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Heat Transfer Enhancement by using Different Types of Inserts in Forced Convection

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Abstract: In the present work heat transfer coefficient were experimentally investigated by using twisted tape and triangular fin type inserts in forced convection. The turbulent flow was created by inserting the twisted tapes and triangular fin type inserts into the test pipe creating high rate of turbulence in pipe, which results in increasing heat transfer coefficient. The insert used in this experiment are plain, notched and circular fin type. The length and width of insert was 1000 mm and 16 mm for twisted tape. For circular fin type insert, the pipe of 9 mm diameter is used, on which 6 mm equilateral triangles are mounted at 90° as a fin. The outside diameter & inside diameter of test pipe is 25.5 mm & 22.5 mm respectively. The length of test section is 1000 mm. The bulk mean temperatures at various positions are used for different flow rate of air. From the obtained results the new Correlations for Nusselt number and friction factor are developed for twisted tape inserts. The Reynolds number is varied from 37000 to 60000. The results of Different type of inserts have been compared with the values for the smooth tube. It showed that the highest heat transfer rate was achieved for plain twisted tape.
Keywords: Heat Transfer Coefficient, Twisted Tapes and Triangular Fin Type Insert, Turbulent Flow, Friction Factor, Reynolds Number, Nusselt Number.

I. INTRODUCTION

In heat exchanger, the enthalpy is transferred between two or more fluids, at different temperatures. The use of heat exchangers is in various industrial processes for heating and cooling applications such as air conditioning and refrigeration systems, heat recovery processes, food and dairy processes, chemical process plants etc. The major challenge in designing a heat exchanger is to make the equipment more compact and achieve a high heat transfer rate using minimum pumping power. Techniques for heat transfer augmentation are relevant to several engineering applications. In recent years, the high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. Sometimes there is a need for miniaturization of a heat exchanger in specific applications, such as space application, through an augmentation of heat transfer. These problems are more common for heat exchangers used in marine applications and in chemical industries. The heat transfer rate can be improved by introducing a disturbance in the fluid flow thereby breaking the viscous and thermal boundary layer. However, in the process pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, several techniques have been proposed in recent years and are discussed under the classification section. The heat transfer enhancement techniques are performed in widespread applications. The results of those studies have been shown that although heat transfer efficiencies are improved, the flow frictions are also considerably increased. In this report the various inserts are used for heat transfer enhancement. The strips are expected to induce a rapid mixing and a high turbulent and longitudinal vortex flow like a delta wing, of course, resulting in an excellent rate of heat transfer in the tube.

II. EXPERIMENTAL SET-UP

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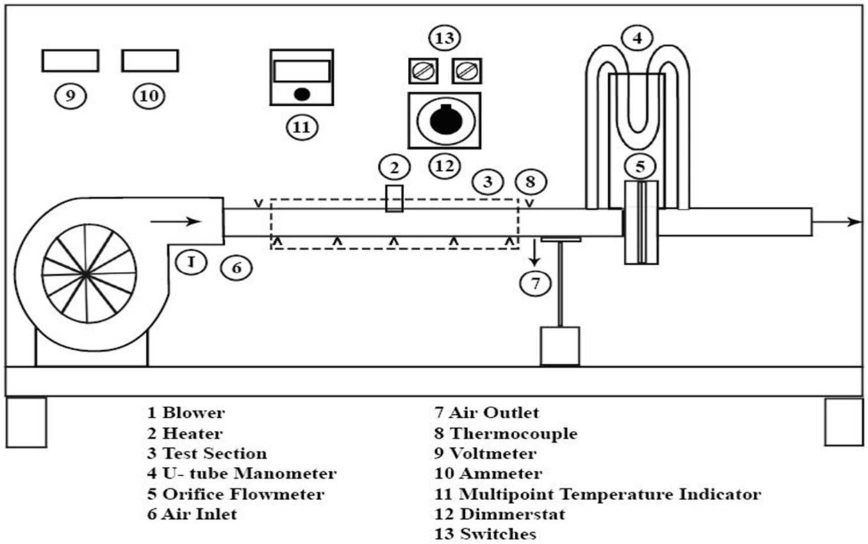


Fig. 1 The schematic diagram of experimentation

Figure 1 shows a schematic representation of forced convection apparatus. It consists of test section, u tube manometer, and blower to force the air inside test section. Nichrome wire band type heater is used to heat the air supply. A test section consists of 1000 mm long copper pipe having inner diameter 22.5 mm and outer diameter 25.5 mm, wall thickness 3 mm. The different types of inserts are inserted from one end of the tube. The thermal performance of the heat exchanger is analyses.

