ISSN 2395-1621

Performance Analysis of Double Coil Wire Inserts for Heat Transfer in a Circular Tube

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ABSTRACT

The present work shows the results obtained from experimental investigations of the augmentation of turbulent flow heat transfer in a horizontal tube by means of double coil wire inserts with air as the working fluid. Experimental work were carried out for plain tube with/without double coil wire insert at constant wall heat flux and different mass flow rates. The double coil wire insert are four different pitch of 12,14,16 & 18 mm and three different wire diameter of 1.6,2 and 2.6 mm for outer coil and 0.9,1.2 and 1.5 mm for inner coil wire diameter with 25 mm outer coil and 11 mm inner coil diameter. The Reynolds number varied from 4000 to 7242. 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 double coil wire inserts as compared to plain tube varied from 3% to 46 % for various inserts. Also the results are compared with the plane tube without insert.

***Keywords*— a Enhancement efficiency, heat transfer, Double coil wire inserts, High/ low density, pressure drop**

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

Heat exchangers are widely used 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 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. Furthermore, sometimes there is a need for miniaturization of a heat exchanger in specific applications, such as space application, through an augmentation of heat transfer. For example, a heat exchanger for an ocean thermal energy conversion (OTEC) plant requires a heat transfer surface area of the order of 10000 m^2/MW . Therefore, an increase in the efficiency of the heat exchanger through an augmentation technique may result in a considerable saving in the material cost. Furthermore, as a heat exchanger becomes older, the resistance to heat transfer increases owing to fouling or scaling. These problems are more common for heat exchangers used in marine applications and in chemical industries. In some specific applications, such as heat exchangers dealing with fluids of low thermal conductivity (gases and oils) and desalination plants, there is a need to increase the heat transfer rate. 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.

II. EXPERIMENTAL SETUP

The schematic diagram of experimental set-up is given in Fig.1. The experimental facility includes a blower, an orifice

meter to measure the volumetric flow rate, the heat transfer test tube (700 mm). The MS test tube 26 mm inner diameter (D1), 30 mm outer diameter (D2), and 2 mm thickness (t). The double coil wire inserts are tested in this experiment, with four different pitch as 12,14, 16 & 18 mm but with three different wire diameter as 1.6,2 and 2.6 mm outer coil diameter and 0.9, 1.2 and 1.5 inner coil diameter. They are fabricated from copper. The schematic figure of the test tube with double coil wire insert is given in Fig.2. The double coil wire inserts contained in the experimental study are shown in Fig. 3. A 0.24 hp blower is used to force air through the test tube. Uniform heat flux is applied to external surface of the test tube by means of heating with electrical winding, whose output power is controlled by a variac transformer to supply constant heat flux along the entire section of the test tube. The outer surface of the test tube is well insulated with glass wool to reduce the convective heat loss to the surroundings. The external surface temperatures of the test tube wall are measured by 6 K-type thermocouples, which are placed on the outer wall of the test tube. Also, the inlet and outlet temperatures of the bulk air are measured by two K-type thermocouples at given points. An another manometer is used to measure pressure drop across the test tube. After air passes the test tube, it enters to the orifice meter for determining volumetric flow rate readings. For this purpose a separate U-tube manometer is placed across orifice meter. The volumetric flow rate of air supplied from the blower is controlled by varying control valve position. The experiments are conducted by varying the flow rate in terms of Reynolds numbers from 4181 to 7242 of the bulk air. The test tube is heated from the external surface during the experiments, and the data of temperatures, volumetric flow rate, pressure drop of the bulk air and electrical output are recorded after the system is approached to the steady state condition. The Nusselt number, Reynolds number, friction factor, heat transfer enhancement are calculated based on the average outer wall temperatures and the inlet and outlet air temperatures.

