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Forced Convection Heat Transfer Enhancement Using Non-Metallic Bipartition Perforated Insert

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Abstract

Enhanced heat transfer has been studied for decades through the application of active, passive, and compound heat enhancement approaches. The current study employs a passive heat transfer enhancement technique using non-metallic flow divider type inserts with air as the working fluid. Whenever inserts are utilized to enhance heat transfer, the pressure drop rises in line with the rate of heat transfer. The rise in pressure drop results in higher pumping expenses. Therefore, in heat enhancement approaches, the advantages of greater pressure drop and heat transfer rate should be balanced. Plain tubes were used for the early experiments in order to verify the arrangement. Its results Experimental experiments were carried out with flow dividers with non-metallic, porous inserts, under different heat input and Reynolds numbers. It's a Nusselt number. At 100V and 150V, respectively, the Nusselt number increases by 36% and 65% relative to a simple tube. For different Reynolds values, OEE (Overall Enhancement Efficiency) was more than one.

Keywords: Forced Convection, Nusselt Number, Reynolds Number, Flow Divider Perforated Insert, Heat Augmentation

Introduction

Heat transfer procedures transform, move, and use heat energy for home, commercial, and automobile purposes. The thermal efficiency of heat exchange equipment can be significantly increased by improving heat transfer in a wide range of applications. Heat enhancement methods are used to increase the fluid's turbulence, which raises the convective heat transfer coefficient. Because of the greater pressure drop caused by this technology, pumping becomes more expensive. Therefore, the enhancement approach should be used to maximize the rate of heat transfer and pressure drop. Heat transfer improvement strategies can be categorized into three general categories:

1 Active techniques,
2 Passive techniques and
3 Compound techniques.

Active approaches have not showed much potential because of their complex architecture, which improves heat transfer by requiring some external power input. In contrast, passive techniques depend on the fluid's inherent energy to supply the extra power required to improve heat transmission, which eventually lowers fluid pressure. Hybrid approaches that integrate both passive and active strategies are called compound methods. Bargles [1] has provided extensive information regarding the active and passive techniques for heating laminar flow.

Although they are effective, their prospective applications are limited by their complex designs [1-3]. The heat augmentation can be further boosted by shaping and arranging the twisted tape's perforations in different ways [3]. Among these techniques is twisted tape [3, 6]. When alternate clockwise and counterclockwise twisted tapes with varying twist ratios are employed, the Nusselt number grows over a range of Reynolds numbers with uniform flux circumstances. It leads to superior chaotic mixing [6]. Square-cut circular rings [4], coiled tubes [5], rough surfaces [8], fluid additives [8], and extended surfaces [10] have all been used to successfully increase heat transmission.

In the first approach, heated fluid is moved from the boundary surface to the core flow by use of a fluid helical flow created by an insert that works as a turbulence enhancer. By expanding the heat transfer surface area and redeveloping the boundary layer, the second strategy enhances the system's thermal

Nomenclature:

A = Surface area of the test section, m^2

C_p = Specific heat of air at constant pressure, J/kgKh = Heat transfer coefficient, W/m^2 k

Nu = Nusselt number, (dimensionless number)

f = Friction factor, (dimensionless number)

d = Smooth tube inner diameter, ml = Length of test section, m

p = Turbulator pitch, m

ṁ = Mass flow rate, kg /s

u = Fluid velocity in x-direction, m/ s

Q = Heat transfer rate, W

Re = Reynolds number, (dimensionless number)

TPF= Thermal performance factor

PEC = Performance evaluation criteria

T_i = Inlet temperature, K

T_o = Outlet temperature, K

T_s = Surface wall Temperature, K

T_b = Mean bulk temperature, K

P=Pressure, Pa

performance.

Experimental Set-up:

