

Computational Analysis of an artificial roughened surface of Solar Air heater

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ABSTRACT: In this paper computational thermal analysis (CFD) of solar air heater equipped with an artificial roughened surface has been investigated using FEV approach. The effect of square shape geometry on heat transfer characteristics, i.e. friction factor, coefficient of heat transfer and performance augmentation has been examined for a wide range of roughness parameters such as relative roughness height (e/D) working for Reynolds number from 5700 to 18000 with a solar radiation of 1000W/m^2 . In the developed computational model different turbulent model has been examined and their comparative assessment has been plotted. It has been found that the RNG $k-\epsilon$ model shows good agreement in accordance to predict heat transfer characteristics and friction factor. The overall augmentation ratio has been computed with the aim of illustrating the overall effect of the working parameters and in roughened geometry. .

KEYWORDS: Artificial roughness, Solar air heater, Nusselt number and friction factor.

I. INTRODUCTION

Generally, in solar air heater the thermal efficiency has been significantly low because of their intrinsically low heat transfer potential between the air flowing and the absorber plate within the duct. Sequentially to make solar air heater economically feasible, their thermal performance needs to advance by improving the heat transfer coefficient. In order to conceive better thermal performance the absorber plate is projected in turbulent flow regime so that the effect of artificial roughness on the surface can be seen.

In solar air heater thermal performance is adversely affected by viscous sub-layer since it reduces the coefficient of heat transfer between absorber plate and air. In order to overcome through this crisis turbulence promoters is employed in the form of artificial roughness on the absorber plate. It creates interruption in the flow path which leads to increase friction and turbulence which ultimately improves the thermal performance of solar air heater.

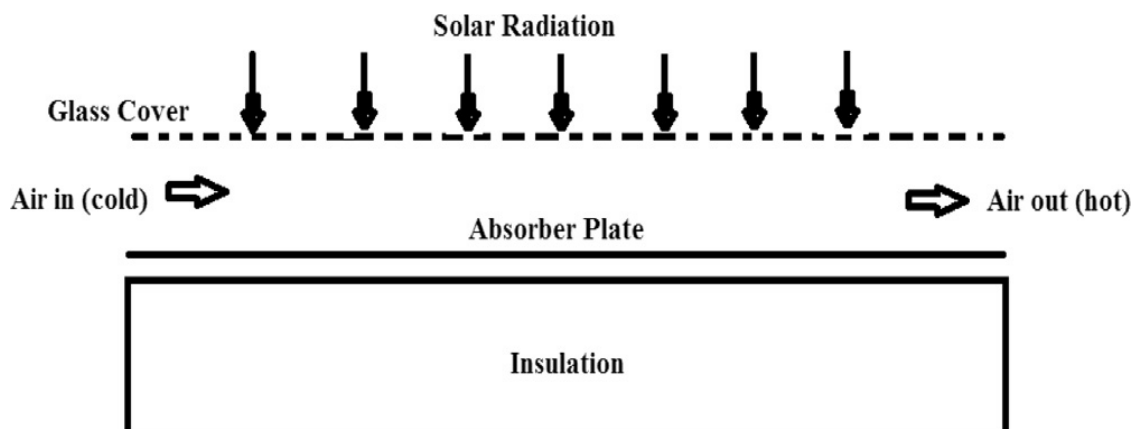


Figure 1 Conventional Solar air heater

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II. LITERATURE REVIEW

In 1995 Ong has expanded a mathematical model and method for predicting thermal performance of single pass solar air collectors using the thermal network analysis procedure and a matrix inversion method a set of simultaneous equations for the mean temperature of the walls and the air streams were calculated and develop a bench mark for designing and analysis the Thermal performance of solar air heaters.

Bhushan et al. 2012 have evaluated thermal and thermo hydraulic performance of the roughened solar air heater having protruded absorber plate using a mathematical model the Optimum value of each roughness geometry parameter has been evaluated on the basis of thermal and effective efficiency.

