"EXPERIMENTAL INVESTIGATION TO STUDY HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS OF HEMISPHERICAL CUP SHAPE PASSIVE INSERT FOR A FULLY DEVELOPED TURBULENT FLOW IN DOUBLE PIPE HEAT EXCHANGER"

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Abstract: In present experimental investigation, heat transfer and pressure drop characteristics of a new passive cup insert for three different pitch sizes are reported for fully developed turbulent flow. Experiments were performed in a tube in tube heat exchanger using water as a working medium in the Reynolds number range of 8000 to 32000. The average Nusselt number ratios and average friction factor ratios for augmented tube case to plain tube case are found to be in the range of 1.67-1.4 and 5.44 - 4.93 respectively. The average performance ratio for constant pumping power are also reported and found to be in the range of 0.69 – 0.6. The effect of pitch of cup on performance is reported.

***IndexTerms* – Average performance ratio, turbulent flow regime, passive insert**

Abbreviations [1]

A	Area
D	Diameter
ΔP	Pressure drop of fluid (N/m^2)
T_{lm}	Logarithimic mean temperature difference
h	Heat transfer coefficient (W/m^{2}K)
k	Thermal conductivity (W/mK)
l	Length of insert
L	Length of test section for heat transfer
L_1	Length of tube between pressure taps
m	Mass flow rate of fluid (kg/sec)
Q	Heat transfer rate (kW)
T	Temperature (K)
V	Mean fluid velocity (m/sec)
C_{pw}	Specific heat of water at constant pressure
ρ	Density of fluid (kg/m3)
µ	Dynamic viscosity (kg/m-sec)

Subscripts

a	Augmented tube case
avg	Average
c	Cold
h	Hot
i	Inner
in	Inlet
o	Outer
out	Outlet
p	Plain tube case
s	Tube wall surface

Dimensionless parameters

Re	Reynolds number
R3	Nusselt number ratio at equal pumping power
Pr	Prandtl number
f	Average friction factor

Nu Average Nusselt number

I. INTRODUCTION

Heat exchangers are frequently used in many engineering applications. Over the past decades, many engineering techniques have been developed for heat transfer enhancement in heat exchangers. One of these techniques is the use of tabulator elements to increase the heat transfer coefficient from the flow surface through an increase in turbulent motion of fluid. It is found that this technique can reduce the sizes of the heat exchangers and save up the energy and hence this technology has been widely applied to many heat exchanger applications like refrigeration, automotive, process industry, solar water heater, etc.

In general, enhancing the heat transfer can be divided into three groups, active method, passive method and compound method [2].Active method involves some external power input for the enhancement of heat transfer. Some examples of active methods include surface vibration, fluid vibration, electro- static fields, mechanical aids, suction or injection and jet impingements requiring external mechanical power to achieve heat transfer enhancement.

For the convective heat transfer, the key ways to enhance heat transfer rate are increasing the effective surface area and residence time of the heat transfer fluids. In passive methods of heat transfer, the above two key principles are followed through application of several techniques to generate the swirl in the bulk of the fluids and disturb the actual boundary layer resulting in increased effective surface area, residence time and consequently heat transfer coefficient. There are hundreds of passive methods for heat transfer enhancement out of which treated surfaces, rough surfaces, extended surfaces, displaced enhancement devices, swirl flow devices, coiled tubes, surface tension devices and additives for liquids and gases are popular in different aspects. In compound method of heat transfer augmentation, techniques of above two methods or more than one techniques of single method are combined for enhancing heat transfer such as rough surface with a twisted tapes or rough surface with fluid vibration, coiled inserts with twisted tapes etc.

Many researchers have tested different geometries of passive inserts like helical tapes, coiled wires, swirl generators, conical rings, rib inserts etc. Some of the works are reported here.

H.Bas and V.Ozceyhan [3] have experimentally studied the heat transfer enhancement in a twisted tape inserted tube by putting the twisted rods separately from the tube wall. They reported the highest heat transfer enhancement ratio as 1.756 for clearance ratio of 0.0178, as 1.744 for clearance ratio of 0.0357 and as 1.789 for clearance ratio of 0 at twist ratio of 2 for a twist ratio range of 2 to 4. They concluded that the twist ratio has major effect compared to clearance ratio on heat transfer for twisted tape inserts. Kumar and Prasad [4] experimentally investigated the effect of twisted tape inserts on heat transfer and pressure drop in the Reynolds number range of 4000 to 21000 with two twist ratios of 3 and 12. They reported increments in heat transfer and pressure drop with decreasing values of twist-pitch to tube diameter ratio.

Yilmaz et al. [5] performed experiments to see the influence of swirl generators with conical and spherical deflecting elements on heat transfer in the Reynolds number range of 32000 to 110000 using air as a working fluid and observed that deflecting elements for swirl generators are not advantageous over plain swirl generators. Anvari et al. [6] based on their experimental investigation performed with conical ring turbulators concluded that heat transfer enhancement is significantly affected by these devices and especially diverging rings are more effective. The pressure drop is also significantly increased. Thus these devices are more suitable where compact size is more significant compared to required pumping power for the flow. Saha [7] tested full and short length twisted tapes with and without oblique teeth for rectangular and square ducts with axial corrugations for turbulent flow with Reynolds number ranging from 10000 to 100000 using air as a working fluid. He observed a minute improvement in thermo hydraulic performance for twisted tapes with oblique teeth compared to twisted tape without oblique teeth.