III. PROCEDURE

Put the dimmerstat to zero position and the switch on the main electric supply. Start the blower and adjust the air flow by means of valve to some desired difference in manometer level. (Preferably open control valve fully). Start the heating of the test section by adjusting desired heater input with dimmerstat. Take the readings of all 7 thermocouples at the interval of 10 minutes until the steady state is reached (last 2 readings should be same). Also note down the heater input (In terms of voltage and current). Repeat the procedure for different inserts. Calculate the heat transfer rate with the help of readings taken for with & without inserts.

Fig. 3 Plain twisted tape

Fig. 4 Notched twisted tape

Fig. 5 Triangular fin type insert

IV. SPECIFICATIONS OF INSERTS:

A. For Twisted Tapes,

1) Width of tape, W = 16 mm,
2) Thickness of inserts, t = 2mm,
3) Length of tape, L = 1150 mm,
4) Pitch, P = 80 mm,
5) Notch type = V notches,
6) Twist ratio, TR = 4.413

B. For Triangular fin Type Insert,

1) Length of tape, L = 1150 mm,
2) Spacing between fins, S = 160 mm,
3) Pipe diameter, d = 9 mm

C. Sample Calculations

*1)*Bulk mean temperature of air: $T_{mean} = (T_1+T_7)/2$

2) Properties of Air at T_{mean} :

ρ_{air} = Density of air

C_{pa} = Specific heat capacity of air

*3)*Surface Temperature: $T_{surface}= (T_2+T_3+T_4+T_5+T_6)/5$

*4)*Properties of Air at F_{ilm},

μ = Dynamic viscosity of air(Ns/m^2)

k =Thermal conductivity of air(W/mK)

*5)*Average Velocity of Air(m/s): $\sqrt{2gH}$ ($H = H_{water}*1000/\rho_{air}$)

7) Mass flow rate of air = $\rho_{air}(V.\pi r^2)C_d$

Where, C_d = Coefficient of discharge

7)$T_{film} = (T_s+T_{mean})/2$

Properties of air at T_{film} from standard air properties at 1atm pressure

$$Re_{th} = \frac{\rho V d}{\mu}$$

$$Nu_{th} = 0.193(Re)^{0.618}(Pr)^{(1/3)}$$

8) Friction factor, (f)