Similarly, an analytical approach was employed by Ammari 2003 to develop a mathematical model for calculating the thermal performance of a solar air heater with slats. An exact analytical solution for fully developed convective heat transfer in rectangular ducts under a constant heat flux was derived by Shahmardan et al. 2012

Paisarn 2005 numerically studied the performance and entropy generation of the double-pass flat plate SAH with longitudinal fins. Parametric investigation has been carried out in order to analyzed the effect of various inlet parameters on heat transfer characteristics, performance, and entropy generation.

Jaurker et al. 2006 experimentally evaluate the effect of rib-grooved artificial roughness on heat transfer and fluid flow characteristics for a wide range of aspect ratio. The obtained result for rib-grooved has been compared with smooth one and reveals that the heat transfer coefficient for rib-grooved configuration is significantly higher than that for the transverse ribs, while, the friction factor is vaguely higher for rib-grooved arrangement as compared to that of rectangular transverse ribs of similar rib spacing and rib height. And develop a correlation for Nusselt number and friction factor.

Varun et al. 2007 presents a comprehensive review report on various element geometries employed as artificial roughness in solar air heaters with the aim of improve the heat transfer capability of SAH ducts. And also reviewed the various correlations developed for heat transfer and friction factor in roughened ducts of solar air heaters by various researchers.

Hatami and Bahadorinejad 2008 experimentally studied the natural convection heat transfer in a vertical flat-plate SAH. In study 6 different air flow configurations are considered and these cases included states that air could flow within spaces between the glass covers and the absorber plate or air was enclosed in such spaces and the efficiency is determined and found that the maximum efficiency can be attained when air heater had two glass covers and air possibly will flow in all channels.

Arvind et al. 2009 use W-shaped roughness to analyze the friction and heat transfer characteristics experimentally with different aspect ratio. The experiment was encompassed in wide range of Reynolds numbers and developed a correlation for nusselt number and friction factor.

Hikmet et al 2009 use novel technique i.e. artificial neural network (ANN) and wavelet neural network (WNN) for examining the heat and flow characteristics for double-pass SAH. The obtained results are compared with experimental result and found that they are within acceptable limits and the proposed methodology can be utilized for approximating the some factors of SAHs with reasonable precision.

Anil et al.2013 conducts and experiment using multi v-shaped with gap rib as artificial roughness and developed a correlation for friction factor and nusselt number at optimum value of geometrical parameters of roughness. They conclude that the maximum enhancement of the order of 6.74 times of the corresponding value of the smooth duct.

Lal et al 2015 examined the Thermal performance of the packed bed solar heat storage system. In their experimentation rock pebbles are used as insulating material. The efficiency solar collector and heat retrieval is found to be in range of 36-51 and 75-77% and also revealed that by introducing phase change material as packed bed filler the heat retrieval efficiency can be improved.

Singh and Dhiman 2016 investigates experimentally and analytically, the energy and exergy efficiencies of double pass packed bed solar air heater under external recycle . The input parameters are optimized by using thermo-hydraulic efficiency criterion. It has been found that on increasing mass flow rate and recycle ratio the energy and exergy efficiency significantly increases. but in case of exergy efficiency it attains a peak value and consequently starts decreasing as the mass flow rate and the recycle ratio further increases.

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Roosbeh et al.2015 experimentally examined the thermal performances of perforated glazed solar air heaters with different inner collector colors and a black colored unglazed transpired SAH. Perforated plexiglas glazing with different configuration with different colors has be constructed and examined. It has found that the color of perforated plexiglas glazing significantly affects the thermal performance of solar air heater.

