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Research Article

Augmentation of Heat Transfer using Drilled Hole & Cut Twisted Tape Inserts

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Abstract

A heat exchanger is a mechanism facilitating the convective heat transmission of fluid within the conduit and is lengthily used in many applications, such as thermal power plants, chemical industries, air conditioning equipments, refrigerators, and radiators intended for vehicles. Till now efforts have been completed to decrease the dimension and price of heat exchangers. In broad, heat transport expansion can be separated in two types. One is the reactive technique without incentive by external command such as rough surfaces, extensive surfaces and swirl flow strategy (twisted tape, wire coils). The other is active technique, which needs additional outside power sources, for example, fluid shaking, fluid inoculation and suction and employing electrostatic fields. Turbulators are put into the stream to give redevelopment of the boundary layer, to add to the heat transfer plane region and to origin improvement of heat transfer by rising the turbulence or rapid amalgamation. Consequently, a added compact and cost-effective heat exchanger with short operation costs can be obtained. In current research work, a twisted tape insert having a hole at various locations & v-cuts along the outer periphery is experimentally analyzed & effects of various parameters on Nusselt number & Reynold's number is studied.

Keywords: *Augmentation, insert, Nusselt number, Reynold's number, twist ratio*

1. Introduction

Heat exchangers are extensively used in a variety of manufacturing processes for heating and cooling applications like air habituation and refrigeration structures, heat revival processes cooking and dairy processes, element procedure plants etc. The main confront in designing a heat exchanger is to build the tools compact and attain a high heat transfer pace using lowest amount pumping power. Methods for heat transmission increase are related to several engineering applications. In current years, the high rate of energy and matter has resulted in an amplified effort meant at producing extra efficient heat exchange utensils. Also, occasionally there is a need for trimness of a heat exchanger in particular applications, such as space appliance, during an expansion of heat transfer. For instance, a heat exchanger for an ocean thermal energy conversion (OTEC) place necessitates a heat transfer shell region of the sort of 10000m^2/MW. Consequently, a raise in the competence of the heat exchanger in the course of an augmentation practice may result in a significant economy in the material price. Furthermore, as a heat exchanger turn out to be older, the opposition to heat transfer augmentation owing to fouling or scaling. These harms are more ordinary for heat exchangers used in marine applications as well as in chemical industries. In number of specific applications, such as heat exchangers selling with fluids of little thermal conductivity (gases and oils) as well as desalination plants, there is a call for to boost the heat transfer pace. The heat transfer rate can be enhanced by introducing a disorder in the fluid stream thus breaking the viscous and thermal boundary film. Still, in the course pumping power may augment significantly and eventually the pumping price becomes elevated.

Therefore, to reach a desired heat transfer rate in a present heat exchanger at a profitable pumping power, a number of techniques have been projected in current years and are discussed in the classification part. For over a century, efforts have been made to create more competent heat exchangers by employing a variety of techniques of heat transfer improvement. The learning of enhanced heat transfer has added severe momentum through current years, however, due to increased burden by industry for heat exchange equipment that is less costly to manufacture and run than typical heat exchange devices. Savings in materials and power use also supply strong motivation for the progress of better methods of improvement. When designing cooling structure for vehicles and spacecraft, it is very important that the heat exchangers are particularly dense and trivial. Also,

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improvement devices are essential for the elevated heat duty exchangers created in power generating utilities (i.e. air-cooled condensers, nuclear fuel rods). These functions, as well as frequent others, have led to the progress of a range of improved heat transfer shells. To compare the results of twisted tape having hole and v cut insert with plane twisted tape insert to observe increment or decrement in parameters like Nusselt number, friction factor, enhancement ratio. Hsieh and Huang carried out investigational and arithmetical study of the forced convective heat transport in plane channels by means of rectangular cross section (Hsieh and Huang, 2003). Sarma, et al focused on the aim of the appropriate ribs used for enhancing heat transfer in a rectangular channel heat exchanger by employing wall heat transfer (Nusselt number), friction loss (friction factor) as well as thermal recital (thermal enhancement factor) information (Sarma, 2008). Hsieh checked the effects of inlet temperature and rib elevation on the fluid flow as well as heat transfer recital of the corrugated conduit within the elevated temperature heat exchanger (Hsieh, 2003). The inlet temperature varied from 850 K to 1265 K and rib height to channel altitude ratio was varied from 0.083 to 0.333. Pairson Naphon carried out liquid crystal found investigational study of the heat transfer improvement abounding by ribs on a vertical plate (Pairson Naphon, 2006). Since the ribs were adiabatic, they did not act as extensive heat transfer surfaces except by redirecting the flow, were used to improve the heat transport. Smith Eiamsa-ard tested repeated ribs on heat exchanger surfaces to encourage disorder and improve convective heat transfer. study of heat transfer from a rectangular conduit (ratio of width to height as equal to five) having one surface heated at unvarying heat flux and roughened by frequent ribs (Smith Eiamsa-ard *et al*., 2007).