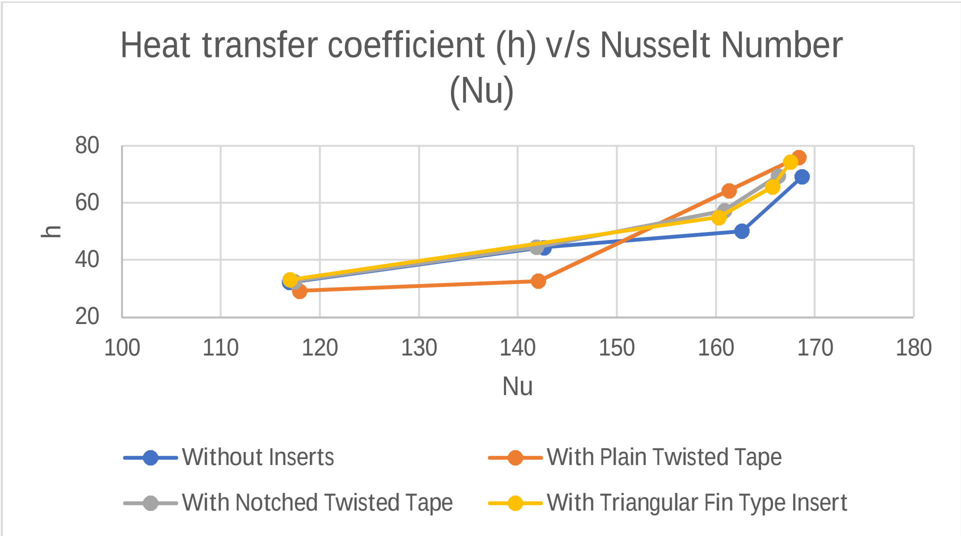
$$f = (0.0791)/((Re)^{0.25}$$

$$Q = m_a C_{pa} (T_7 - T_1)$$

$$Q = h(\pi d L)*(T_{surface} - T_{mean})$$

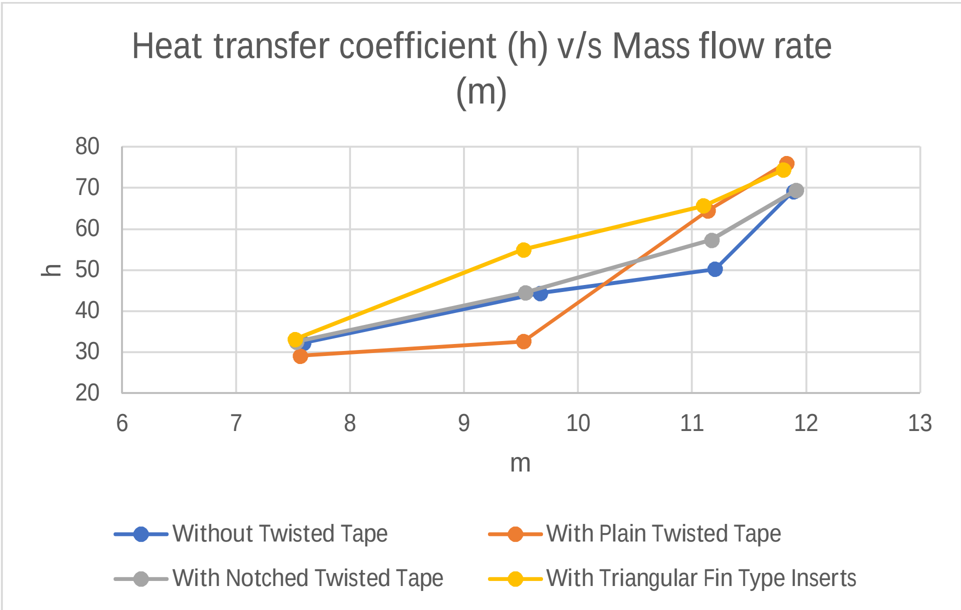
IV. RESULT AND DISCUSSION

After the experimental study, the Nusselt numbers, heat transfer coefficient and friction factors were calculated for plain tube and tube with inserts. These results were compared with the correlation of Dittus and Boelter for Nusselt number & Koo's equation for friction factor. Graph 1 shows the relation of Heat transfer coefficient vs Nusselt number. It is found that there is an increase in heat transfer coefficient with respect to Nusselt number.