Dongxu et al. 2015 computational analyzed the heat transfer and fluid flow characteristics of solar air heater equipped with V-shaped ribs on the absorber plate. By means of ANSYS FLUENT k- ε turbulence model the effect of various parameters such as pitch ratio, angle of attack, number of ribs are scrutinized at different Reynolds number. At last they conclude that multi V-shaped ribs produce stream-wise helical vortex flows, which improves the fluid mixing between the colder mainstream fluid and the warmer fluid near the absorber wall which leads to improvement in thermal performance parameter.

Dong Ho et al.2015 experimentally and theoretically investigate the effect of wire mesh packed on performance of SAH. It has been observed that the obtained result has 1.07-9.32%. deviations with theoretically and experimentally for different configuration such as single pass ,double pass (flat plate) and recycling wire mesh packed double-pass. On comparing these configurations the collector efficiency for the recycling wire mesh packed double-pass is significantly higher than other arrangements under various mass flow rates and recycles ratios.

III. MATHEMATICAL MODELLING

The governing equations for solar air heater duct with artificial roughness this problem are those of Navier-Stokes along with the energy equation. The Navier-Stokes equations are applied to incompressible flows and Newtonian fluids, including the continuity equation and the equations of conservation of momentum on the x and y

According to equations

$$\frac{\partial u_2}{\partial t} + u_1 \frac{\partial u_1}{\partial x_1} + u_2 \frac{\partial u_1}{\partial x_2} = -\frac{1}{\rho} \frac{\partial \rho}{\partial x_2} + \nu \left(\frac{\partial^2 u_1}{\partial x_1^2} + \frac{\partial^2 u_1}{\partial x_2^2} \right) + g\beta(T - T_\infty) \quad (1)$$

$$\frac{\partial u_1^*}{\partial x_1^*} + \frac{\partial u_2^*}{\partial x_2^*} = 0 \quad (2)$$

x1 momentum equation

$$\frac{\partial u_1^*}{\partial t^*} + u_1^* \frac{\partial u_1^*}{\partial x_1^*} + u_2^* \frac{\partial u_1^*}{\partial x_2^*} = -\frac{\partial p^*}{\partial x_1^*} + \text{Pr} \left(\frac{\partial u_1^*}{\partial x_1^*} + \frac{\partial u_1^*}{\partial x_2^*} \right) \quad (3)$$

x2 momentum equation

$$\frac{\partial u_2^*}{\partial t^*} + u_1^* \frac{\partial u_2^*}{\partial x_1^*} + u_2^* \frac{\partial u_2^*}{\partial x_2^*} = -\frac{\partial p^*}{\partial x_2^*} + \text{Pr} \left(\frac{\partial u_2^*}{\partial x_1^*} + \frac{\partial u_2^*}{\partial x_2^*} \right) + Gr \text{Pr}^2 T^* \quad (4)$$

Energy equation

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$$\frac{\partial T^*}{\partial t^*} + u_1^* \frac{\partial T^*}{\partial x_1^*} + u_2^* \frac{\partial T^*}{\partial x_2^*} = \left(\frac{\partial^2 T^*}{\partial x_1^{*2}} + \frac{\partial^2 T^*}{\partial x_2^{*2}} \right) \tag{5}$$

Where Gr is the Grashof number given as

$$Gr = \frac{g \beta \Delta T L^3}{\nu^2} \tag{6}$$

Often, another non-dimensional number called the Rayleigh number is used in the calculations. This is given as

$$Ra = Gr Pr = \frac{g \beta \Delta T L^3}{\nu \alpha} \tag{7}$$

$$Nu = \frac{hl}{k} \tag{8}$$

$$\text{Over all enhancement ratio} = \frac{\frac{Nu_r}{Nu_s}}{\left(\frac{f_r}{f_s} \right)^{1/3}} \tag{9}$$

Table 1 Operating parameters used for CFD analysis.

