Analyzing augmentation of heat transfer in horizontal tube with insert by using FEA and Taguchi method

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***Abstract*- In this thesis, investigations are carried out to determine the heat transfer rates in a horizontal tube by means of varying width of twisted tape inserts with air as the working fluid. Analysis is carried out for plain tube with/without twisted tape insert at constant wall heat flux and different mass flow rates calculated for Reynolds number 6000, 10000, 12000, at different widths. 3D models of the plain tube and tube with inserts of widths 10mm, 14mm and 18mm are done in Creo 2.0. CFD analysis is performed for all tubes with inserts to determine heat transfer coefficient, heat transfer rate, Nusselt number and friction coefficient and compared with that of plain tube. Analysis is performed in Ansys. Optimization of Reynolds Number and width of twisted tube insert is done using Taguchi method in Minitab software.**

***IndexTerms*—heat transfer,tube,twisted tape,Reynolds number,Nusselt number,CFD,Ansys,Taguchi method.**

I. INTRODUCTION

The various heat transfer enhancement techniques can be classified broadly as passive and active techniques. Passive techniques do not require direct input of external power, unlike active techniques. They generally use surface or geometrical modifications to the flow channel, or incorporate an insert, material, or additional device. Except for extended surfaces, which increase the effective heat transfer surface area, these passive schemes promote higher heat transfer coefficients by disturbing or altering the existing flow behavior. This, however, is accompanied by an increase in the pressure drop. In the case of active techniques, the addition of external power essentially facilitates the desired flow modification and improvement in the rate of heat transfer. The use of two or more techniques (passive and/or active) in conjunction constitutes compound augmentation techniques. The effectiveness of any of these methods is strongly dependent on the mode of heat transfer (single-phase free or forced convection, pool boiling, forced convection boiling or condensation, and convective mass transfer), and type and process application of the heat exchanger.

II. LITERATURE SURVEY

In the present work S. Naga Sarada[1], shows the results obtained from experimental investigations of the augmentation of turbulent flow heat transfer in a horizontal tube by means of varying width twisted tape inserts with air as the working fluid. In order to reduce excessive pressure drops associated with full width twisted tape inserts, with less corresponding reduction in heat transfer coefficients, reduced width twisted tapes of widths ranging from 10 mm to 22 mm, which are lower than the tube inside diameter of 27.5 mm are used. Experiments were carried out for plain tube with/without twisted tape insert at constant wall heat flux and different mass flow rates. The twisted tapes are of three different twist ratios (3, 4 and 5) each with five different widths (26-full width, 22, 18, 14 and 10 mm) respectively. The Reynolds number varied from 6000 to 13500. Both heat transfer coefficient and pressure drop are calculated and the results are compared with those of plain tube. It was found that the enhancement of heat transfer with twisted tape inserts as compared to plain tube varied from 36 to 48% for full width (26mm) and 33 to 39% for reduced width (22 mm) inserts.Correlations are developed for friction factors and Nusselt numbers for a fully developed turbulent swirl flow, which are applicable to full width as well as reduced width twisted tapes, using a modified twist ratio as pitch to width ratio of the tape.

In the work done by Y. Raja Sekhar[2], Solar thermal energy is currently used for low temperature heating applications using flat plate collectors. The absorbed solar energy is transferred to the working fluid flowing in the pipe. The performance of the system is influenced by heat transfer from tube to working fluid, with minimum convective losses, which has to be considered as one of the primary design factor. In tube and channel flows, to enhance the rate of heat transfer to the working fluid, passive augmentation techniques such as twisted tapes and swirl generators are used in the fluid flow path. In this paper, convective heat transfer analysis for a horizontal circular pipe with fluid in mixed laminar flow range is performed using experimental simulation under constant heat flux boundary condition. The variation of heat transfer coefficient and pressure drop in the pipe flow for water and water based Al2O3 nanofluids at different volume concentrations and twisted tapes are studied. The dependence of particle concentration and Reynolds number for enhancement in heat transfer and increase in the pumping power due to pressure drop is analysed in the range of parameters considered.

