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NUMERICAL AND CFD ANALYSIS OF DUCT WITH INSERTS

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ABSTRACT

The present numerical and CFD investigations have been carriedout to examine friction factor and heat transfer rate of twisted square duct with inserts. simulation is conducted under uniform heat flux conditions with air is a working fluid, twist angle is 270 degree, Reynolds number is varied from 7500 to 38000. The results of friction factor, and enhancement ratio are presented. Friction factor for twisted circular rod is 2.85 times more than twisted duct. Efficiency of twisted duct heat exchanger 42 percentage , twisted circular rod is 36 percentage more than plain square duct under same operating conditions. In this work Numeric simulations were calculated by using the CFD software package ANSYS 18.2 FLUENT has been used.

Keywords: *Twisted square duct, Reynolds number, Nusselt number, Friction factor, Turbulent flow, Twist ratio, Twisted circular rod ,Computational fluid dynamics.*

I. INTRODUCTION

Heat exchangers design procedure is quite complicated, as it needs proper analysis of pressure drop and heat transfer rate estimations distant from issues such as economic aspect of the equipment and long term performance. The primary intention of designing of a heat exchanger is to make compact size and achieve maximum heat transfer by utilizing minimum pumping power. In ongoing years , the high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. Furthermore, as a heat exchanger becomes older, the resistance to heat transfer increases owing to fouling or scaling, there is a need to increase the heat transfer rate. An increase in heat transfer coefficient generally leads to additional advantage of reducing temperature driving force, which increases second law efficiency and decreases entropy generation. Square ducts are widely used in heat transfer devices. For instance in compact heat exchangers, gas turbine cooling systems cooling chambers in combustion chambers and nuclear reactors. Chang et al. [1] numerically studied laminar flow in a twisted elliptic tube for large twist ratios (H = 21, 53, 106) using finite difference method. The effect of twist ratio and aspect ratio of ellipse was investigated with respect to their role in determining the axial and circumferential velocities and streamline patterns. Xiangi et al. [2] investigated heat transfer and pressure drop performances of twisted oval tube have been studied experimentally and numerically and the result shows that with an increasing of pressure drop heat transfer process can be enhanced. Saha and co-workers [3-7] have studied experimentally laminar flow through square and rectangular ducts having twisted tapes with oblique teeth, axial corrugations, transverse ribs and wire coil inserts. Masliyah and Nandakumar [8,9] numerically studied the fully developed steady laminar flow through twisted square ducts with rotation coordinates system. Axial conduction in fluid was neglected to preserve the two dimensional nature of the problem. The temperature along the periphery was assumed to be constant for each wall. However, this constant temperature might be different for each of the four walls. The swirling motion enhanced the heat transfer for a twist ratio of 2.5 and a Reynolds number range of 1–1000. Similar enhancement was however not observed for the other twist ratios. Yang et al. [10] experimentally evaluated performance of five twisted elliptical tubes. Aspect ratio (major diameter/minor diameter) of elliptical tubes used was in the range of 1.49 to 2.15 and twist ratio range covered was 17.4– 32.8. Water was used as the working fluid for Re range of 600–55,000 covering laminar, transition and turbulent regime. They concluded that for twisted duct flow remains laminar for Re less than or equal to 2300. In a twisted tube, the heat transfer enhancement is higher for laminar regime compared to transition and turbulent flow regimes. RambirBhadouriya et al. [11] reported that friction factor and heat transfer characteristics of air flow inside twisted square duct under uniform wall temperature conditions with

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twist ratio of 11.5 and 16.5, Reynolds number varied from 600-70000. The results show significant improvement in pressure drop and heat transfer in laminar and turbulent flow regimes till Reynolds number of 9500. Twist ratio of 11.5 shows quite higher pressure drop and heat transfer compared to plain square duct. Nihal UĞURLUBİLEK et.al [12] according to this study, heat transfer and turbulent flow characteristics through twisted square duct has been numerically investigated. The working fluid is considered as water and the Reynolds number range between10000-1200000. The governing equations implied using the commercial code FLUENT. It has been observed that Twisted square duct provides significant increase in terms of Nusselt number to 138 % over the plain square duct and utmost gain of 1.3 on thermal performance factor is obtained for the case of Reynolds number 10000. This represents the secondary flow occurred through the twisted square duct can increase the heat transfer rate. The edge size of square cross-section, the twist angle and the length of the channel are taken 0.01m, 360º and 0.2 m, respectively.