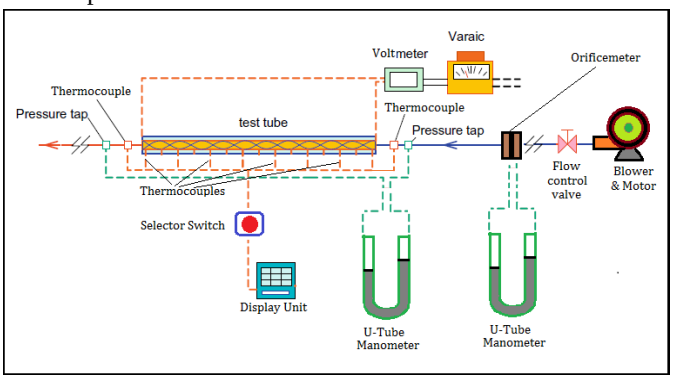

Figure 1: Experimental setup block diagram

TABLE I
Detailed specifications of coil inserts

INSERT NO.	Size	W.D.	O.D.	I.D.	Pitch
I	**Outer**	1.6	25	21.9	12
	Inner	0.9	11	9.2	12
II	**Outer**	1.6	25	21.7	14
	Inner	0.9	11	9.2	14
III	**Outer**	1.6	25	21.9	16
	Inner	0.9	11	9.2	16
IV	**Outer**	1.6	25	21.9	18
	Inner	0.9	11	9.2	18
V	**Outer**	2	25	21	12
	Inner	1.2	11	8.6	12
VI	**Outer**	2	25	21	14
	Inner	1.2	11	8.6	14
VII	**Outer**	2	25	21.08	16
	Inner	1.2	11	8.6	16
VIII	**Outer**	2	25.4	21.2	18
	Inner	1.2	11	8.6	18
IX	**Outer**	2.6	26	20.8	12
	Inner	1.5	11.3	8.2	12
X	**Outer**	2.6	26	20.8	14
	Inner	1.5	11	8.3	14
XI	**Outer**	2.6	26	20.8	16
	Inner	1.5	11	8	16
XII	**Outer**	2.6	26	20.8	18
	Inner	1.5	11	8	18

III.DATA COLLECTION ANALYSIS

The data reduction of the obtained results is summarized in the following procedures:

A. Heat Transfer Calculations

$$T_s = (T_2+T_3+T_4+T_5+T_6+T_7)/6 \quad (1)$$

$$T_b = (T_1+T_8)/2 \quad (2)$$

Equivalent height of air column, $h_{air} = (\rho_w * h_w)/\rho_a$ (3)

Discharge of air, $Q_a = C_d * A_o \sqrt{2 * g * h_{air}}$ (4)

Velocity of air flow, $V = Q_a / A$ (5)

Reynolds number, $Re = V D/\nu$ (6)

$$Q = \dot{m} * C_p * (T_8 - T_1) \quad (7)$$

$$h = \frac{Q}{A\,(T_s - T_b)} \quad (8)$$

$$Nu = \frac{h\,D}{k} \quad (9)$$

$$f = \frac{\Delta P}{\frac{L}{D} * \frac{\rho_a V^2}{2}} \quad (10)$$

$$\eta = \frac{(Nu_i/Nu)}{(f_i/f)^{0.333}} \quad (11)$$

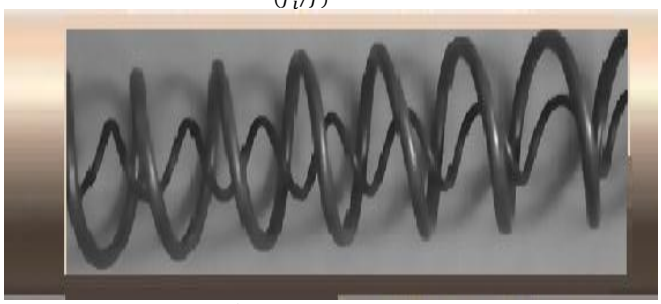

Figure 2: Schematic of test tube with double coil wire inserted

B. Validation experiments of plain tube

In this study, experimental results of Nusselt number and friction factor for the plain tube are obtained and validated with equations of Dittus Boelter and Petukhov as given below;

