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CFD Investigation of Artificially Roughened Solar Air Heater for Performance Enhancement

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Abstract: A numerical investigation of fluid flows through a solar air heater with semi-circular sectioned transverse rib roughness has been executed. The physical problem is represented mathematically by a set of governing equations, and the transport equations are solved using the finite element method. The effect of relative roughness pitch on average Nusselt number, average friction factor, and thermohydraulic performance parameter (THPP) has been investigated. This investigation is taken for relative roughness pitch of 7.5 and relevant Reynolds numbers in the range of $3800 \leq Re \leq 18{,}000$. The maximum thermohydraulic performance parameter is 1.616 and it is occurred for the relative roughness height of 0.06 and relative roughness pitch of 7.5.

Keywords: Solar air heater, Solar energy, CFD, THPP, Nu, Re.

I.INTRODUCTION

Energy is released by sun as the electromagnetic waves. This energy reaching the earth's atmosphere consist of 8% UV radiation, 46% visible light and 46% infrared radiations.

Solar energy can be used in two ways:

1. Solar heating.
2. Solar electricity.

Solar heating is to capture/concentrate sun's energy for heating building and for cooking/heating foodstuffs etc.

Since most renewable energy is ultimately "Solar Energy" that is directly collected from the sunlight. Solar electricity is mainly produced by using photovoltaic solar cells which are made of semi-conducting materials that directly convert sunlight into electricity. Obviously the sun does not provide constant energy to any spot on earth, so its use is limited. Therefore, often solar cells are used either as a secondary source of energy or for other applications of intermittent use such as night lighting or water pumps, etc. A solar power plant offers the best option for electrification in the areas of disadvantageous locations such as hilly regions, forests, deserts, and islands where other resources are neither available nor exploitable in techno economically viable manner.

Interest in solar energy has been growing at an increasingly rapid rate over the past several years. Systems to heat and cool houses, heat water, distilled water, dry food crops, and generate electricity are being shown to be feasible throughout the United States and the world. The primary roadblock to expanded solar energy utilization is not the lack of technology, but the unavailability of easily understandable information.

The word Solar stems from the Roman word for the god of the sun. Therefore, the word solar refers to the sun and "solar power" is power from sun. The sun provides Earth with two major forms of energy, heat and light. Some solar power systems utilize the heat energy for heating, while others transform the light energy into electrical energy. The solar air heater is used for heating purpose.

In countries and where traditional power sources are not available it is more economical to power a house with solar energy. To these people, solar is not an alternative energy, it is their primary energy sources.

In just one second, the sun gives off 13 million times the energy that is generated by all the electricity consumed in one year in the United States. Only one million of sun's energy reaches to earth, but this scant amount would be more than sufficient to meet the energy requirement of the entire planet.

Solar energy has proven more effective and has been more widely utilized for both water and space heating and cooling systems. As a water heater, solar energy is most commonly used to heat swimming pools and space heating.

Solar air heater is one of the basic equipment through which solar energy is converted into thermal energy. A solar air heater is a type of heat exchanger which transfers solar radiation into heat energy. Solar air heaters, because of their simple designing, are cheap and most widely used as a collection devices of solar energy. A solar air heater requires little maintenance. A conventional solar air heater generally consists of an absorber plate, a rear plate, insulation below the rear plate, transparent cover on the exposed side, and the air flows between the absorbing plate and rear plate. The air gets heated up while the absorber plate absorbs the heat. The hot air is drawn through the plates with a blower which is operated electrically.