Operating parameters	Range
Uniform heat flux (q)	1000W/m ²
Reynolds number (Re)	5700–18,000
Prandtl's number (Pr)	0.7441
Relative roughness pitch (e/D)	0.0229-0.0666
Duct aspect ratio (W/H)	5
Hydraulic diameter of duct	45
Roughness height	1.4 and 2.2

IV. METHODOLOGY

The ANSYS 14.5 finite element program was used for simulating the SAH circular wire transverse circular rib absorber plate as shown in figure 1. For this purpose, the key points were first created and then line segments were formed. The lines were combined to create a surface. Finally, this surface is provided thickness model is made. We modeled 4 different geometrical configurations of the absorber plate with different number of fin relative roughness height (e/D). The duct was discretized into 21870 elements with 47550 nodes as shown in figure 2 and 3. SAH duct is divided in four different sections namely inlet, outlet, absorber plate, circular rib respectively. Where boundary conditions are been applied using table 1 during simulation for evaluating the heat transfer and fluid flow characteristics.

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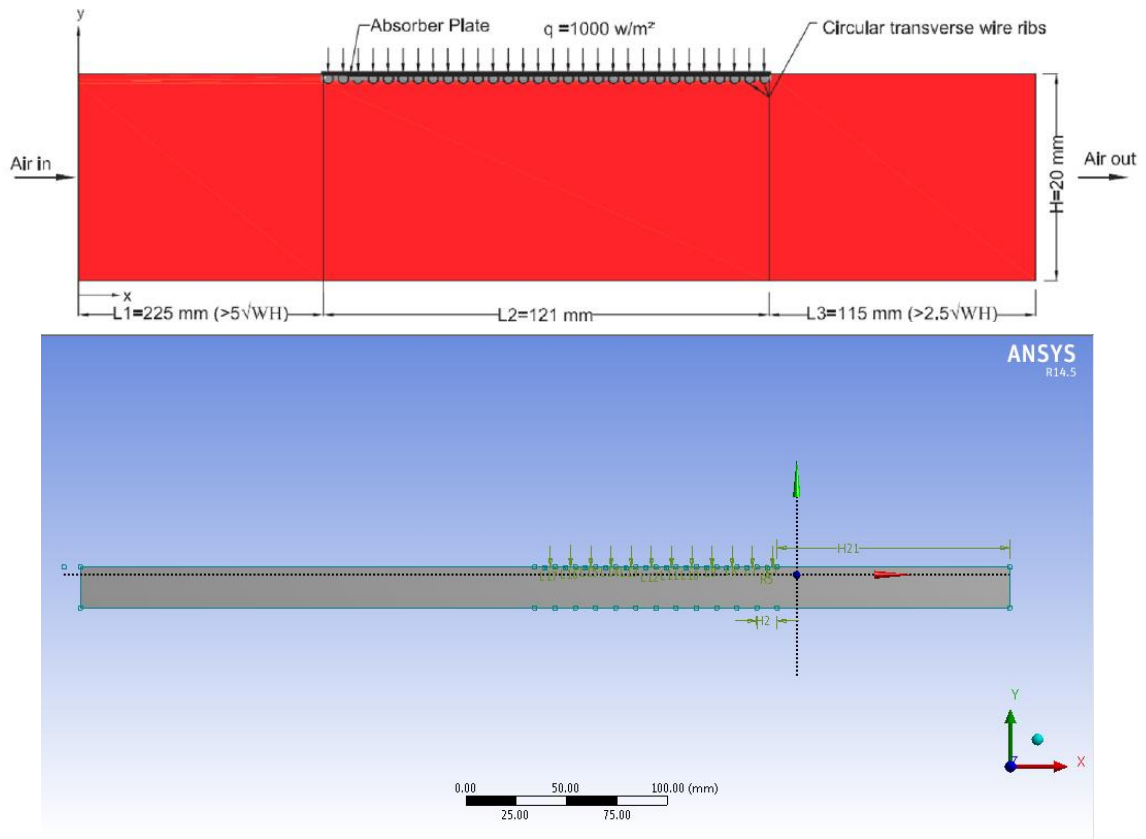


