ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Analysis to Determine Heat Transfer Using Twisted Tape Inserts In a Horizontal Tube

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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 7000, 9000, 11000, 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.

Keywords—heat transfer; twisted tape; horizontal tube; FEA

Introduction

The process of improving the performance of a heat transfer system is referred as the heat transfer enhancement technique .In recent years, the high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment .The major challenge in designing a heat transfer is to make the equipment compact and achieve a high heat transfer rate using minimum pumping power. The subject of heat transfer growth in heat exchanger is serious interest in the design of effective and economical heat exchanger. Augmentation techniques increase convective heat transfer by reducing thermal resistance in a heat exchanger. A decrease in heat transfer surface area, size, and hence weight of heat exchanger for a given heat duty and pressure drop. The heat transfer can be increased by the following different augmentation techniques. They are classified as (i) Passive Techniques (ii) Active Techniques (iii) Compound Techniques.

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.

Literature survey

In the present work S. Naga Sarada[1], Experiments were carried out for plain tube with/without twisted tape insert at constant wall heat flux and different mass

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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.

Mass flow rate calculations for different Reynolds number

Fluid – Air

Reynolds number – 7000

$Re=\rho vL/\mu$

ρ= density Kg/m3

V=Velocity m/s

L=length of the tube

µ =viscosity kg/ms

d = inner diameter of tube

Mass flow rate m=ρv A

A = Cross Sectional Area = Π/4 * d2 = 0.000593m2

Re = m * L / µ * A

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

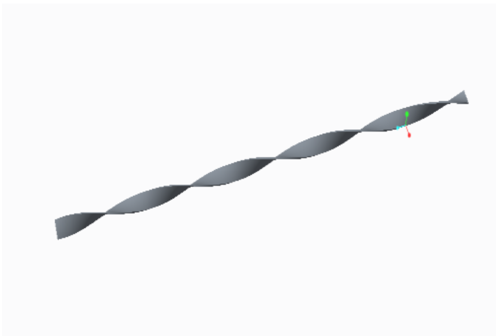

Fig – 3D model of twisted tube insert 10mm width

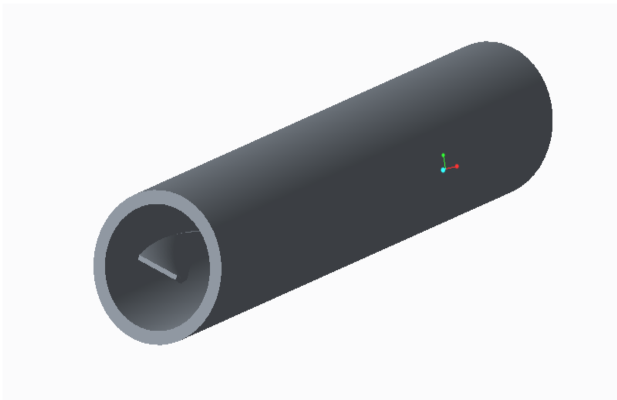

Fig – Assembly of Tube and Insert

CFD ANALYSIS ON HORIZONTAL TUBE WITH TWISTED INSERTS

The boundary conditions for analysis are mass flow rate, pressure and heat flux.

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The mass flow rates are calculated theoretically for different Reynolds number7000, 9000 and 11000. The heat input is 40W.
TWISTED TAPE WIDTH - 10mm
BOUNDARY CONDITIONS

Mass flow rate(Kg/s)	0.0001856, 0.00023875, 0.0002918
Inlet pressure(Pa)	101325 pa
Heat Flux on outer wall	939.44 W/m2

