds number. It is clear from the graph that Nusselt number is enhanced with the addition of fins. Each fin acts as a vortex generator and it increases the turbulence within the receiver tube because of the generated vortices. The Nusselt number increases with the fin height and it is maximum for an h of 3.5d. As the fin height is increased, the area of contact between the fluid in the tube and inner wall of the tube increases. And this increased area results in an increase in heat transfer from the hot receiver tube to the bulk of the fluid. Even though h=3.5d is having more heat transfer area, the effective heat transfer area for h = 3d and h = 3.5d will be almost same. Therefore the difference between Nusselt number for h = 3d and h = 3.5d are minimal.

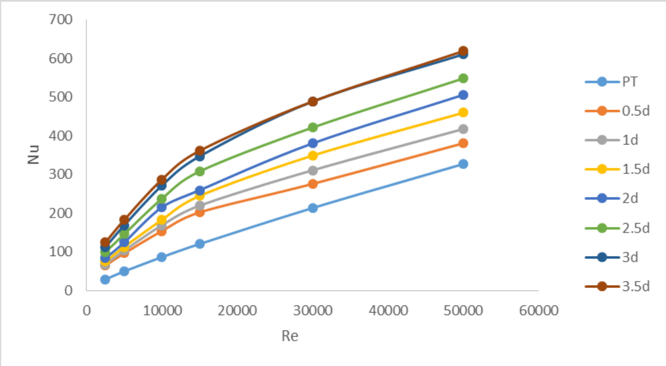

Fig -6: Nusselt number for different fin heights

High mass flow rate or high Reynolds number indicates high thermal performance as well as high pressure drop. And to evaluate the pressure drop, friction factor is used. The variation of friction factor with Reynolds number is shown in Fig. 7. The friction factor decreases with the increase of Reynolds number. The introduction of fin acts as an obstruction to the flow and it leads to significant drop in pressure. And as the fin height is increased, the pressure loss increases and the increase in pressure loss contribute to an increase in the friction factor. Therefore friction factor will be high for h = 3.5d. The increase in pressure drop leads to an increase in the pumping power and it leads to high electrical consumption during the operation of PTC.

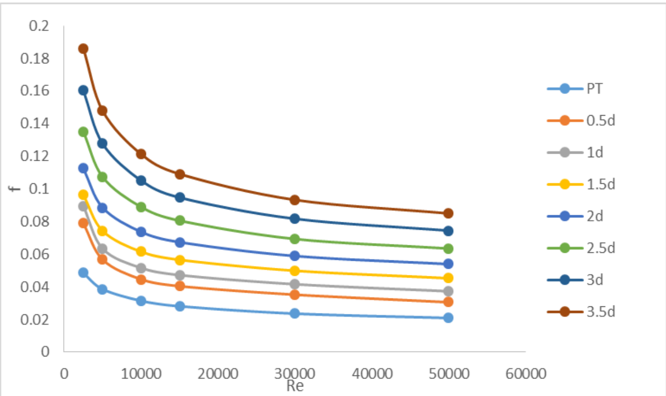

Fig -7: Friction factor for different fin heights

4.2. Nusselt number ratio and friction factor ratio

Fig. 8 represents the variation of Nusselt number ratio with Re. For a Reynolds number of 30000, the pipe with h = 3.5d is having the highest Nusselt number and the Nusselt number is 2.29 times as that of the plain tube. So the increase in Nusselt number is about 129%. At lower Reynolds number the superior nature of the finned pipe with h = 3.5d is clearly visible and as the Reynolds number increases, the difference in Nusselt number for the pipes with h = 3d and h = 3.5d will be minimal.