2. Present Work

The sample design of insert used in the experimentation is shown in the figure given below.

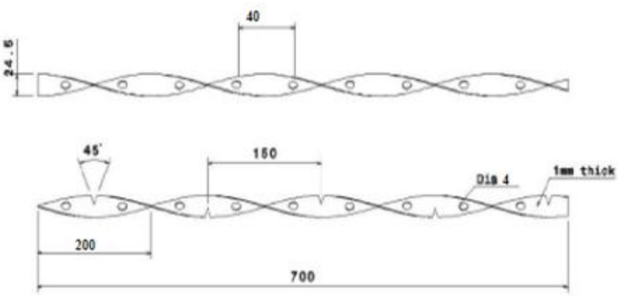

Fig.1 Drilled & Cut Twisted Tape Inserts

The insert shown in the above drawing consist of pitch of 200mm.They consist of hole of 4 mm with length between two horizontal hole as 40mm. The distance between two v-cut is 150mm.The overall length of the insert is 700mm and thickness of 1mm. The trial readings on passive heat transfer amplification using warped tape inserts was passed in a single phase flow heat exchanger having the specifications as shown under.

Fig.2 Experimental Setup

3. Mathematical modeling carried out on insert

The heat flux applied to the test tube causes an increase in the outer surface temperature T_{out} of the test pipe in axial route. Consequently, the heat loss is designed for each division of the test pipe in which the thermocouples exist. The heat loss q_{loss} is the heat which gets transferred from the outer tube wall to the surroundings.

$$q_{air} = q_{conv} \quad (1)$$

$$q_{air} = m\,Cp_{air}\,(To-Ti) = \Delta VI - q_{loss} \quad (2)$$

where,
To and Ti are the temperatures at outer and inner wall of test pipe.
V= Voltage supplied by the heater.
I = The current supplied to the test pipe.
The heat supplied by the electrical cable in the test pipe is merely about 3 to 4% higher than the heat absorbed by the air for the thermal equilibrium test due to the convection as well as radiation heat losses (Q_{loss}) from the test section to the surroundings. Therefore, simply the heat transport rate engrossed by the air is taken into contemplation for the convective heat transfer coefficient calculation.

$$q = qair/(\pi\ Do\ L) \quad (3)$$

The procedure for calculating the various parameters for with and without inserts is given below. The experimentation was done at constant heat supply of 87V; hence the calculations are done at constant heat supply. A sample observation table is shown below to understand the parameters need to be observed during experimentation.

Avg. Surface Temp is given as

$$T_s = \frac{(T_2+T_3+T_4+T_5+T_6+T_7)}{6} \quad (4)$$

Avg. Temp of air,

$$T_b = \frac{T_1+T_8}{2} \quad (5)$$

From table of properties of air the parameters like density of air (ρa), Kinematic viscosity (ν), thermal conductivity (k), specific heat of air (Cp), Prandtl number (Pr) at 1atm pressure.
Manometer difference = water head = hw
Air head,

$$h_a = h_w \times \frac{\rho_w}{\rho_a} \quad (6)$$

where,

ρ_w = Density of water = 1000 kg/m^3
Air volume flow rate,

$$Q_a = C_d \times A_o \times \sqrt{(2 \times g \times h_{air})} \quad (7)$$

where,

A_0= cross sectional area of orifice.
Mass flow rate is given as,

$$\dot{m} = Q_a \times \rho_a \quad (8)$$

Velocity of air is given as,

$$V = \frac{Q_a}{A} \quad (9)$$

where,
A= cross sectional area of pipe.
Heat carried out is given as,

$$q = \dot{m} \times C_p \times (T_8 - T_1) \quad (10)$$

$$h = \frac{Q}{A\,(T_S - T_b)} \quad (11)$$

where,
h = heat transfer coefficient.
Ts = surface temperature
The Reynolds number for the fluid is defined by,

$$Re = \frac{VD}{\upsilon} \quad (12)$$

where,
V= velocity of the fluid.
ν = Kinematic viscosity of the fluid.