$$Nu_{th} = 0.023\, Re^{0.8}\ Pr^{0.4} \quad (12)$$

$$f_{th} = (1.82 * \log_{10} Re - 1.64)^{-2} \quad (13)$$

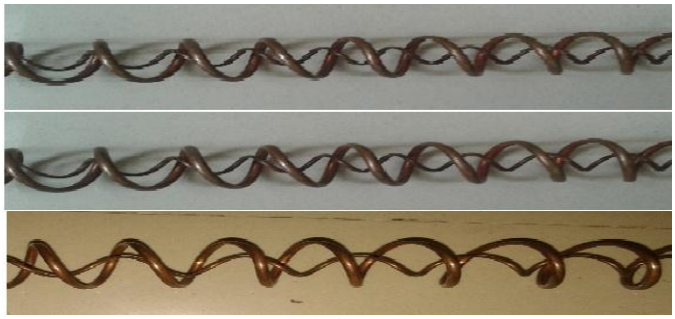

Figure 3: Actual view of double coil wire insert

The comparisons of Nusselt number and friction factor for the present plain tube with existing correlations are shown in Figs. 4 and 5, respectively.

These figures shows that validation experiments of heat transfer in terms of Nusselt number and friction factor for the plain tube are in good agreement with the results obtained from Dittus-Boelter and Petukhov equations. The results of present plain tube and previous equations are nearly the same. Thus, this accuracy provides reliable results for heat transfer and friction factor in a tube with twisted tape inserts in this present study.

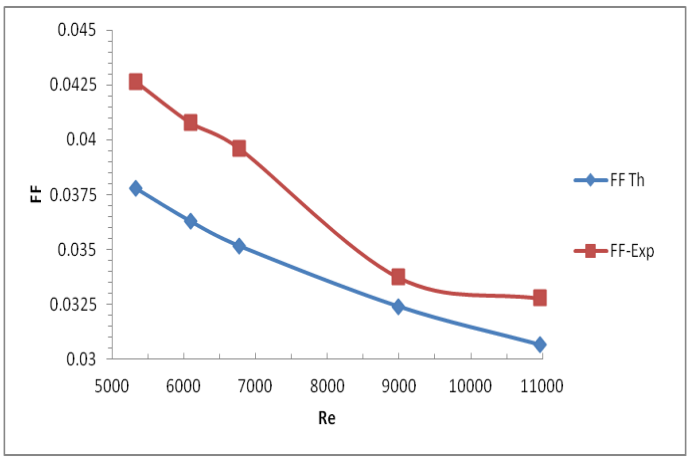

Figure no.4: Validation results for friction factor for plane tube

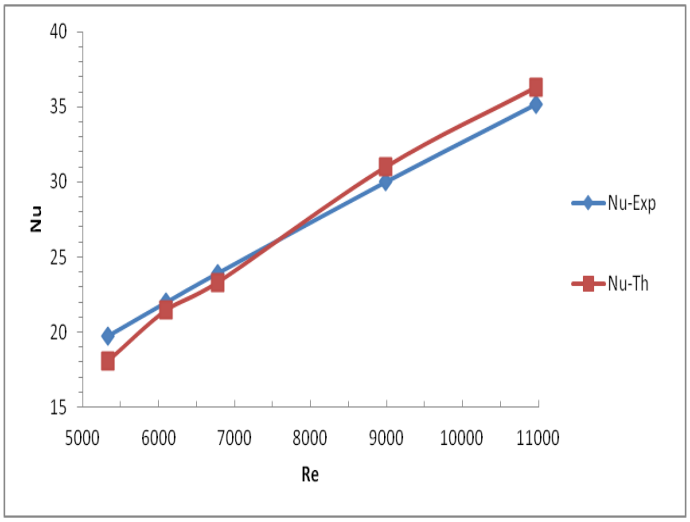

Figure no 5: Validation results for Nu for plane tube

The Reynolds number for validation test were ranged from 4000 to 11000 i.e. the range of Reynolds number used is for turbulent flow. Turbulent flow means Reynolds number greater than 4000. The results of the tests carried for performance checking of present wavy twisted tape are discussed further in results & discussion.