Conventional solar air heaters have poor thermal efficiency primarily due to high heat losses and low convective heat transfer coefficient between the absorber plate and flowing air stream, leading to higher absorber plate temperature and greater thermal losses. Among the many effective ways to augment the convective heat transfer rate in channel flows, those that increase the heat transfer surface area and those that increase turbulence inside the channel with fins or corrugated surfaces and

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surface roughness have been the most popular. For this reason, the surfaces are sometime roughened in the air flow passage. The use of artificial roughness on a surface is an effective technique to enhance the rate of heat transfer to fluid flowing in a duct. Artificial roughness in the form of repeated ribs has been found to be a convenient method to enhance the rate of heat transfer. Ribs of various shapes and orientations have been employed and the performance of such system has been investigated. Artificial roughness has been used to enhance the heat transfer coefficient by creating turbulence in the flow. However, it would also result in an increase in friction losses and hence greater pumping power requirements for air through the duct. In order to keep the friction losses at a low level, the turbulence must be created only in the region very close to the duct surface, i.e. in the laminar sub-layer.

The purpose of this work is to investigate the effect of roughness height, roughness pitch, relative roughness pitch and relative roughness height on flow friction.

II.RELATED WORK

The concept of artificial roughness was first applied by Joule (1861) [1] to enhance heat transfer coefficients for in-tube condensation of steam and since then many experimental investigations were carried out on the application of artificial roughness in the areas of cooling of gas turbine, electronic equipment, nuclear reactors, and compact heat exchangers etc. Nunner (1956) [2] was the first who developed a flow model and likened this model to the temperature profile in smooth tube flow at increased Prandtl number. The proposed flow model predicts that roughness reduces the thermal resistance of the turbulence dominated wall region without significantly affecting the viscous region. A friction correlation for flow over sand-grain roughness was developed by Nikuradse (1950) [3]. Based on law of the wall similarity, Nikuradse presented the pressure drop results in terms of roughness function R and roughness Reynolds number e+. Dipprey and Sabersky (1963) [4] developed a heat-momentum transfer analogy relation for flow in a sand-grain roughened tube and achieved excellent correlation of their data. The concept proposed by Dipprey and Sabersky was so common and it can be applied to any roughness for which law of the wall similarity holds. Prasad and Mullick (1983) [5] were the first who introduced the application of artificial roughness in the form of small diameter wire attached on the underside of absorber plate to improve the thermal performance of solar air heater for drying purposes. After Prasad and Mullick's [5] work, a number of experimental investigations on solar air heater involving roughness elements of different shapes, sizes and orientations with respect to flow direction have been carried out in order to obtain an optimum arrangement of the roughness element geometry by Hans VS, Saini RP, Saini JS (2009) [6]; Bhushan B, Singh R (2010) [7]; Kumar A, Saini RP, Saini JS (2012) [8]. Karwa et al. (2001) [9] conducted an experimental investigation of the performance of solar air heaters with chamfered repeated rib-roughness on the air flow side of the absorber plates. The authors found substantial enhancement in thermal efficiency (10-40%) over solar air heaters with smooth absorber plates. Saini and Verma (2008) [10] conducted an experimental investigation on fluid flow and heat transfer characteristics of solar air heater duct having dimple-shaped artificial roughness. The authors found the maximum value of Nusselt number corresponds to the relative roughness height (e/D) of 0.0379 and relative roughness pitch (P/e) of 10. The authors also found minimum value of friction factor correspond to the relative roughness height (e/D) of 0.0289 and relative pitch (P/e) of 10. Singh et al. (2011) [11] experimentally investigated the heat transfer characteristics of rectangular duct having its one broad wall heated and roughened with periodic 'discrete V-down rib'. The authors found the maximum value of Nusselt number and friction factor corresponds to relative roughness pitch (P/e) of 8. Tanda (2011) [12] experimentally investigated the heat transfer coefficients and friction factors for a rectangular channel having one wall roughened having angled continuous ribs, transverse continuous and broken ribs, and discrete V-shaped ribs. Author reported that roughening the heat transfer surface by transverse broken ribs appeared to be the most promising enhancement technique for investigating rib geometries. Lanjewar et al. (2011) [13] carried out an experimental investigation on heat transfer and friction factor characteristics of rectangular duct roughened with W-shaped ribs. Author reported that thermo-hydraulic performance improved with angle of attack of flow and relative roughness height and maxima occurred at angle of attack 60°. Bhagoria et al. (2002) [14] performed experiments to determine the effect of relative roughness pitch, relative roughness height and wedge angle on the heat transfer and friction factor in a solar air heater roughened duct having wedge shaped rib roughness. Authors found maximum enhancement of Nusselt number up to 2.4 times while the friction factor up to 5.3 times for the range of parameters investigated. Kumar et al. (2009) [15] experimentally investigated heat transfer and friction characteristics in solar air heater by using discrete W-shaped roughness on one broad wall of solar air heater. The maximum enhancement of Nusselt number and friction factor as a result of providing artificial roughness was found to be 2.16 and 2.75 times than that of smooth duct for an angle of attack 60° and relative roughness height of 0.0338. Yadav and Bhagoria (2013) [16] conducted CFD investigations of an artificially roughened solar air heater to examine the effect of circular transverse ribs roughness on the absorber plate. 2D CFD simulation of the duct was based on CFD code FLUENT v12.1. The simulation results revealed that RNG k -ε turbulence models predicted flow distribution and heat transfer very well. A CFD simulation was conducted with different turbulence models in order to find out the best appropriate turbulence model for simulation of artificially roughened solar air heater. The results obtained by the RNG $k - \varepsilon$ model were in good agreement with the Dittus–Boelter and Blasius empirical correlations. Yadav and Bhagoria (2013) [17] conducted a numerical analysis of the heat transfer and flow friction characteristics in an artificially