P.Samruaisin et.al [13] examined pressure drop, heat transfer behaviors of tube integral with commonly spaced quadruple twisted tape elements under turbulent flow regime. Experiments were performed at twist ratio equal to 2.5 under conditions of constant heat flux. Over the range reported, commonly spaced quadruple twisted tapes in cross-arrangement with s/y=0.5 gives up to 6.6% heat transfer rate as well as thermal enhancement factor value is equal to 1.27.

II. FLOW DOMAIN IDENTIFICATION AND NUMERICAL METHODOLOGY

The present work carried out by square and Twisted duct heat exchangers. Initial velocity of flow of air 2.2 m/sec, Reynolds number is 8666. Therefore the type of flow is Turbulent flow

Fluent commercially available Computational Fluid Dynamics software is used for the numerical analysis. The software has the ability to give solution of heat transfer and fluid flow by means of periodic flow concept. As for periodic flows, problems are analysed by restricting the numerical model to a single module of a given periodic length. This permits the use of a scaled model and reduces the computational time.
Based on the experimental study, the flow is to be considered as laminar in the Reynolds number from 100–3000. Turbulent flow numerical studies are undertaken for Reynolds greater than 10,000. Three dimensional models used in the analysis with hexahedron mesh elements. Peripherally and axially constant heat flux boundary conditions are used in the simulations. Body forces due to gravity are neglected. Solutions are obtained for steady state, compressible conditions.

Twisted square duct Specifications

Length of the tested duct	- 1500 mm,
Width	- 65 mm
Height	- 65 mm,
Thickness	- 2 mm.
Material	- Aluminum
Twist angle	- 270 degree

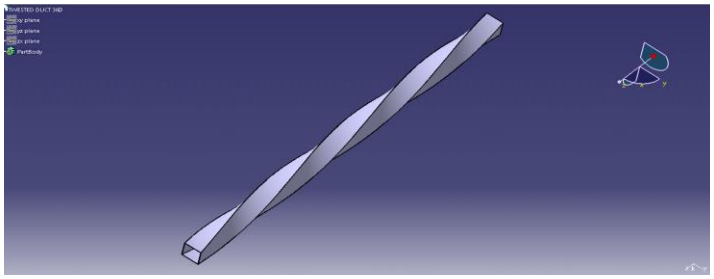

Fig 2.1 Twisted square duct modeling

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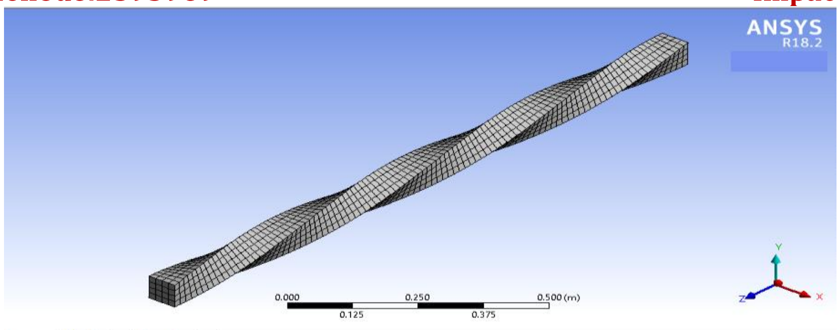

Fig .2.2 Twisted square duct Meshing

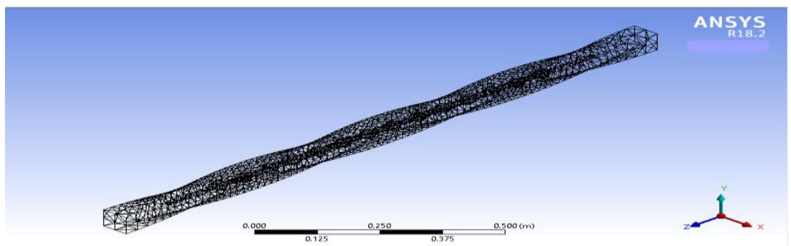

Fig: 2.3 Twisted circular rod meshing

Governing equations and Boundary conditions

The investigation is carried out at one atmospheric pressure, room temperature is 300^0 K .Velocity is taken as inlet condition at 27^0 C where outlet is pressure. Constant heat flux is applied for the all four walls of the duct, which is equal to 480 W/m^2.In this work the inlet velocity is varied as 2.2 m/sec, 3.3 m/sec, 4.5 m/sec, 6.4 m/sec, 8.5 m/sec and 9.5 m/sec respectively. Plain and twisted square duct with inserts of periodic length is meshed with hexahedron elements. Grid independence check was carried out. The parameter values employed in the Table 2.3

Turbulent flow described by using **RANS** equation **i.e.** (Reynolds-averaged Navier –Stokes equations).Where an instantaneous quantity is decomposed into its time-averaged and fluctuating quantities. Here we are using **Standard** k-€ equation to solve our turbulence problem.