Figure 2 2-D computational domain of a solar air heater

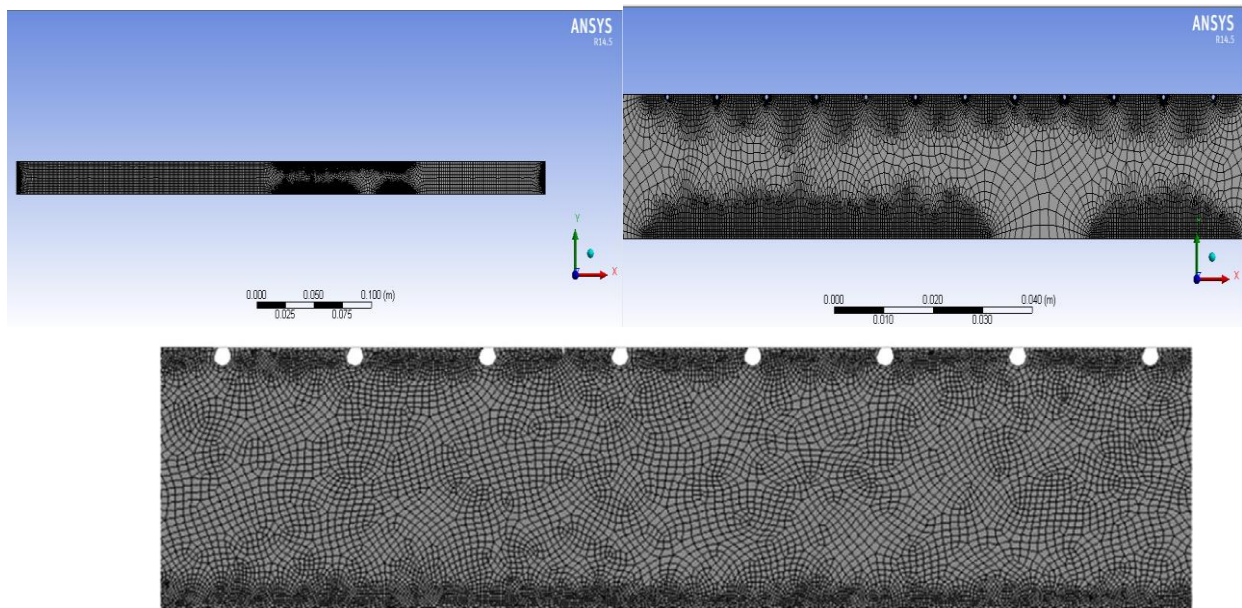


Figure 3 Computational Mesh model

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Some essential Hypothesis in the mathematical model

- The flow regime is steady, fully developed, turbulent and two dimensional.
- The thermal conductivity of the duct wall and roughness material remain constant
- The roughness material and duct wall is homogeneous and isotropic.
- The working fluid, air is presumed to be incompressible for the operating range of SAH since deviation in density is very less.
- The inlet velocity is calculated by using Reynolds number