III. MASS FLOW RATE CALCULATIONS FOR DIFFERENT REYNOLDS NUMBER

Fluid – Air

$Re=\rho vL/\mu$
ρ= density Kg/m^3
V=Velocity m/s
L=length of the tube
μ =viscosity kg/ms = 1.7894 e^{-5} Kg/ms
d = inner diameter of tube
Mass flow rate $m=\rho v A$
A = Cross Sectional Area = $\Pi/4 * d^2$

$R_e = m * L / \mu * A$

IV. 3D MODELING OF TWISTED TUBE INSERTS

The references for modeling is taken from, [1] Enhancement of heat transfer using varying width twisted tape inserts, S. Naga Sarada, International Journal of Engineering, Science and Technology, Vol. 2, No. 6, 2010, pp. 107-118.

Tube – Outer dia = 33.9mm and Inner dia = 27.5mm
Twisted Tape Pitch – 82.5mm
Twisted Tape Widths – 10mm, 14mm and 18mm

Fig - 3d model of Tube

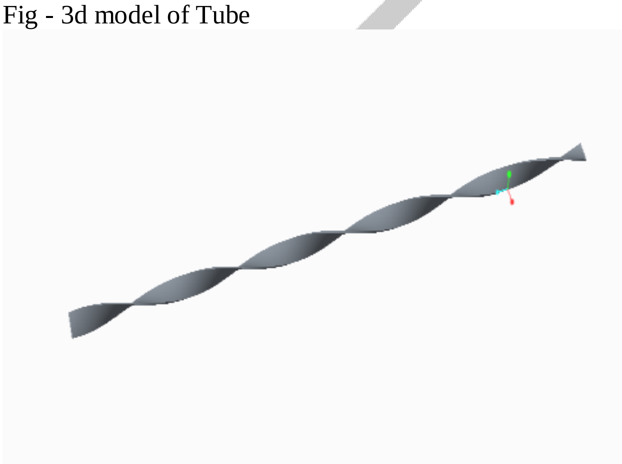


Fig – 3D model of Twisted tube insert 10mm width

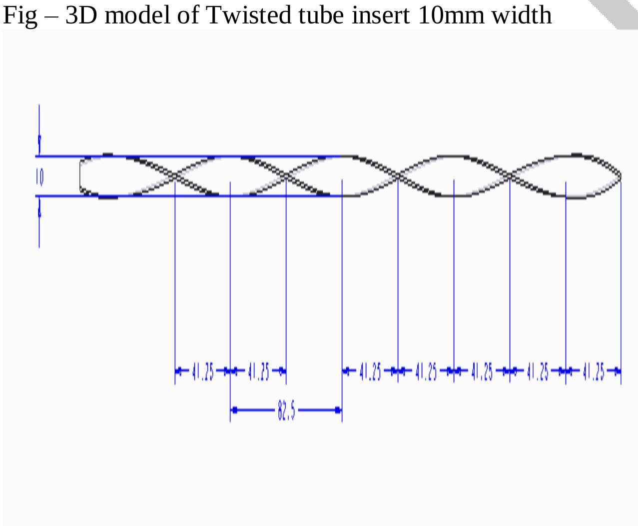


Fig – 2D Drawing of Twisted tube insert 10mm width

Fig – Assembly of Tube and Insert

V. CFD ANALYSIS ON HORIZONTAL TUBE WITH TWISTED INSERTS

The boundary conditions for analysis are mass flow rate, pressure and heat flux.
The mass flow rates are calculated theoretically for different Reynolds number 6000, 10000 and 12000.
The heat input is 40W.

TWISTED TAPE WIDTH - 10mm
BOUNDARY CONDITIONS

Mass flow rate(Kg/s)	0.000159,0.000265, 0.000318
Inlet pressure(Pa)	101325 pa
Heat Flux on outer wall	939.44 W/m2