Transport equations for standard k-epsilon model

For turbulent kinetic energy k

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k + P_b - \rho\epsilon - Y_M + S_k$$

For dissipation ϵ

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k + P_b - \rho\epsilon - Y_M + S_k$$

Boundary conditions	Inlet	Velocity
	Outlet	Pressure
	Right wall	Heat flux
	Left wall	Heat flux
	Side walls	Heat flux
	Upper wall	Heat flux

In order to obtain the temperature variation at different Reynolds number for each model contour plot is plotted. The different models and its temperature variation are determined for the calculation of Nusselt number. To examine the results, we study temperature based results in graphical mode and velocity results.

III. NOMENCLATURE

A=Convection heat transfer area of duct, m2
AR=Aspect ratio of duct, (W/H)
Cp=Specific heat capacity of air, J/kgK
Dh =Hydraulic diameter of duct, (H), m
f=Friction factor for plain duct
f=Friction factor for twisted square duct
H= Duct height, m
h =Heat transfer coefficient, W/m2K
I =Current, A
K=Thermal conductivity of air, W/mK
L= Length of test duct, m
m=Mass flow rate of air, kg/s
Nuo =Nusselt number of plain duct,
Nu =Nusselt number of Twisted square duct
Pr =Prandtl number
Re =Reynolds number, (UD/ν) (dimensionless)
Q =Heat transfer, W
q =heat flux, W/m^2
T =Temperature, K
Ti =Air inlet temperature, K
To =Air outlet temperature, K
Tb =Bulk temperature, (Ti+To)/2 K
Ts=Surface temperature, K
U =Mean velocity, m/s
V= Voltage, V
W =Width of the duct, m
Y=Twist ratio, dimensionless (s/D), m
ρ_a=Density of air, kg/m3
η=Thermal performance factor (enhancement ratio)

IV. RESULTS AND DISCUSSION

4.1 Validation of smooth square duct:
The experimental results of Nusselt number and friction factor of the present plain Square duct are compared with those from correlations of Dittus–Boelter, Blasius and Petukhov found for turbulent flow in ducts.
Correlation of Dittus-Boelter,
$Nu = 0:023Re^{0:8}Pr0:4$ for heating (1)
Correlation of Petukhov,
$f = 0.79\ (\ln Re-1:64)^{-2}$(2)

The comparison of Nusselt number and friction factor obtained from the present plain square duct with those from correlations of Equations. (1), (2) are represented.

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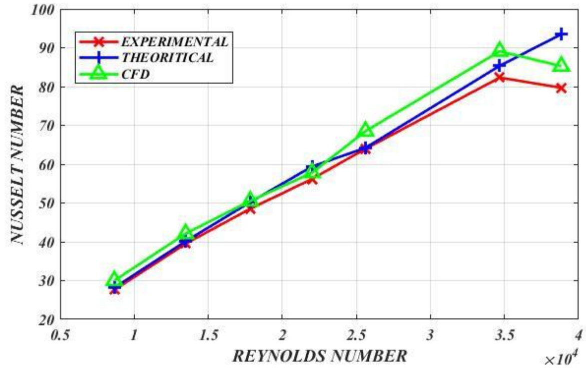

Figure 4.1 : Data verification of Nusselt number versus Reynolds number for plain square duct.