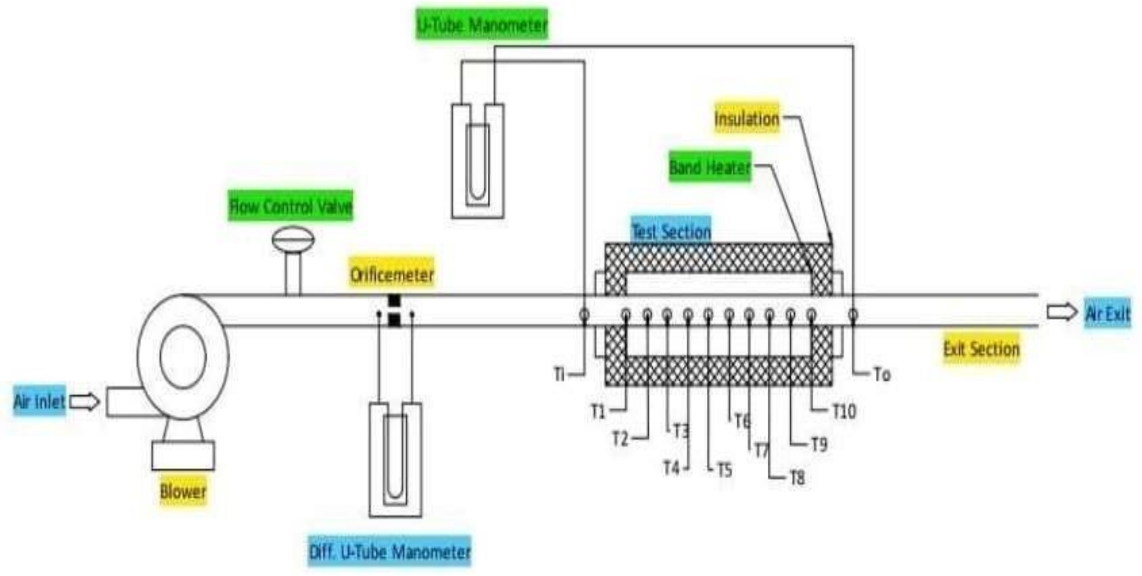


Fig. Schematic Experimental Setup of Project

The suggested experimental setup is shown in the above figure. The test segment will be built out of G.I. pipes that are roughly one meter long. The necessary inner and outer diameters of the test section will depend on the range of Reynolds numbers to be examined in the turbulent zone. A centrifugal blower will be utilized in order to generate the air flow in a configuration with various mass flow rates. An electric heater will be installed on the test section in order to heat it at different constant heat flux wall boundary conditions. In order to keep the test section's heat flux wall condition constant, the dimmer-stat will regulate the input electric power. To measure the fluid flow, a calibrated orifice meter will be employed.

The test section's outer wall will be insulated to reduce heat loss to the surrounding area. The temperature of the test section's wall surfaces and the air passing through them will be measured using calibrated thermocouples on the surface and at the entrance and outflow of the test section. The pressure drop brought on by friction as a fluid moves across the test portion will be measured using a calibrated pressure measuring equipment.

Every temperature will be measured with a temperature indication gadget. The reading will be taken once a steady condition has been reached. The experiment will investigate the properties of pressure drop and heat transfer using different types of bipartition inserts. Passive inserts of various bipartition types can be created by varying insert characteristics such as twist angle and ratio. The mass flow rate, the heater's heat input, and other input parameters will be changed in order to conduct the insert experiments.

Procedure to carry out the work:

1. A research problem statement will be created and the research gap for the forced convection heat transfer enhancement employing passive inserts will be determined.
2. The experimental design to investigate forced convection heat transfer will use a horizontal plain tube with and without bipartition inserts.
3. The Reynolds number of the turbulent flow and the heat input from the heater will be used as input

variable parameters. The parameters under control will be unique.

4. A comprehensive steady state experiment on a plain tube will be carried out to investigate the properties of heat transmission and fluid flow.
5. The typical correlations for internal flow—Nusselt's number and friction factor—found in the literature will be utilized to confirm the plain horizontal tube experiment's outcome.
6. Modifying insert parameters such as pitch to diameter ratio and twist angle will be necessary during the production of the unique non-metallic bipartition type passive inserts.
7. A comprehensive, steady-state experiment with different input parameters will be conducted on these inserts to evaluate the properties of heat transmission and fluid flow.
8. In order to validate the experimental results for bipartition inserts in a horizontal tube, modelling software will be utilized for a numerical investigation.
9. A statistical analysis will be conducted to compare the fluid flow characteristics and heat transfer enhancement for a horizontal plain tube with and without a bipartition insert.
10. The features of fluid flow and heat transfer between two horizontal plain tubes—one with a bipartition insert and the other without—will be analysed.
11. The bipartition insert of a horizontal plain tube with a high overall enhancement ratio will be identified.

Sample calculation

(PLAIN TUBE, V=100, H=0.045m)

Properties of air, Cp, µ, k and Pr are calculated at bulk mean temperature i.e. at Tmb..