IV. RESULT & DISCUSSION

A. a Heat Transfer and Overall Enhancement Characteristics

The variation of Nusselt number with Reynolds number for various double coil wire inserts is shown in Figure 6. Highest Nusselt number was obtained for double coil wire with 18 mm pitch and 2.6 mm outer 1.5 inner coil wire diameter. From these graph it has been observed that the Nu increases 2.25 times than smooth tube for all the Copper double coil wire inserts. This is due to strong turbulence intensity generated by corrugations on inserts leading to rapid mixing of the flow causing heat transfer enhancement.

The variations of friction factor with Reynolds number for double coil wire inserts are presented in Figure 7. It is observed that the friction factor gradually reduced with rise in Reynolds number. It is observed to be maximum for insert having pitch 18 mm and 2.6 mm outer and 1.5 mm inner coil wire diameter. It is evident from Figures 6, 7 and 8 that when a double coil wire is inserted into a plain tube there is a significant improvement in Nusselt number

because of secondary flow, with greater enhancement being realized at lower Reynolds numbers.

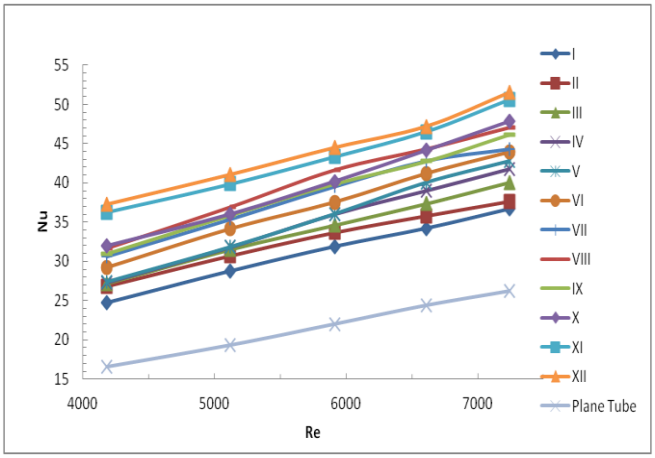

Figure no 6: Comparison of Nu for all Copper inserts

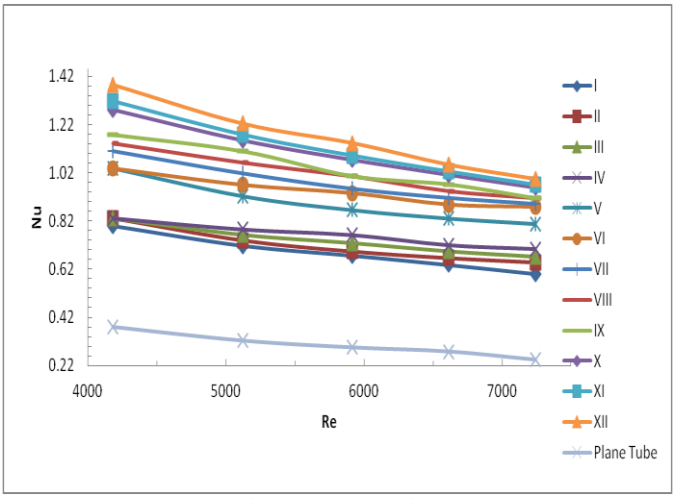

Figure no 7: Friction factor comparison for all copper inserts

This enhancement is mainly due to the centrifugal forces resulting from the spiral motion of the fluid and partly due to the wire coil acting as fin. It is observed that the increase in pitch and wire diameter causes increment in Nusselt numbers as well as rise in pressure drop.