Heat Flux = Heat / П*D*L

Reynolds number – 7000, Mass flow rate 0.0001856 kg/s

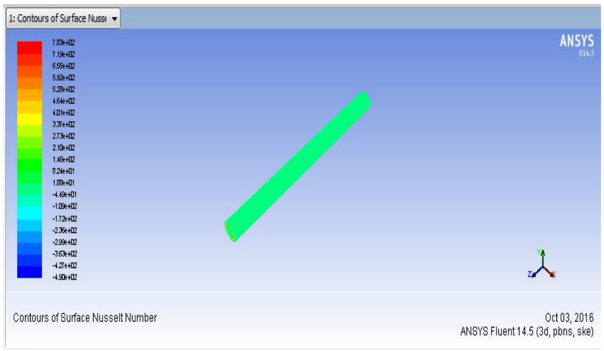

Fig – Contour of Surface Nusselt Number

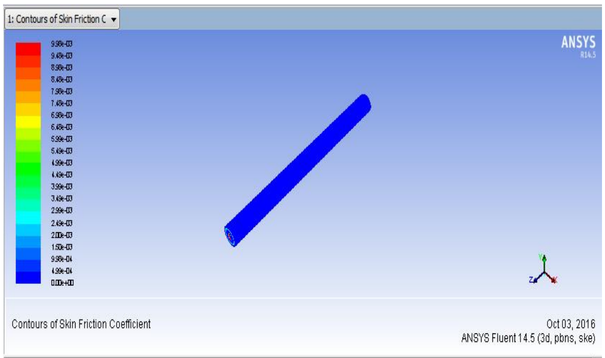

Fig – Contour of Skin Friction Coefficient

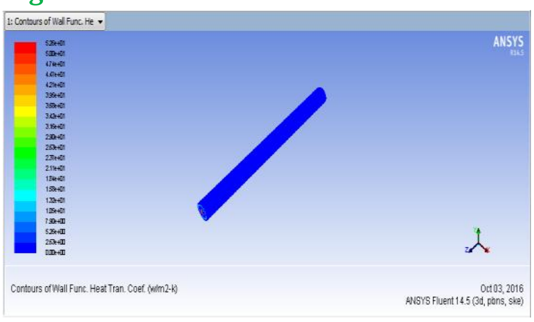

Fig – Contour of Wall Func Heat Transfer Coefficient

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                 Total Heat Transfer Rate
                        (w)
 ----------------------------------------
contact_region-src                      0
contact_region-trg                      0
contact_region_2-src                    0
contact_region_2-trg                    0
inlet                           0.5048691
outer_wall                      39.905918
outlet                         -40.404041
wall-10                        -39.904961
wall-10-shadow                  39.899258
wall-18                                 0
wall-19                                 0
wall-20                     6.9932314e-05
wall-20-shadow             -5.7329657e-05
wall-21                                 0
wall-22                                 0
wall-insert                             0
wall-tube                               0
 ----------------     --------------------
  Net                     0.0010555647
```

RESULTS TABLES

Width 10 mm

Reynolds number	Nusselt number	Heat transfer coefficient (W/m² K)	Skin Friction Coefficient	Total heat transfer rate (W)
7000	18.8	52.63	9.98e-3	0.0010555
9000	21.8	52.63	2.19e-2	0.0026066726
11000	28.6	52.63	2.73e-2	0.020506951

Width 14 mm