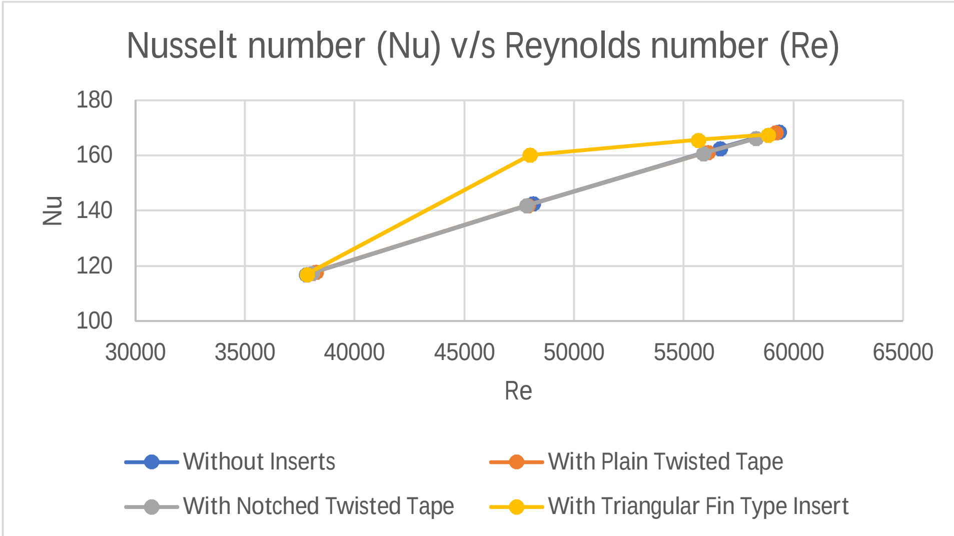
Graph 1 Heat transfer coefficient (h) v/s Nusselt Number (Nu)

Graph 2 shows the relation of heat transfer coefficient v/s mass flow rate. It is found that heat transfer coefficient is directly proportional to mass flow rate. As mass flow rate increases, heat transfer coefficient also increases.



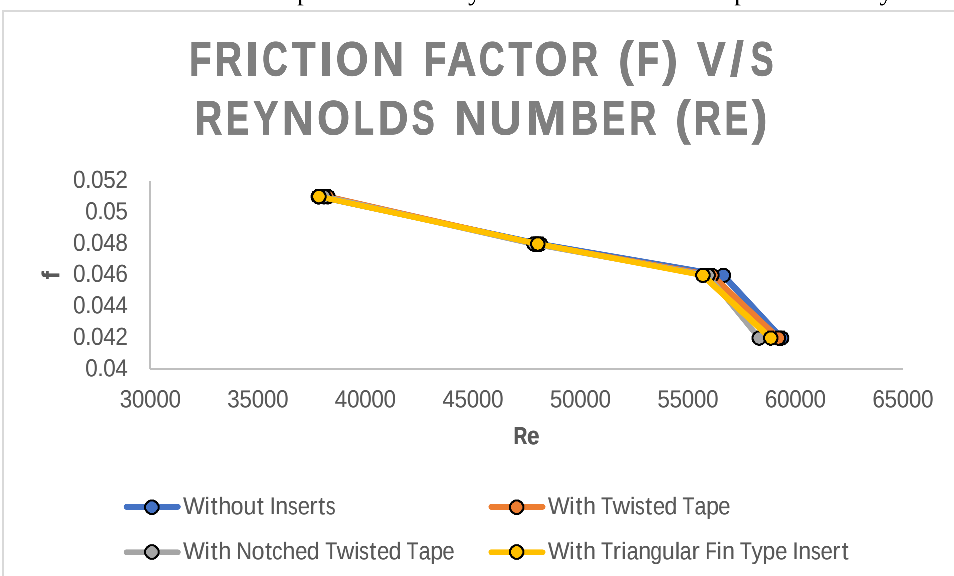
Graph 2 Heat transfer coefficient (h) v/s Mass flow rate (m)

Graph 3 shows the relation of Nusselt number Vs Reynolds number for plain tube. It is found that there is linear behaviour of Nusselt number along Reynolds number. Nusselt number is a function of Reynolds number.



Graph 3 Nusselt Number (Nu) v/s Reynolds Number (Re)

Graph 4 shows the relation of friction factor v/s Reynolds number. It is found that friction factor is inversely proportional to Reynolds number. The value of friction factor depends on the Reynolds number. It is independent of any other variable.



Graph 4 Friction Factor (f) v/s Reynolds Number (Re)

V. CONCLUSIONS

The convective heat transfer performance a flow characteristic of fluids flowing in a pipe has been experimentally investigated. The effect of Reynolds number on the heat transfer performance and flow behaviour of the fluid has been experimentally determined.