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roughened solar air heater having square sectioned transverse rib roughness considered at the underside of the top heated wall. The thermohydraulic performance parameter under the same pumping power constraint was calculated in order to examine the overall effect of the relative roughness pitch. The maximum value of the thermohydraulic performance parameter was found to be 1.82 corresponding to a relative roughness pitch of 10.71.

III.CFD SIMULATION

The 2D heat transfer and fluid flow phenomenon through an artificially roughened solar air heater is investigated by means of a numerical model. The 2D CFD simulation of artificially roughened solar air heater is carried out using commercial CFD simulation code ANSYS FLUENT (version 14.5) to solve the conservation equations for mass, momentum and energy. The numerical model for fluid flow and heat transfer through an artificially roughened solar air heater is developed under the following assumptions:

(1) Steady 2D fluid flow and heat transfer.
(2) Both thermally and hydraulically fully developed flow (steady-state conditions).
(3) The thermal conductivity of the duct wall, absorber plate and roughness material are independent of temperature.
(4) The duct wall, absorber plate and roughness material are homogeneous and isotropic.

Computational domain

Artificial roughness has been used to enhance heat transfer by increasing the level of turbulence mixing in the flow. In numerically simulating such flows, it is common to simulate a 3D rib-roughened duct with a 2Ddomain in order to reduce the computational power and time.

Hence, in this work, the 2D flow is therefore carried out for saving computer memory and computational time. The computational domain of an artificially roughened solar air heater is represented in 2D form by a semi-circle and displayed in Figure 1.

According to ASHRAE Standard (2003), This investigation is taken for relative roughness height (e/D) of 0.06, relative roughness pitch (P/e) of 7.5 and Reynolds numbers (Re) in the range of $3800 \leq Re \leq 18{,}000$ (relevant in solar air heater). Artificial roughness in the form of semi-circular sectioned transverse ribs is considered at the underside of the top of the duct, i.e. on the absorber plate while other sides are considered as smooth surface. A uniform heat flux of 1000W/m^2 is considered on the top of the absorber plate for numerical analysis.

The geometrical and operating parameters employed in this CFD investigation are listed in Table 1, Table 2, and Table 3.

The domain consisted of three sections, namely entrance section (L_1), test section (L_2), and exit section (L_3). The internal duct cross section is 100 × 20mm^2.