The graph represents variation of Nusselt number with Reynolds number for plain square duct using air is test fluid The experimental values were compare with standard equation. It was observed that Percentage deviation between experimental and theoretical values is ±5.1. Therefore the experimental setup is deemed to be validated

4.2 Effect of inserts in the duct

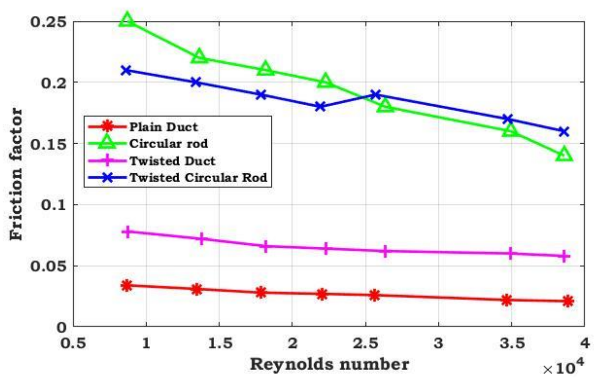

Fig. 4.2 Friction factor versus Reynolds number for plain and twisted square duct with inserts.

It is visible in the above figure 4.2 that, the use of twisted circular rod insert leads to a substantial increase in friction factor above the plain duct and also friction factor shows the decreasing tendency with the increment of Reynolds number. In the plain square duct frictional losses are less. Twisted square duct with circular rod insert results more frictional losses due to swirl flow action and turbulence.

4.2 CFD ANALYSIS

4.2.1 Twisted square duct

Fig. 4.2 Temperature contour of twisted duct at a velocity of 2.2 m/sec.

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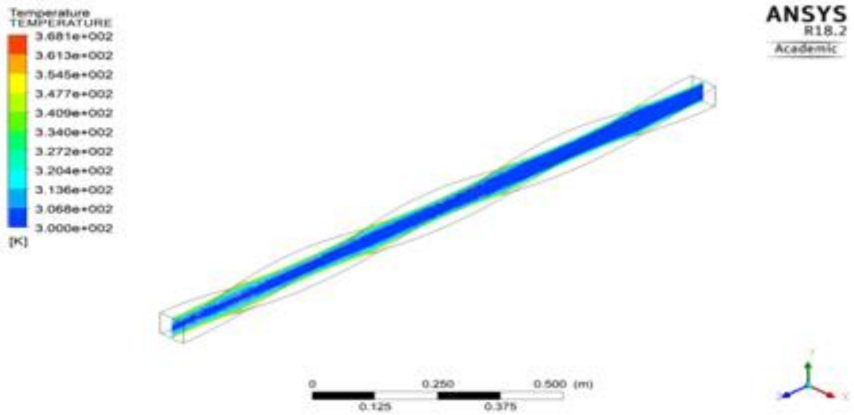

Figure 4.3Variation of temperature in the twisted duct

Due to swirling motion is created by secondary flow helped to increase temperature differences between entry and exit, which in turn to enhance heat transfer rate simultaneously increase heat transfer coefficient also.

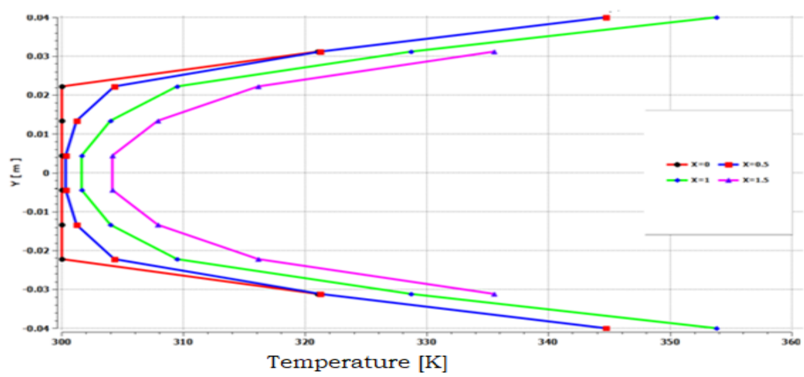

Fig. 4.4 Temperature profile of twisted duct

The Figure 4.4 represents temperature profile of twisted square duct at velocity of 2.2 m/sec. It is observed that at X=1m it is clearly noticed that difference between outlet and inlet temperature is increased. This is due to air flow velocity direction is twisted in spiral manner which helped to get higher temperature difference between exit and entry of the duct.