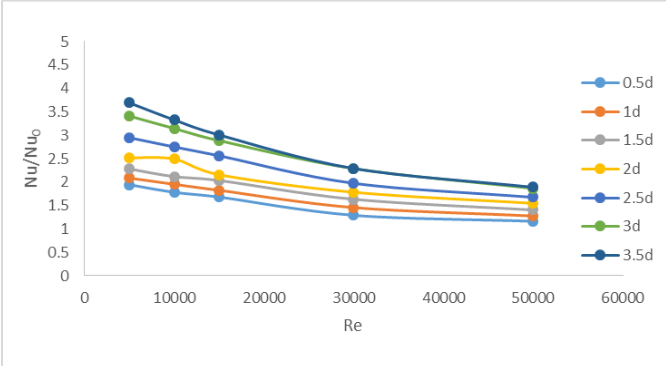

Fig -8: Nusselt number ratio for different fin heights

The graph of friction factor ratio vs Reynolds number is shown in Fig. 9. The friction factor ratio is almost constant for an h = 0.5d. When h = 0.5d, it means that the fin height is low and therefore the turbulence created by the fin will be low and the obstruction to the flow will be low. Due to the less obstruction to the flow, the pressure drop will be low and therefore friction factor will be also low. But as the h increases, the obstruction to the flow and the turbulence increases and the friction factor ratio will be highest for h = 3.5d. For a Reynolds number of 30000, the increase in friction factor for the pipe with h = 3d and h = 3.5d are 245% and 294% respectively.

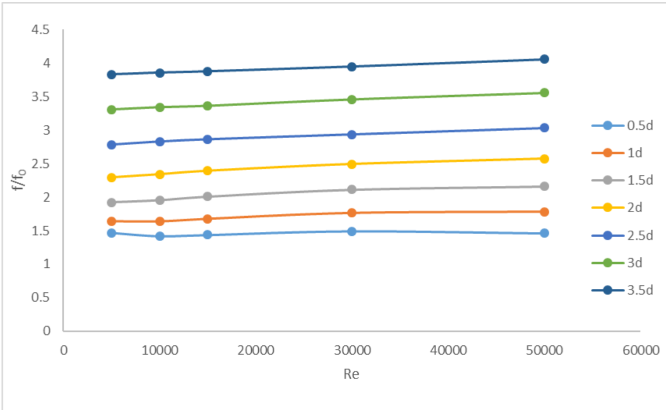

Fig -9: Friction factor ratio for different fin heights

4.3. Performance Evaluation Criteria

Fig. 10 shows the variation in PEC vs Reynolds number for different h values. At lower Reynolds number the performance is high as the percentage of increase in heat transfer of the finned tube is high and percentage of increase in pressure drop is low comparing to the plain tube. But as the Reynolds number increases, the effect of pressure drop will be high comparing to the rise of heat transfer. Therefore PEC value decreases as the Reynolds number increases. At lower Reynolds number, the finned pipe with h = 3.5d is having the highest PEC. This is due to its percentage of increase in heat transfer is high compared to the percentage of rise in pressure drop to that of the plain tube. But above a

Reynolds number of 10000, the finned pipe with h = 3d is showing the highest PEC value. It is because at higher Reynolds number the Nusselt number for the h = 3d and h = 3.5d are almost similar, but the friction is higher for the finned pipe with h = 3.5d. Therefore at higher Reynolds number, PEC value will be higher for the finned pipe with h = 3d.

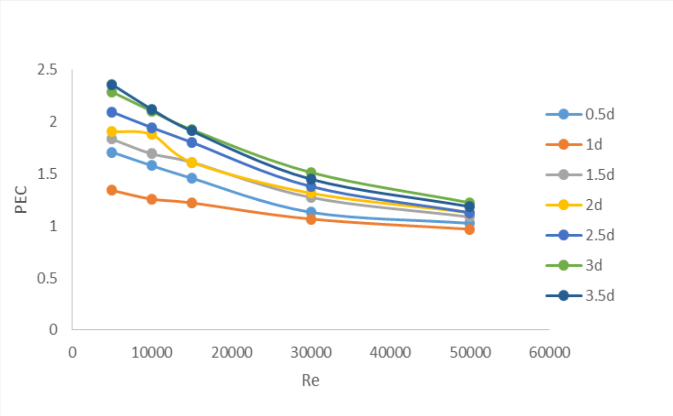

Fig -10: PEC for different fin heights

5. CONCLUSION

A numerical simulation of turbulent flow with heat transfer through PTC receiver tubes with staggered pin fins was conducted. The results showed that the inclusion of the fins on the receiver tube enhances the heat transfer. As the fin height was increased, the area of contact between the fluid in the tube and inner wall of the tube increased. And this increased area resulted in an increase in heat transfer from the hot receiver tube to the bulk of the fluid. But when value of h is increased from 3d to 3.5d, the difference in increase of heat transfer is found to be minimum. The friction factor of the tube with value of h = 3.5d is also higher. So fin with h = 3d give the best performance.

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