It is found that with increase in mass flow rate, Nusselt number also increases, but the value of friction factor decreases. New correlations for the Nusselt number and heat transfer coefficient based on the present experimental data are given for practical uses. The agreement between the results obtained from the experimental data and those obtained from the proposed correlations is reasonable

A_o	Area of orifice, m^2	Re	Reynolds number
C_p	Specific heat, J/Kg K	V	Velocity of air, m/sec
f	Friction factor	L	Length of test pipe, mm
h	Heat transfer coefficient, W/m2K	ν	Kinematic viscosity of water ,m^2/sec
k	Thermal conductivity, W/mK	ρ	Density of water, kg/m^3

Nu	Nussult number	Pr	Prandlt number
q	Air heating rate, kcal/kg x	H	Manometric head, mm of water
C_d	Coefficient of discharge		

REFERENCES

[1] Raj M. Manglik & Arthur E. Bergles "Characterization of Twisted-Tape-Induced Helical Swirl Flows For Enhancement Of Forced Convective Heat Transfer In Single-Phase And Two-Phase Flows" Fellow ASME Thermal-Fluids & Thermal Processing Laboratory, June 2013, vol. 5.

[2] Giovanni Tanda and Roberto Abram "Forced Convection Heat Transfer in Channels with Rib Tabulators Inclined at 45 deg" Journal of Turbo machinery fellow ASME, APRIL 2009, Vol. 131 / 021012- 1 to 10

[3] Jozef Cernecky, Jan Koniar and Zuzana Brodnianska "The Effect of Heat Transfer Area Roughness on Heat Transfer Enhancement by Forced Convection" Journal of Heat Transfer fellow ASME APRIL 2014, Vol. 136 / 041901-1 to 8.

[4] M. Moawed, Experimental study of forced convection from helical coiled tubes with different parameters, Energy Conversion and Management 52 (2011) 1150–1156

[5] Dr. Anirudh Gupta, Mayank Uniyal "Review of Heat Transfer Augmentation through Different Passive Intensifier Methods" IOSR Journal of Mechanical and Civil Engineering (IOSRJMCE) ISSN: 2278-1684 Volume 1, Issue 4 (July-Aug 2012), PP 14-21.

[6] M.M.K. Bhuiya, M.S.U. Chowdhury, J.U. Ahamed, M.J.H. Khan, M.A.R. Sarkar, M.A. Kalam, H.H. Masjuki , M. Shahabuddin, Heat transfer performance for turbulent flow through a tube using double helical tape inserts, International Communications in Heat and Mass Transfer 39 (2012) 818–825

[7] Naga Sarada S., Kalyani K. Radha and A. V. S. Raju, experimental investigation in circular tube to enhance turbulent heat transfer using mesh inserts, ARPN Journal of Engineering and Applied Sciences, VOL. 4, NO. 5, JULY 2009, ISSN 1819-6608

[8] Amnart Boonloi, Withada Jedsadaratanachai, Turbulent Forced convection and Heat Transfer Characteristic in a Circular Tube with Modified-Twisted Tapes, Hindawi Publishing Corporation Journal of thermodynamics Volume 2016, Article ID 8235375

[9] S.G. Mushan, B.E.Patil, N.K.Sane, D.V.Ghewade, heat augmentation using nonmetallic insert sin forced convection, IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE) e-ISSN: 2278-1684,p-ISSN: 2320-334X

[10] Suhas V. Patil and P. V.Vijay Babu "Heat Transfer Augmentation in a Circular tube and Square duct Fitted with Swirl Flow Generators: A Review" International Journal of Chemical Engineering and Applications, Vol. 2 , No. 5 , October 2011

[11] A Dewan, P Mahanta, K Sumithra Raju & P Suresh Kumar, "Review of passive heat transfer augmentation techniques", Proc.Instn Mech. Engrs. Vol. 218 Part A: J Power & Energy, (2004), 509-527.

[12] Kreith F., Timmerhaus K., Lior N., Shaw H., Shah R.K., Bell K. J.,etal..“Applications.” The CRC Handbook of Thermal Engineering. Ed. Frank Kreith. Arthur E. Bergles. Boca Raton: CRC Press LLC, 2000, 408-457.