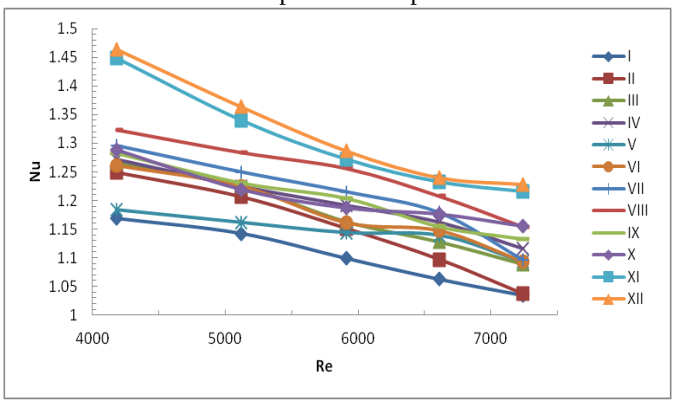

Figure no.8: Enhancement for all inserts

From Figure 6, the percentage rise in Nusselt numbers for double coil wire inserts compared to plain tube are about 33- 97%. The overall enhancement ratio is useful to evaluate the quality of heat transfer enhancement obtained over plain tube at constant pumping power. It is found to be more than unity for all the double coil wire inserts used.

Variations of overall enhancement ratio η against Reynolds number for various inserts are shown in figure 8. It is observed that overall enhancement tended to decrease gradually with the rise of Reynolds number for all pitch and wire diameter. The maximum value of overall enhancement is 1.46 for double coil wire insert having pitch of 18 mm with 2.6 mm outer and 1.5mm inner coil wire diameter. It is seen in Figure 8 that, for double coil wire inserts curves are of decreasing order for a given pitch in the range of Reynolds number from 4000 to 7242.

V.CONCLUSION

NOMENCLATURE

A_0 area of orifice, (m^2)
A test section inner tube area, ($\pi/4\ D^2$) (m^2)
Cp specific heat of air, (J/kg K)
Q_a air discharge through test section (m^3/sec)
D Inner diameter of test section, (m)
H pitch, (mm)
f_{th} friction factor(theoretical) for plain tube
f friction factor(experimental) for plain tube
f_i friction factor obtained using tape inserts
h experimental convective heat transfer coefficient, (W/m^2K)
h_w manometer level difference,(m)
h_{air} equivalent height of air column, (m)
k thermal conductivity, (W/mK)
L length of test section, (m)
$\dot{m}$ mass flow rate of air, (Kg/sec)
Nu_i Nusselt number (experimental) with tape inserts, (hD/k)
Nu Nusselt number (experimental) for plain tube
Nu_{th} Nusselt number for plain tube (theoretical)
Pr Prandtl number
p pitch, (m)
ΔP pressure drop across the test section, (Pa)
Q total heat transferred to air (W)
Re Reynolds number, ($\rho\ V\ D/\mu$)
T_1, T_8 - air temperature at inlet and outlet, (°K)
T_2, T_3, T_4, T_5,T_6,T_7 - tube wall temperatures, (°K)
Ts average Surface temperature of the working fluid, (°K)
T_b bulk temperature, (°K)
V air velocity through test section, (m/sec)

Greek symbols

ν Kinematic viscosity of air, (m^2/sec)
μ dynamic viscosity, (kg/m s)
η Over all enhancement
ρ_w density of water, (Kg/m^3)
ρ_a density of air (Kg/m^3)

ACKNOWLEDGMENT

I would like to express my deep sense of gratitude to Prof. S. A. Kale for his inspiring and valuable suggestions. I am indebted to him for helping to select my project topic and providing guidance throughout the completion of the Project.

I also thank Prof. Mutallik Desai for valuable advice and guidance. I am also thankful to the Head, Mechanical Engineering Department Prof.K.D.Sapate Trinity Collge of Engineering Pisoli, Pune.

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