Heat Flux = Heat / Π*D*L
Reynolds number – 6000, Mass flow rate 0.000159 kg/s

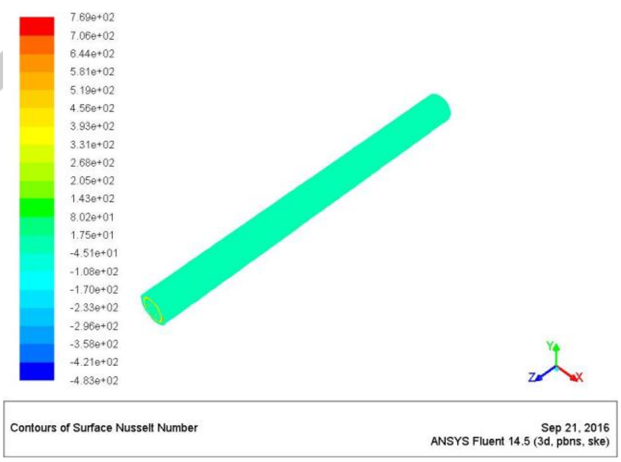


Fig – Contour of Surface Nusselt Number

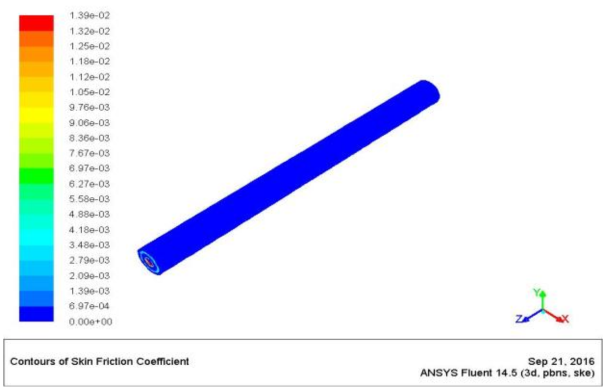


Fig – Contour of Skin Friction Coefficient

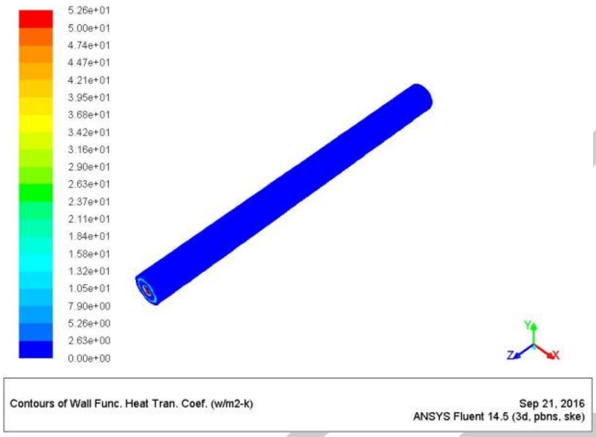
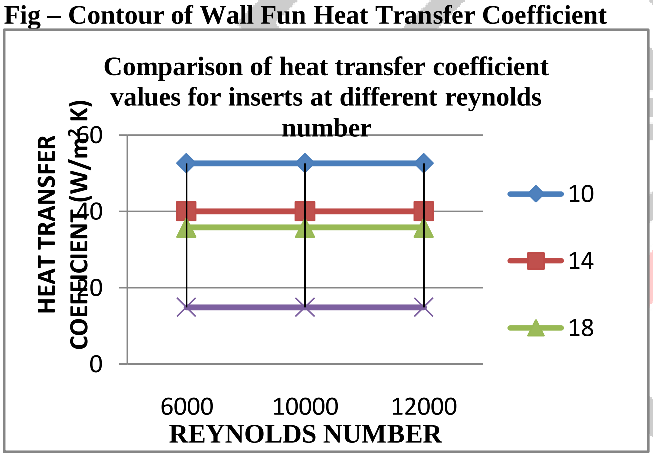
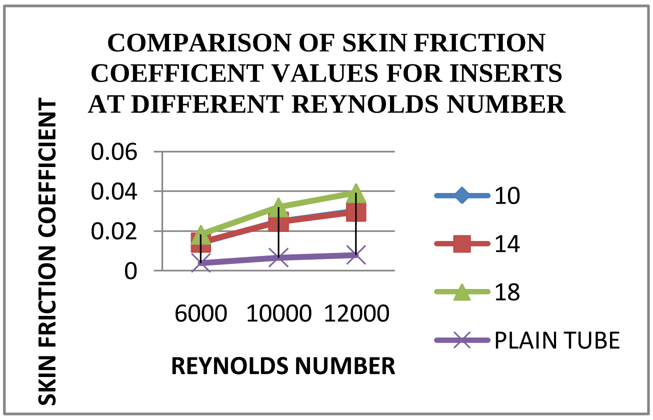
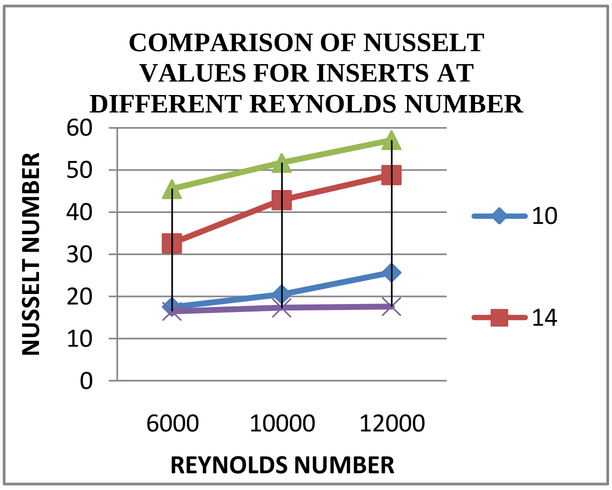
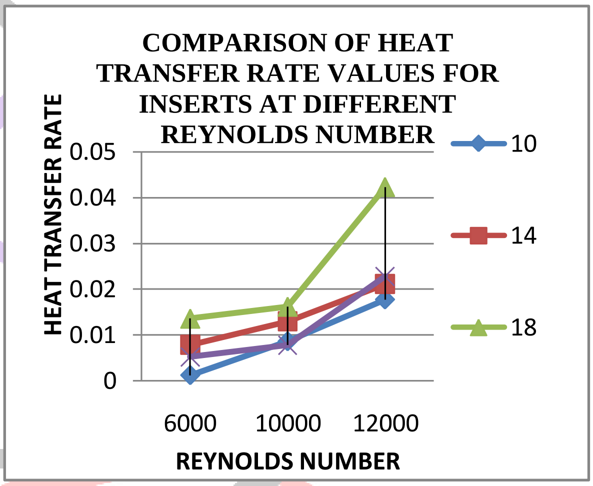