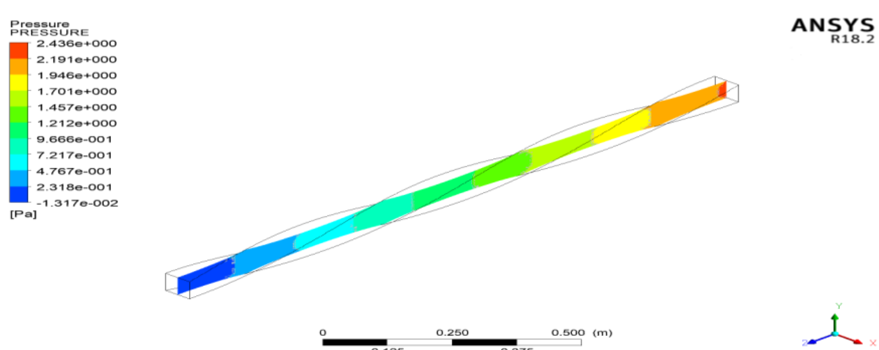

Fig. 4.5 Pressure variation of twisted duct

It is visible that the friction factor tends to increase with raising blockage ratio and Reynolds number.

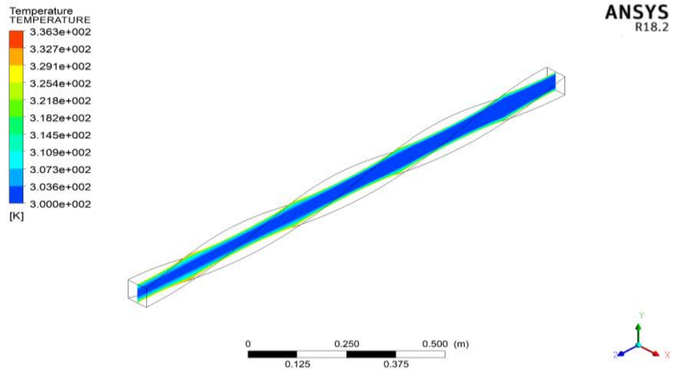

Fig. 4.6 Effect of twisted square duct at a velocity of 4.5 m/sec

It is visible from the figure 4.6 while velocity is increased, it causes the flow to be spiral along the duct length and creates turbulence in the entire flow field that leads to higher heat transfer rate.

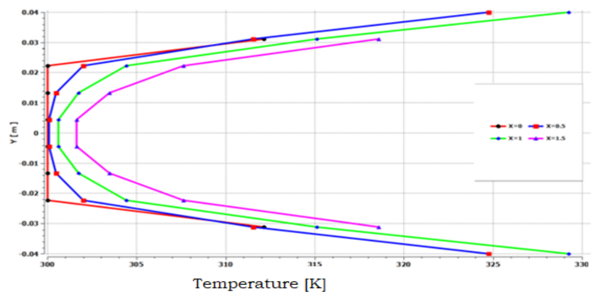

Fig. 4.7 Temperature profiles of twisted square duct

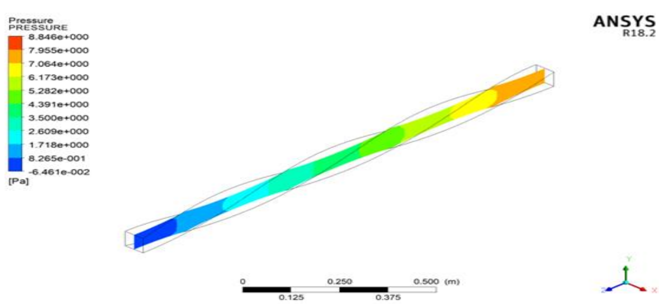

Fig.4.8 Pressure contour of twisted square duct

This is due to twisted square duct creates irregular flow, which leads to increase pressure drop as shown in fig 4.7.

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4.3 Twisted circular rod

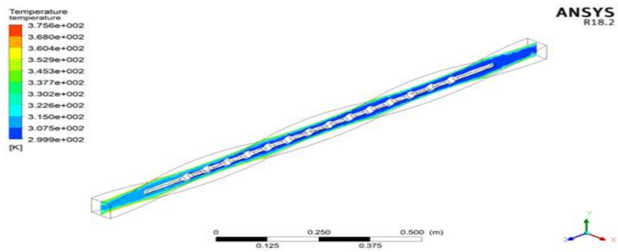

Fig .4.9 Temperature contour of twisted circular rod at a velocity of 2.2 m/sec.

It has been observed that the effect of twisted circular rod creates turbulence, which leads to increase heat transfer rate simultaneously increases heat transfer coefficient.