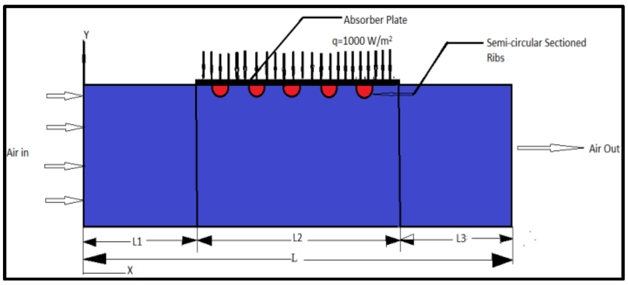

Figure 1: Schematic of 2D computational domain.

Table 1: Geometric parameters of the artificially roughened solar air heater duct.

L_1 (mm)	L_2(mm)	L_3(mm)	W (mm)	H (mm)	D (mm)	e (mm)	P (mm)
225	121	115	100	20	33.33	2	15

Table 2: Range of operating parameters for computational analysis

Operating parameters	Range
Uniform heat flux 'q'	1000 W/m^2
Reynolds Number 'Re'	3800, 5000, 7200, 10000,
Prandlt number 'Pr'	12000, 15000, 18000
Rib hight 'e'	0.7441
Pitch 'P'	2 mm
Relative Roughness Pitch	15mm
'P/e'	7.5
Relative Roughness hight	0.06
'e/D'	5
Duct aspect ratio 'W/H'	

Table 3: Thermo-physical properties of working fluid (air) and absorber plate (aluminum) for computational analysis

Properties	Working fluid (air)	Absorber plate (aluminum)
Density, 'ρ' (kg	1.225	2719
m^{-3})	1006.43	871
Specific heat,	1.7894e-05	-
'C_p' (Jkg^{-1}K^{-1})	0.0242	202.4
Viscosity, 'μ' (Nsm^{-2})		
Thermal conductivity, 'k' (W m^{-1} K^{-1})		

IV.MESH GENERATION

Non-uniform grids are generated for all numerical simulations performed in this work. Non-uniform grids are commonly used in modeling when large gradients are expected.

Automatic non-uniform grids are generated for all numerical simulations performed in this work. Grids are generated using ANSYS ICEM CFD v14.5 software. A non-uniform grid contained 65100 quad nodes and 62058 elements with cell size of 0.0002m is used to resolve the laminar sub-layer, as shown in Figure 2.

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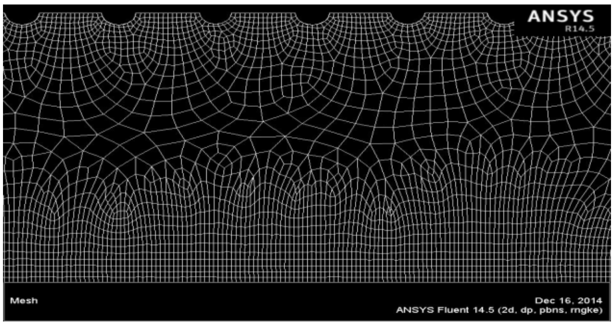

Figure 2: Two-dimensional non uniform meshing.

Governing equations

The flow phenomenon in artificially roughened solar air heater duct is governed by the steady 2-dimensional form of the continuity, the time-independent incompressible Naviere Stokes equations and the energy equation. In the Cartesian tensor system these equations can be written as:

Continuity equation:

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

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial p}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u_i' u_j'}\right) \quad (2)$$

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j}\left((\Gamma + \Gamma_t)\frac{\partial T}{\partial x_j}\right) \quad (3)$$

Where Γ and Γ_t are molecular thermal diffusivity and turbulent thermal diffusivity, respectively, and are given by the following.

$$\Gamma = \mu/Pr \quad \text{and} \quad \Gamma_t = \mu_t/Pr_t \quad (4)$$

Boundary condition:

The solution domain of the considered 2d, rectangular duct flow is geometrically quite simple. It is a rectangle on the x-y plane, enclosed by the inlet, outlet and wall boundaries. No-slip conditions for velocity in solid surfaces are assumed and the turbulence kinetic energy is set to zero on all solid walls. The top wall boundary condition is selected as constant heat flux of 1000 W/m^2 and bottom wall is assumed to be in adiabatic condition.