Fig – Contour of Wall Fun Heat Transfer Coefficient



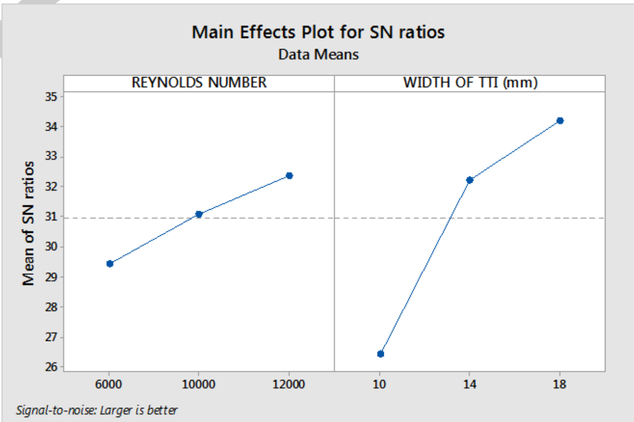






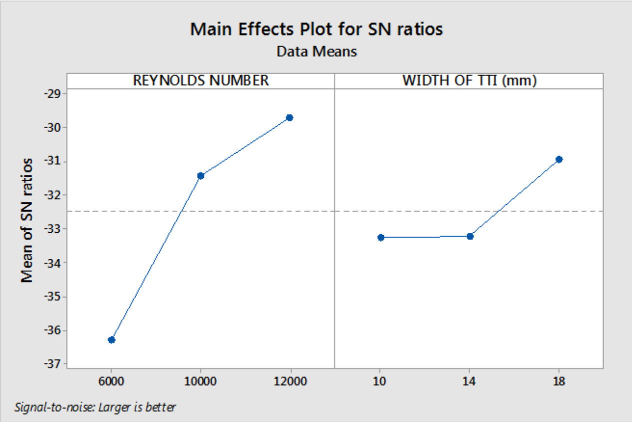
VI. OPTIMIZATION OF REYNOLDS NUMBER AND WIDTH TWISTED TUBE INSERTS USING MINI TAB SOFTWARE