For internal flows if Reynolds (Re) number exceeds by 4000 then the flow is turbulent. After the flow is decided i.e. laminar or turbulent subsequently the Nusselt number can be considered. The hypothetical Nusselt number is considered under exclusive of considering the friction which is theoretical Nusselt number and next considered by taking into account friction which will be experimental Nusselt number.

$$N_{uth} = 0.023 \times R_e^{\,0.8} \times P_r^{\,0.4} \quad (13)$$

This equation is called Dittus-Boettier equation.

$$fs = (1.82 \log_{10} Re^{-1.64})^{-2} \quad (14)$$

This equation is used to come across friction factor named as Petukhov equation for flat surface.

where,
fs= Friction factor for smooth tube .
Re= Reynolds number.

The actual pressure drop & friction factor is calculated with the help of tappings at both the ends of test conduit associated to U-tube manometer and the friction factor is designed from the formula given below:

$$f = \frac{\Delta p}{\frac{L}{D} \times \rho V^2} \quad (15)$$

The overall augmentation competence is articulated as the ratio of the Nusselt number of an enhanced tube with wavy twisted tape insert to that of a even tube, at a steady pumping power.

$$PEC = \eta = \frac{Nu_{\,with}/Nu_{\,w/o}}{(f_{with}|f_{w/o})^{1/3}} \quad (16)$$

4. Results and discussions

The parameters mentioned in the above objectives are achieved by carrying out the experimentation and then making the comparisons of them for varying twist ratios and varying other parameters in the graphs drawn below.

The Nusselt number, friction factor are the significant constraints which decides the achievement of any experimentation job as both parameters are conflicting to each other. The Nusselt number illustrates the percentage rise in heat transfer improvement when inserts are positioned inside a test tube due to boost in heat transfer coefficient through comparison of it, without inserts. Contradictory to it is when inserts are placed inside the test pipe the friction gets produced inside the test pipe due to which there is drop in pressure hence the desire increase in heat transfer coefficient is offset by pressure drop. Hence, the inserts should be designed in such a way that there pumping cost should get offset by heat enhancement.

The Nusselt number and the friction factor are attained for a smooth pipe to authenticate the untried method used previous to the tests with twisted wire inserts. The consequences of Nusselt number and friction factor for smooth pipe are evaluated in the company of the outcomes achieved from the well-known steady state flow correlations of Dittus Boelter as well as Petukhov, for the completely developed disordered flow in circular pipes. The figures shown below illustrate the assessment among the outcomes of the current smooth pipe and the correlations of Dittus Boelter and Petukhov, respectively.