The temperature of air inside the duct is also taken as 300 K at the beginning. At the exit, a pressure outlet boundary condition is specified with a fixed pressure of 1.013×10^5 Pa. A uniform air velocity is introduced at the inlet and the mean inlet velocity of the flow is calculated using Reynolds number.

V.DATA REDUCTION FOR THERMO-HYDRAULIC PERFORMANCE

The aim of present CFD work is to investigate the effect of relative roughness pitch (P/e) on flow friction, and THPP in artificially roughened solar air heater having semi circular sectioned transverse rib roughness on the underside of the absorber plate.

Average Nusselt number for artificially roughened solar air heater is computed by

$$Nu_r = \frac{hD}{k} \quad (5)$$

Where h is convective heat transfer co-efficient.

The average friction factor for artificially roughened solar air heater is computed by

$$f_r = \frac{(\Delta p/l)D}{2\rho v^2} \quad (6)$$

Where Δp is pressure drop across the duct of an artificially roughened solar air heater.

The Reynolds number is defined as

$$Re = \frac{\rho v D}{\mu} \quad (7)$$

A well-known method of THPP evaluation is proposed by Webb and Eckert (1972) and defined by the following equation

$$THPP = \frac{Nu_r/Nu_s}{(f_r/f_s)^{1/3}} \quad (8)$$

For an enhancement scheme to be viable, the value of this index must be greater than unity.

Nu_s represent Nusselt number for smooth duct of a solar air heater and can be obtained by the Dittus–Boelter equation (McAdams 1942) [18].

$$Nu_s = 0.023Re^{0.8}Pr^{0.4} \quad (9)$$

f_s represents friction factor for smooth duct of a solar air heater and can be obtained by Blasius equation (Fox, Pritchard, and McDonald 2010) [19].

Blasius equation

$$f_s = 0.0791Re^{-0.25} \quad (10)$$

VI.RESULT AND DISCUSSION

CFD computations of heat transfer and fluid flow characteristics in an artificially roughened solar air heater provided with semi-circular sectioned transverse rib roughness on the underside of the absorber plate are performed. The effects of grid density, Reynolds number, and relative roughness pitch (P/e) on the average heat transfer for flow of air in an artificially roughness solar air heater are discussed below. The results have been compared with those obtained in case of smooth duct operating under similar operating conditions to discuss the enhancement of heat transfer on account of artificial roughness.

Grid independence test:

To study variation of heat transfer and flow friction with change in grid size, grid independence study is carried out. Five sets of grids (element size) with different sizes are used for the simulation to insure that the results are grid independent. A grid independence test is implemented over grids with different numbers of nodes, 53,468,

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57,781, 62,305, 65,100, and 68,519 are used in five steps. It is found that the variation in Nusselt number marginally increases when moving from 65,100 cells to 68,519. Hence, there is no such advantage in increasing the number of cells beyond this value. Thus, the grid system of 65,100 cells is adopted for the current computation.

Effect of Reynolds number (Re):

Figure 3 shows the variation of friction factor with Reynolds number for relative roughness pitch (P/e) of 7.5 and relative roughness height (e/D) of 0.06. It is observed that the friction factor decreases with increase in Reynolds number because of the suppression of viscous sub-layer.