Optimization for Nusselt number



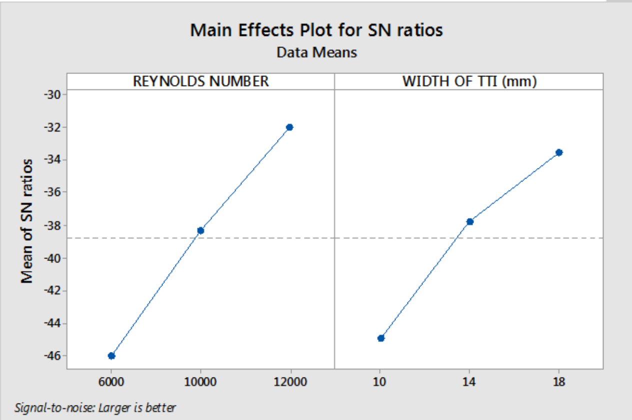
Effect of Reynolds Number and Width of Twisted Tube Inserts for S/N ratio

Optimization for skin friction coefficient



Effect of Reynolds Number and Width of Twisted Tube Inserts for S/N ratio

Optimization for heat transfer rate



Effect of Reynolds Number and Width of Twisted Tube Inserts for S/N ratio.

VII. RESULT TABLES

Width 10 mm

Reynolds number	Nusselt number	Heat transfer coefficient (W/m^2 K)	Skin Friction Coefficient	Total heat transfer rate (W)
6000	17.5	52.63	0.0139	0.0012
10000	20.5	52.63	0.0246	0.008588
12000	25.7	52.63	0.03	0.0177

Width 14 mm

Reynolds number	Nusselt number	Heat transfer coefficient (W/m^2 K)	Skin Friction Coefficient	Total heat transfer rate (W)
6000	32.6	40.035	0.01423	0.0078
10000	42.8	40.035	0.0245	0.01289
12000	48.8	40.035	0.0296	0.021192

Width 18 mm

Reynolds number	Nusselt number	Heat transfer coefficient (W/m^2 K)	Skin Friction Coefficient	Total heat transfer rate (W)
6000	45.5	35.7866	0.01818	0.0136
10000	51.7	35.7866	0.032	0.0162
12000	57.1	35.7866	0.03916	0.042292

PLAIN TUBE

Reynolds number	Nusselt number	Heat transfer coefficient (W/m^2 K)	Skin Friction Coefficient	Total heat transfer rate (W)
6000	16.5	14.861	0.00386	0.0052
10000	17.3	14.861	0.006513	0.0078
12000	17.6	14.861	0.00784	0.0228

VIII. CONCLUSION

By observing analysis results, the following conclusions can be made:

The Nusselt Number is more when twisted tube inserts are used than that of plain tube. Nusselt Number is increasing by increase of Reynolds Number and with increase of width of twisted tube insert. The Nusselt Number is increasing by 46% for 10mm width than that of plain tube at Reynolds Number 12000, increasing by 64% for 14mm width than that of plain tube at Reynolds Number 12000 and increasing by 69% for 18mm width than that of plain tube at Reynolds Number 12000. The heat transfer coefficient is not changed with change of Reynolds Number and increasing with decrease of width of twisted tube insert.

The Skin Friction Coefficient is more when twisted tube inserts are used than that of plain tube and Skin Friction Coefficient is increasing by increase of Reynolds Number and with increase of width of twisted tube insert. The Skin Friction Coefficient is increasing by 73.86% for 10mm width than that of plain tube at Reynolds Number 12000, increasing by 73.6% for 14mm width than that of plain tube at Reynolds Number 12000 and increasing by 79.9% for 18mm width than that of plain tube at Reynolds Number 12000. The Heat Transfer Rate is almost equal for twisted tube inserts and that of plain tube. Heat Transfer Rate is increasing by increase of Reynolds Number and increasing with increase of width of twisted tube insert. By observing the Taguchi Method results, for higher Nusselt Number, Skin Friction Coefficient and Heat Transfer Rate, the optimum Reynolds Number is 12000, optimum width of twisted tube insert is 18mm.

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