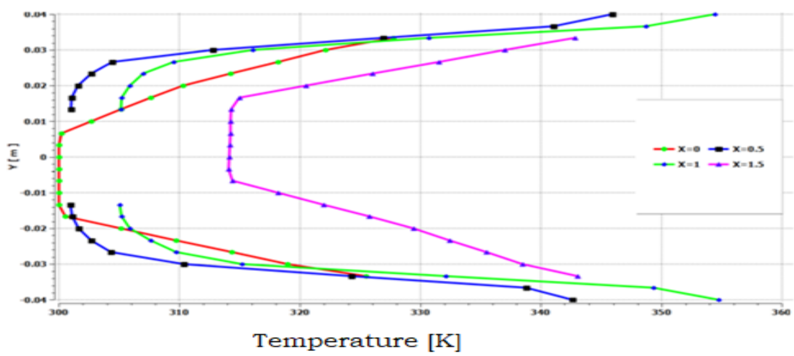

Fig. 4.10 Temperature profile of twisted circular rod

It is clearly observed that at X=1m the inlet and outlet temperature difference is maximum. This is due to proper mixing of fluid in the duct.

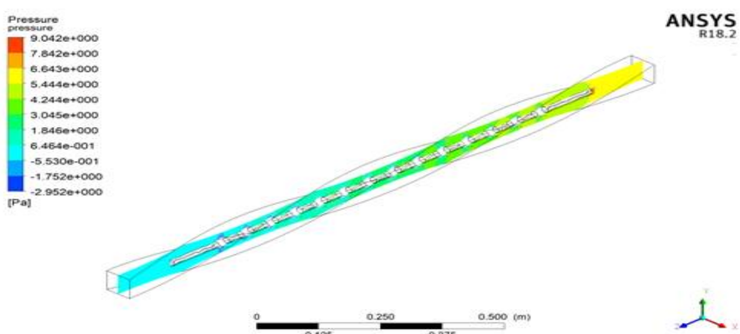

Fig. 4.11 Pressure contour of twisted circular rod

The Figure 4.10 represents pressure drop for twisted square duct with circular rod. Pressure drop is more due to large amount of obstruction of flow by inserting circular rods in the twisted square duct.

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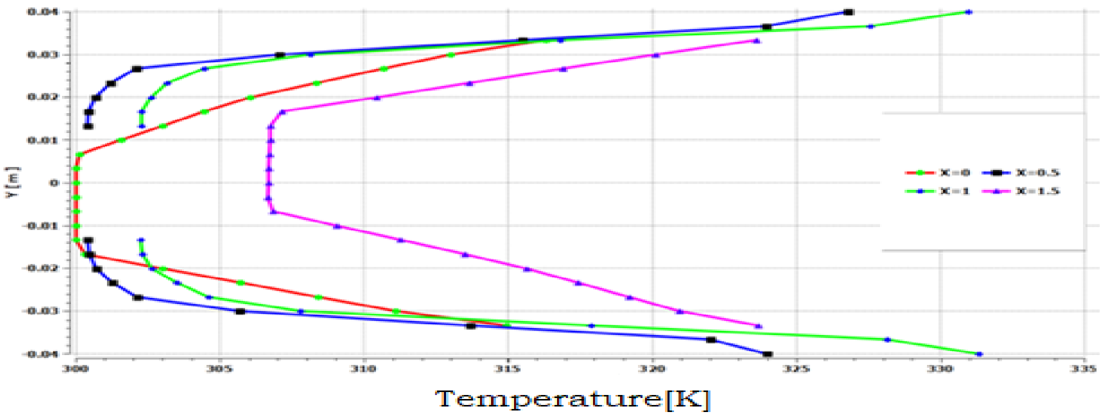

Fig . 4.12 Temperature profile of twisted circular rod at a velocity of 4.5 m/sec.

The Figure 4.12 represents temperature profile at various distances throughout the duct. At X=0,5,1m distance from axis line 12.5 mm distance above and below there is no heating takes place in the region. At X=1.5 meter distance temperature difference is slightly decreased.

V. CONCLUSION

Following main conclusions are drawn from these investigations

- The maximum heat transfer coefficient is obtained at velocity of 8.6 m/sec. in the twisted square duct This is due to continuousswirli flow in the duct
- Nusselt number for the twisted circular rod is about 1.36 times above twisted square duct while friction factor is 2.85 times higher (f/fo= 2.29).
- Efficiency of twisted square duct is more than twisted circular rod
- Final outcome of investigation shows that twisted duct performs better heat transfer rate than plain duct.

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