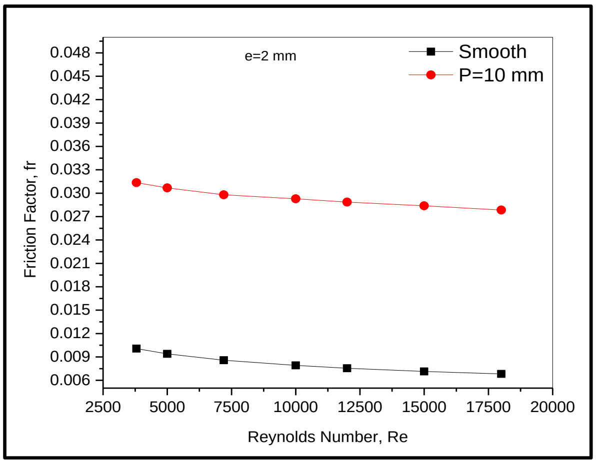

Figure 3: Variation of friction factor with Reynolds number for rib height e=2 mm

The heat transfer phenomenon can be observed and described by the contour of turbulence kinetic energy. The contour plot of turbulence kinetic energy is shown in Figure 7. Since the k-ε models use as the turbulent viscosity, the high values of k in the flow field leads to high heat transfer predictions.

The intensities of turbulence kinetic energy are reduced at the flow field near the rib and wall and a high turbulence kinetic energy region is found between the adjacent ribs close to the main flow which yields the strong influence of turbulence intensity on heat transfer enhancement. Figure 8 shows the contour plot of turbulence intensity at a Reynolds number of 10,000 and e/D=0.06 and P/e=7.5.

The peak value of turbulence intensity is seen on the top of rib front region, and then it decreases with increase in distance from the wall. This can be attributed to the formation of eddies in the vicinity of wall due to the high shear stresses which diffuses in the main flow. Low level of turbulence intensity leads to a low level of heat transfer. Figure 5 shows the contour plot of velocity for Reynolds number of 10,000 and e/D=0.06, P/e=7.5 and it displays stronger vortices because of the presence of semi-circular wire rib roughness, which results in more heat transfer rate.

The peak velocity magnitude is occurred at center line of the duct and lowest magnitude is near the wall and between the adjacent ribs.

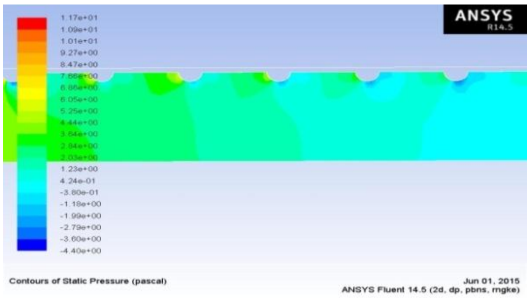

Figure 4: Contour plot for static pressure

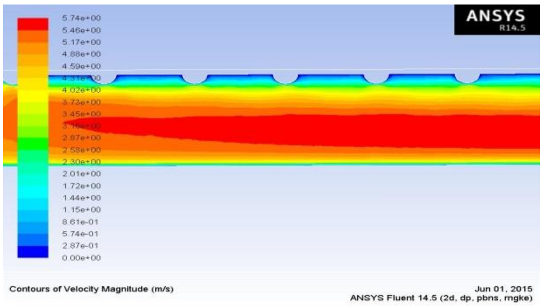

Figure 5: Contour plot for velocity magnitude.

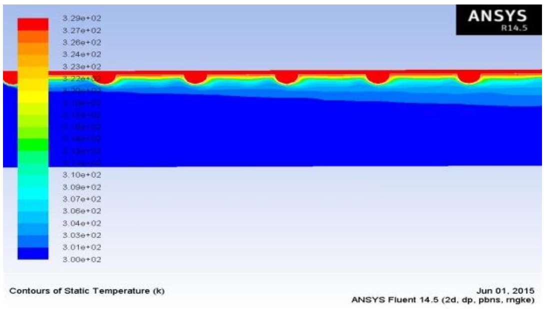

Figure 6: Contour plot for static temperature.