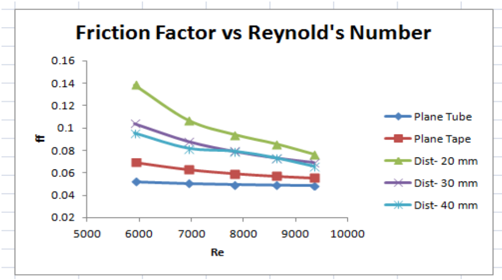

Fig.3 Friction factor Vs Reynolds number

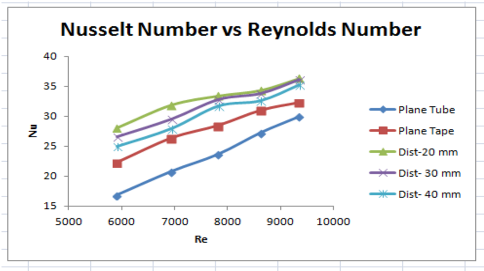

Fig.4 Nusselt Number Vs Reynolds number

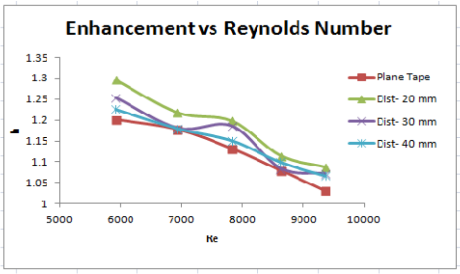

Fig.5 Enhancement Vs Reynolds number

The graphs shown above are for the insert with varying vertical distance. It is observed that the Reynolds number increases with the decrease in friction, so at lower Reynolds number the friction factor is high however as the Reynolds number increases the friction factor drops as compared to the initial value. The Nusselt number raises with the rise in Reynolds number. The heat transfer enhancement takes place at lower Reynolds number. The 20 mm distance has higher heat transfer enhancement as compared to 30 mm and 40mm distance. Also it is greater than plane tube It is also observed that Reynolds number from 7000 to 8000 have higher heat transfer enhancement.

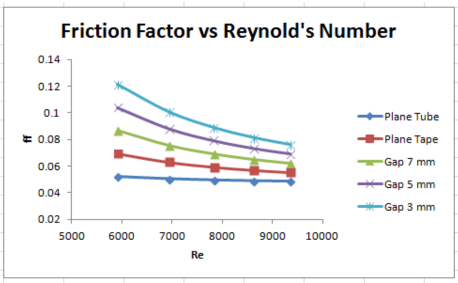

Fig.6 Friction factor Vs Reynolds number

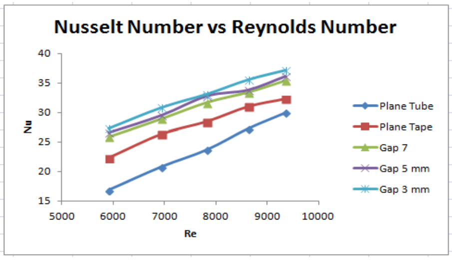

Fig.7 Nusselt Number Vs Reynolds number

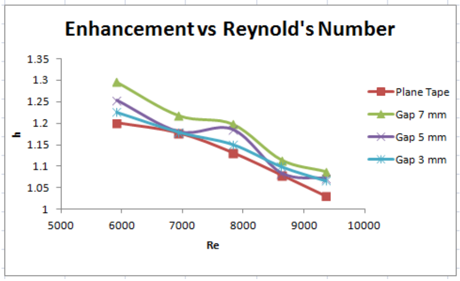

Fig.8 Nusselt Number Vs Reynolds number

The graphs are shown above for the insert with varying gap. It is seen that as the Reynolds number rises there is a decrease in friction, so at lower Reynolds number the friction factor is high however as the Reynolds number increases the friction factor drops as compared to the initial value. The Nusselt number augments with the rise in Reynolds number. Also heat transfer enhancement is more at lower Reynolds number.

The gap 7 mm distance has higher heat transfer enhancement as compared to gap 5mm and 3mm. Also it is greater than plane tube. It is also observed that Reynolds number from 7000 to 8000 have higher heat transfer enhancement.

Conclusions

The readings present an investigational examination of the possible use of longitudinally drilled twisted tape turbulators to enhance the rate of heat transfer in a horizontal round pipe having inner diameter 26 mm and air as operational fluid. The Reynolds number ranged between 41500 to 9550. The effects of parameters such as modified wave-width, Reynolds number taking place on heat transport as well as overall augmentation ratio are considered.

The subsequent conclusions can be suggested:

1) The enhancement of heat transfer with longitudinally drilled twisted tape turbulators as compared to plain tube varied from 6 to 29% for gaps of 3 mm to 7 mm. This enhancement is mostly due to the centrifugal forces ensuing from the spiral movement of the fluid.
2) Reduction in tape width causes rise in Nusselt numbers as well as friction factors. The greatest

friction factor increase seen was around 57% for 3mm compared to plain tube.

3) The overall enhancement for the tubes with longitudinally drilled twisted tape turbulators is 1.29 for gap of 7 mm.
4) Thus the enhanced performance can be achieved using longitudinally drilled twisted tape turbulators as matched up with the plane twisted tape. Thus, considering the enhanced heat transfer and savings in pumping power longitudinally drilled twisted tape turbulators are seen to be attractive for improving disordered flow heat flow in a horizontal round pipe.
5) Amongst the two type of inserts containing v cut the insert with 8mm v cut had efficiency greater than the insert having v cut of 4 mm
6) It is found that the insert having v cut has less heat transfer enhancement as compared to plane tape insert with hole.

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