1) Tbm = (Ti + To) / 2
= (34.6 + 55.5)/2
= 45.05 °C

Ts = (T1 + T2 + T3 + T4 + T5 + T6 + T7 + T8 + T9) /9,
= (96.3+88.5+86.6+87.2+89.6+87.7+86.1+85.8+90.1+34.6+55.5+32.1) /9
=8 8 . 6 5 oC

3) d1 = 0.026m, d0 = 0.013m.

eas

y head difference.

Volumetric discharge (Q)= (a1*a0*sqr. root of (2gH[(rhow/rhoa) – 1])) / (a1^2 – a0^2)

a1= (pi/4)*(0.026)^2 = 5.30929*10^-4
a2= (pi/4)*(0.0125)^2 = 1.3273*10^-4.

(5.30929*10^-4*1.327310^-4* sqr. root of (2*9.81*0.026[(1000/1.128) – 1]))
Q = ____
sq.root of [(5.30929*10^-4)^2 - (1.3273*10^-4)^2]

Q = 0.003810 (m^3/s).

Mass flow rate (m) = rho * Q
= 1.128 * 0.003810

= 0.004173 Kg/sec.

Heat transfer rate (q) = mCp(T0-Ti)
= 0.004173 * 1007.2 * 20.9
= 87.85 W/m^2.

Heat transfer coefficient (h) = q / (As*[Ts-Tbm])
= 87.85/(pi*0.026*1*[88.65 – 45.05])
= 24.67 W/(m^2*k).

7) Nusset number (Nu) = hd1/k
= 24.67* 0.026 / 0.026618
= 24.09

Reynold's Number=(Q*d)/a1*neo

=(0.003810*0.026)/ 5.30929*10^-4*0.000016258

= 11478.48

Result Table and Graphs:

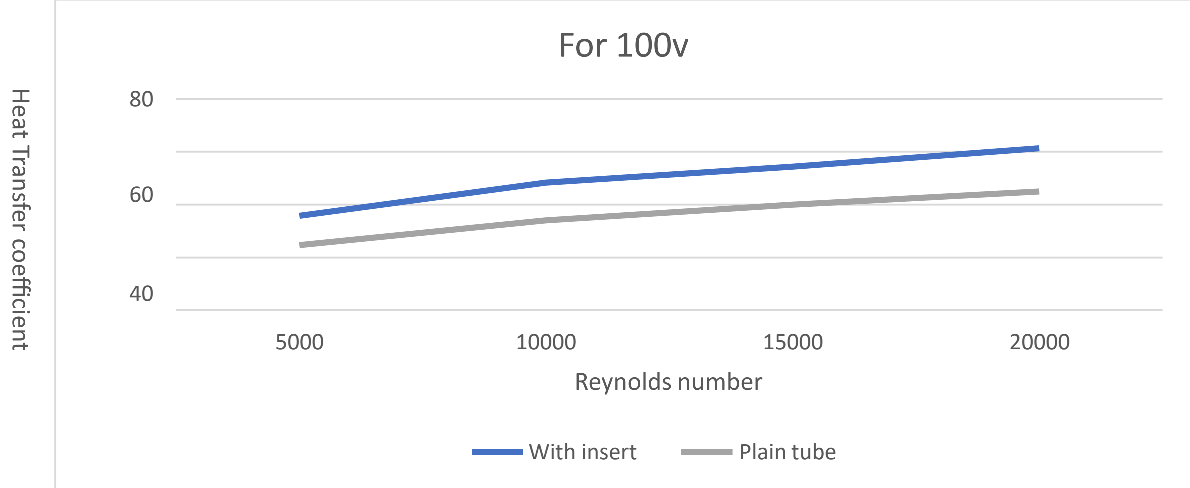
Table No.1 : Plain tube with perforated Insert