Reynolds number	Nusselt number	Heat transfer coefficient (W/m² K)	Skin Friction Coefficient	Total heat transfer rate (W)
7000	35.8	40.035	1.68e-2	0.0018957
9000	40.8	40.035	2.2e-2	0.02619819
11000	46.9	40.035	2.71e-2	0.028398536

Width 18 mm

Reynolds number	Nusselt number	Heat transfer coefficient (W/m² K)	Skin Friction Coefficient	Total heat transfer rate (W)
7000	46.8	35.7866	2.16e-2	0.0028705338
9000	50.4	35.7866	2.85e-2	0.0031318831
11000	54.1	35.7866	3.55e-2	0.018449228

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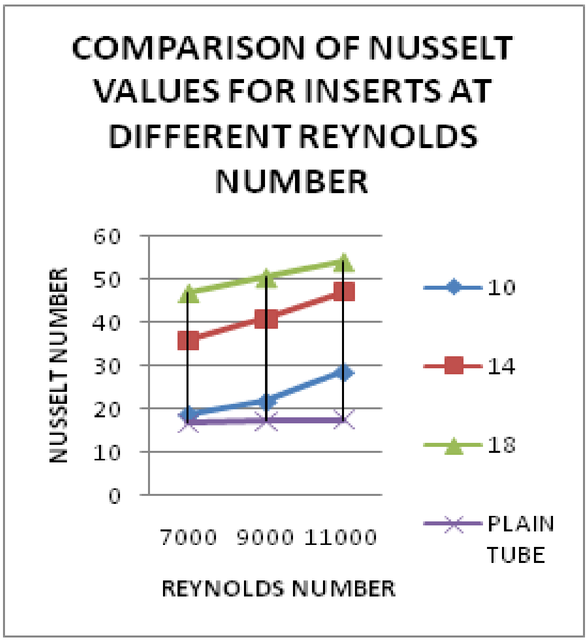
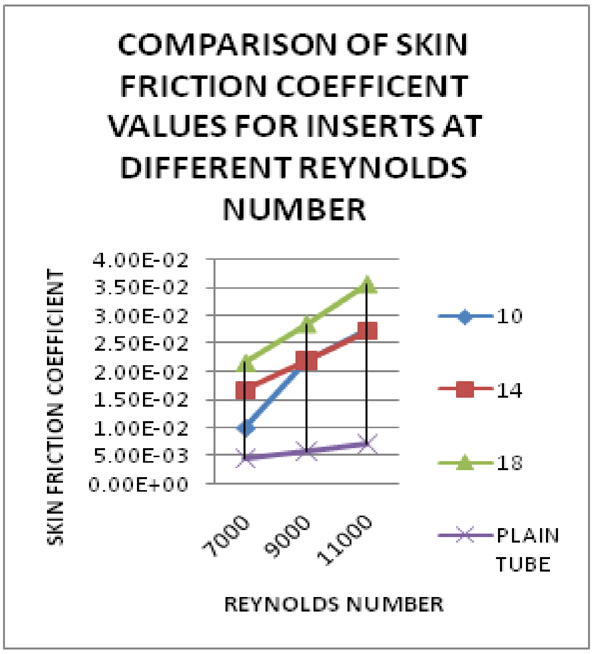
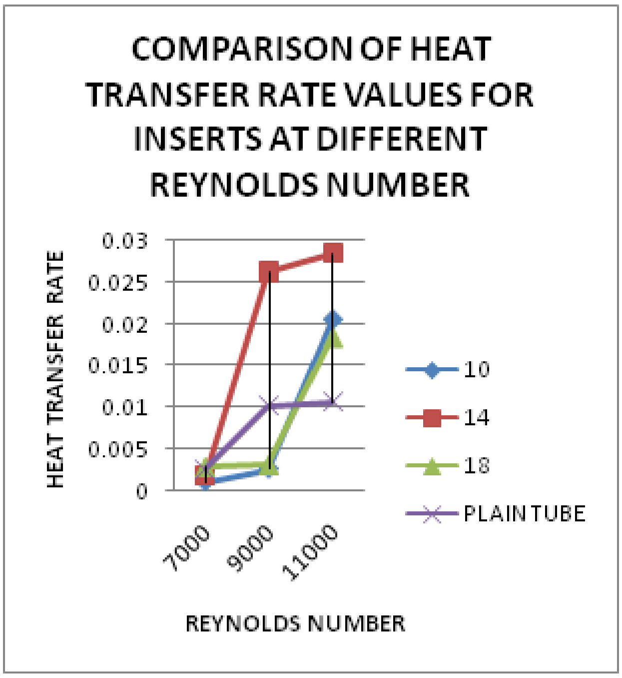
PLAIN TUBE

Reynolds number	Nusselt number	Heat transfer coefficient (W/m² K)	Skin Friction Coefficient	Total heat transfer rate (W)
7000	16.8	14.861	4.53e-3	0.0025437474
9000	17.2	14.861	5.86e-3	0.010138541
11000	17.5	14.861	7.18e-3	0.010565549







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 38.8% for 10mm width than that of plain tube at Reynolds Number 11000, increasing by 62.68% for 14mm width than that of plain tube at Reynolds Number 11000 and increasing by 67.65% for 18mm width than that of plain tube at Reynolds Number 11000. 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 79% for 10mm width than that of plain tube at Reynolds Number 11000, increasing by 79.43% for 14mm width than that of plain tube at Reynolds Number 11000 and increasing by 78.366% for 18mm width than that of plain tube at Reynolds Number 11000. 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.

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