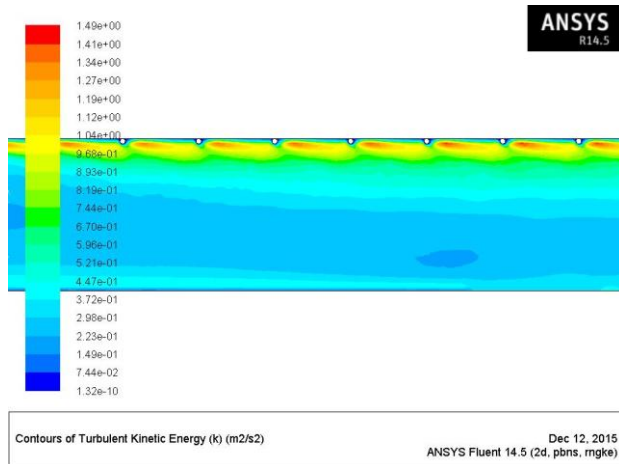


Figure 4 Contours of Turbulent kinetic energy

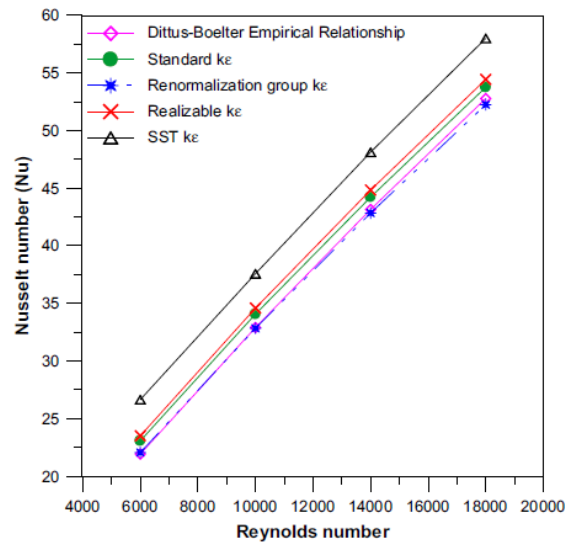


Figure 5 Comparison between Nusselt number predictions of different CFD models with Dittus–Boelter empirical relationship for smooth duct.

Figure 4 shows the turbulent kinetic energy within the duct. It has been observed that the major portion of turbulence occurred at the transverse wire region as shown in above figure. This turbulence leads to increase in friction factor and increases heat transfer rate.

Figure 5 illustrates the comparison of various turbulent models to calculate nusselt number for different Reynolds number. It has been observed that on increasing Reynolds number nusselt numbers significantly increases and on comparing it with Dittus–Boelter empirical relationship for a smooth duct it has found that the RNG $k\epsilon$ model shows good closer agreement with it. Whereas, SST $k\epsilon$ model shows more deviation as compared to other model such as standard $k\epsilon$, realizable $k\epsilon$ therefore RNG $k\epsilon$ model is selected for simulation of roughness absorber plate. It is interesting to know that at low Reynolds number the RNG $k\epsilon$ model shows very accurate result and at high Reynolds number there is deviation of 2%-3% on comparing with Dittus–Boelter empirical relationship.

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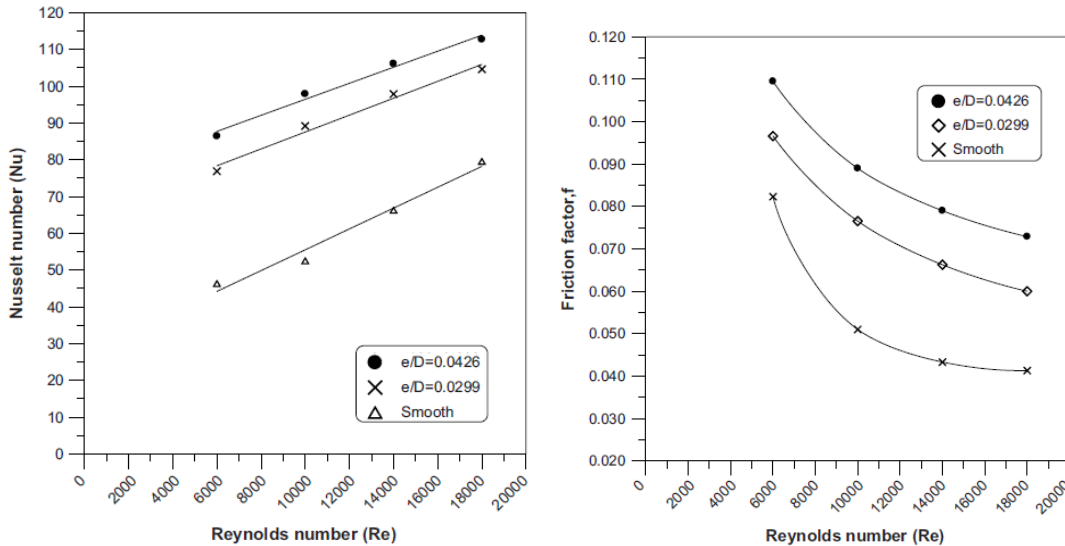


Figure 6 Variation of Nusselt number and Friction factor with Reynolds number for different values of relative roughness height (e/D).