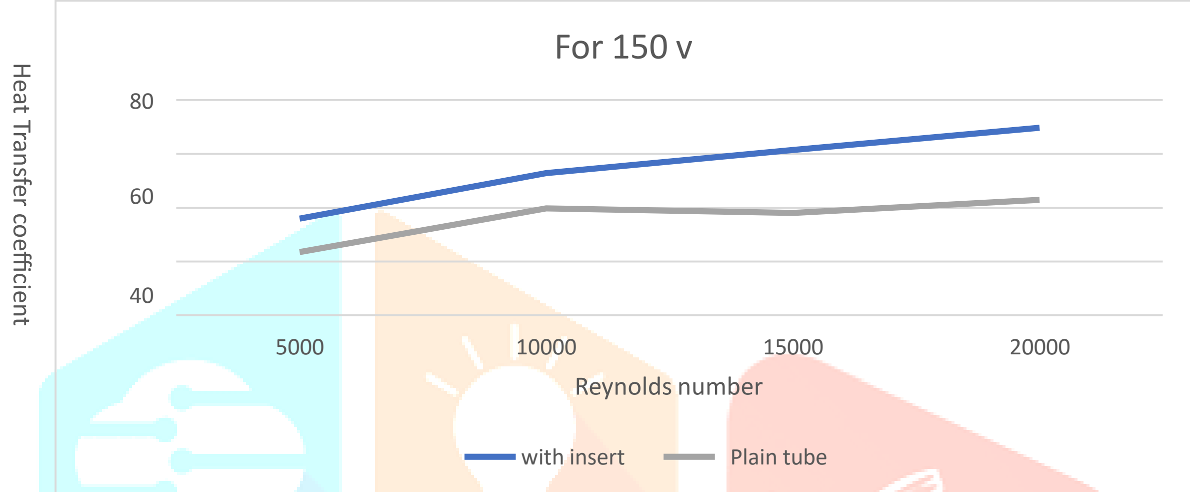
Volt	Head diff.(x)	Ts	dt	Tbm	m'	Q	H	Nu	%h	%Nu
100	0.045	73.06	20.5	43.65	0.0041	0.0038	41.21	34.98	45.01	31.04
100	0.09	64.65	15.5	41.25	0.0059	0.0053	81.59	47.31	41.88	29.52
100	0.135	61.72	13.2	40.0	0.0072	0.0065	122.12	53.15	36.15	26.56
100	0.18	60.28	12.2	39.7	0.0084	0.0075	162.26	59.96	36.27	26.61
150	0.045	127.45	49.0	58.9	0.0040	0.0038	42.78	35.21	52.77	34.53
150	0.09	104.1	38.5	52.15	0.0058	0.0054	85.08	51.78	33.14	24.89
150	0.135	96.24	32.3	49.8	0.0071	0.0066	125.10	60.11	61.25	37.98

150	0.18	93.61	30.6	48.8	0.0082	0.0076	166.56	68.20	62.38	38.41

Table No. 2 : Plain tube without Insert

olt	Head diff.(x)	Ts	dt	Tbm	ı'	Q	H	Nu	%h
100	0.045	88.65	20.9	45.05	0.0038	0.0041	87.85	24.69	24.12
100	0.09	81.72	17.9	43.45	0.0053	0.0059	106.66	34.14	33.34
100	0.135	72.77	14.3	40.65	0.0065	0.0072	104.81	39.97	39.03
100	0.18	69.48	12.8	40	0.0075	0.0084	108.43	45.05	44.00
150	0.045	166.24	50.5	58.75	0.0039	0.0040	207.05	23.60	23.05
150	0.09	142.17	39.8	53.5	0.0054	0.0058	235.10	39.82	38.89
150	0.135	131.4556	34.6	51.3	0.0066	0.0071	249.76	38.17	37.28
150	0.18	119.6556	29.7	48.95	0.0076	0.0083	248.22	43.00	42.00





Result:

1. The heat transfer coefficient with the insert is larger than the value without the insert, as the graph demonstrates.

2. It has been found that the maximum heat transfer coefficient with insert is higher than the heat transfer coefficient without insert by 36% at 100 volts and 65% at 150 volts.

Conclusion:

Following overall conclusions have been drawn from this project work:

1) The Nusselt number and Heat Transfer coefficient for a flow divider type insert in a plain tube are 65% greater for a 150 volt input than they are without an insert.
2) The Heat Transfer coefficient and Nusselt number of a plain tube with a flow divider type insert and a 100 volt input are 36% greater than they would be in the absence of the insert.
3) As flow velocity increases, so does the heat transfer coefficient.
4) This project effort creates a novel flow divider kind of insert to enhance forced convection-related heat transfer.
5) The majority of researchers used metallic inserts; non-metallic insert project work is quite rare.
6) The flow friction and heat transfer characteristics of a circular tube equipped with perforated Inserts with dual porosities were investigated in an experimental investigation. It has been found that the perforated tape inserts greatly increased the heat transfer rate with a matching rise in friction factor when compared to a plain tube.

7) The picture above illustrates that passive tube inserts work well in laminar flow and less well in turbulent flow.

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