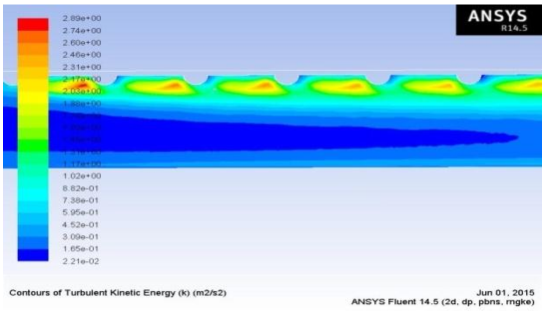

Figure 7: Contour plot for turbulent kinetic energy.

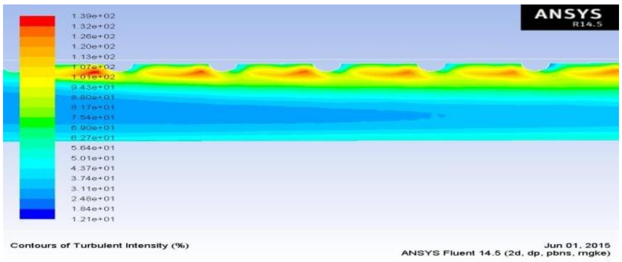

Figure 8: Contour plot for turbulent intensity.

Figure 4 shows the contour plot of static pressure for relative roughness pitch (P/e)=7.5 at a fixed value of Reynolds number of 10,000 and relative roughness height (e/D) of 0.06. It is found that very low pressure variation is occurred in the duct.

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Figure 6 shows the contour plot of static temperature for relative roughness pitch (P/e)=7.5 at a fixed value of Reynolds number of 10,000 and relative roughness height (e/D) of 0.06. Maximum temperature is occurred at the absorber plate and decreases with increase in distance from absorber plate.

Thermal enhancement factor decreases with increase in Reynolds number (Re) and it is found that maximum enhancement factor 1.6165 occurred at Re=3800. Figure 9 shows the variation of Thermal enhancement factor with Reynolds number.

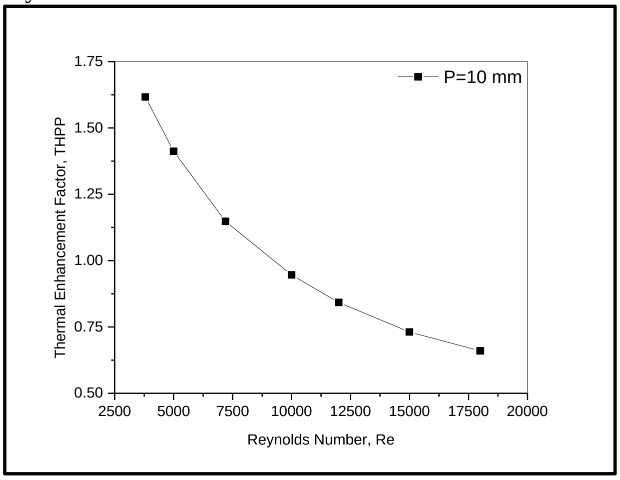

Figure 9: Variation of thermal enhancement factor with Reynolds number

VII.CONCLUSION

A 2-dimensional CFD analysis has been carried out to study heat transfer and fluid flow behavior in a rectangular duct of a solar air heater with one roughened wall having semi-circular transverse wire rib roughness. The following conclusions are drawn from present analysis:

1. The maximum Friction factor ratio is found to be 4.08324 corresponding to relative roughness pitch (P/e) of 7.5 at a Reynolds number of 18,000.
2. It is found that the solar air heater roughened with semi-circular sectioned transverse rib roughness on the absorber plate with relative roughness pitch, P/e = 7.5 provide the better THPP of 1.6165 at a Reynolds number of 3,800.

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