Figure 6 illustrates the variation of Nusselt number with respect to Reynolds number for different ratios of relative roughness height (e/D). It has been observed that on increasing Reynolds number nusselt number linearly increases and similarly on increasing relative roughness height (e/D) nusselt number increases. This means that increased in nusselt number increases the heat transfer rate. It has been observed that on comparing nusselt number for (e/D) =0.0299 is 68.10% and for (e/D)= 0.0426 is 90.80% high as compared to smooth at Reynolds number of 6000.

In case of friction factor, it decreases with increase in Reynolds number. It has been found that friction factor increases as relative roughness height (e/D) increases from 0.0299 to 0.0426 for a given value of Reynolds number. It has been observed that for smooth one there is 49.58% drop in friction factor from 5700 to 18000 Reynolds number. Similarly for e/D=0.0299 and e/D =0.0426 the drop in friction factor is 38.14% and 33.77% . From this it can be conclude that for e/D =0.0426 has minimum friction drop and which results in higher heat transfer rate.

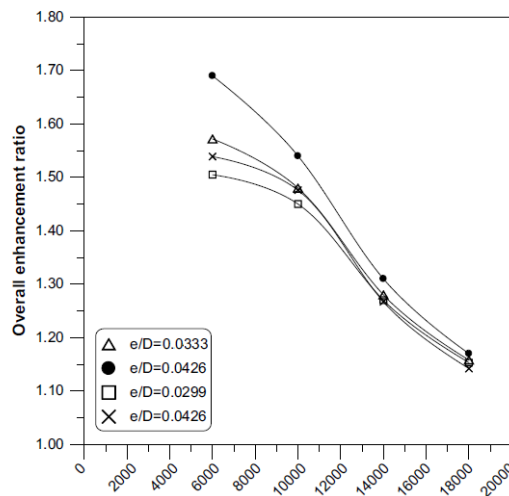


Figure 7 Overall enhancement ratio of various roughness geometries

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Figure 7 demonstrates the overall enhancement ratio of various roughness geometries. It has been observed that on increasing Reynolds number the overall enhancement ratio decreases drastically this is because of drop in friction factor. But there is interesting to know that for all roughened geometrics the overall enhancement ratio is greater than unity and the ratio is highest at Reynolds number 5700 and beyond Reynolds number 10000 the overall enhancement ratio sharply decreases till it reaches nearly to 1.

V. CONCLUSION

- An effort has been made in order to examine the effect of artificial roughness (circular transverse wire ribs) using CFD. The heat transfer and fluid flow characteristics has been analyzed for wide range of Reynolds number. On the basis of this analysis several conclusions has been draw.
- The obtained CFD results are well validated and compared with smooth duct as well as with the empirical relationship for smooth duct and found that the results shows good agreement with CFD results.
- The RNG k- ϵ model shows better converging result as compared with other turbulence model.
- $e/D = 0.0426$ has minimum friction drop and which results in higher heat transfer rate.
- On increasing Reynolds number the heat transfer and fluid flow characteristics increases.
- The combined consequence of swirling motion, detachment and reattachment of fluid which leads to enhancement of heat transfer rate
- The Nusselt number increases as Reynolds number increases.
- Similarly the friction factor decreases with increase in Reynolds number for all combinations of (e/D)
- The overall enhancement ratio is found to be 1.7 maximum for relative roughness height (e/D) of 0.0426

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