It is seen that from literature that cup shaped geometries of inserts is not reported. Cup shaped geometries are expected to create a waviness to the flow which may result in disturbing the boundary layer. Hence, in this experimental study, attention is focused on use of a proposed cup shape insert with threaded core rod as a passive heat transfer augmentation device for circular pipe in turbulent flow regime.

II. EXPERIMENTAL SET UP AND METHODOLOGY

2.1. Details of insert:

For cup shape inserts, hemispherical cups with central hole are prepared separately and mounted on a threaded rod with locking nut so as to alter the distance between the cups to change the pitch as per the requirement. For these inserts, the threaded core rods are manufactured in three different segments with each segment length equal to 1/3rd of total test section length. The geometric details are listed in table 1. The actual pictures of inserts are also shown in fig.1.

Table 1. Details of cup shape with threaded core rod inserts

Insert	Material	Cup outer diameter	Cup thickness	Core rod diameter	Pitch
Cup 1	Aluminium	17.9 mm	3 mm	5.8 mm	0.15 m
Cup 2	Aluminium	17.9 mm	3 mm	5.8 mm	0.2 m
Cup 3	Aluminium	17.9 mm	3 mm	5.8 mm	0.25 m

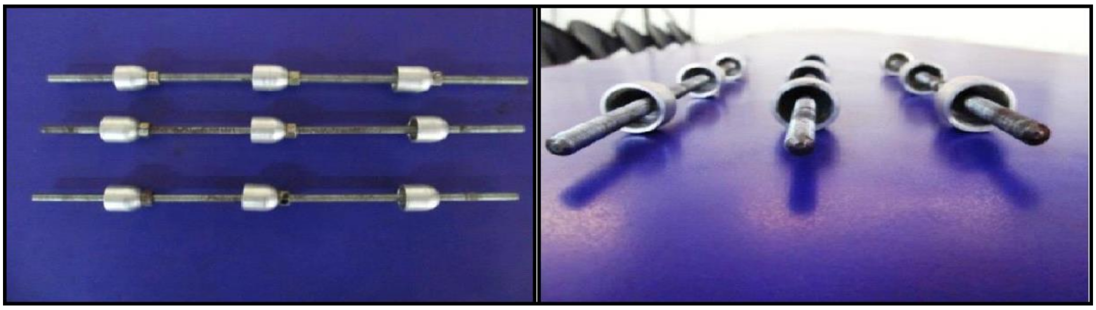

Fig.1. Pictures of cup shape inserts

2.2. Experimental set up:

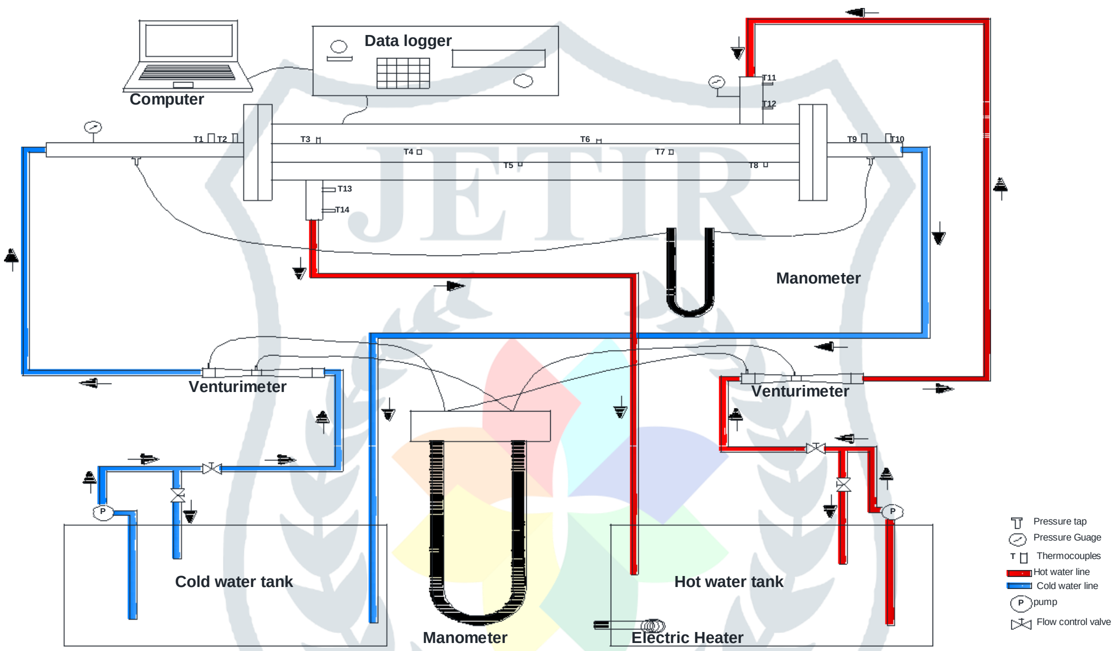

Fig. 2 Experimental set up

Fig.2 shows the details of experimental set up. For performing trials, a double pipe heat exchanger of test section length = 0.8m was used. Inner tube was made of copper with ID =21.4 and OD = 28.7 mm and outer tube was made of steel with ID = 48.3 mm and OD = 60mm. For storage of hot water, a 100 lit tank was used. For heating of water, two heaters of 2 kW capacities each were used. The temperature of hot water was maintained to 75 $^{\circ}$C and then hot water was circulated in a closed loop. A gate valve was used to maintain the required flow. Venturimeter was used for measurement of flow rate of water.

Thereafter the cold water was flowed through the inner copper tube using circulation pump. One bye- pass valve and two gate valves were used for maintaining flow rate of cold water. A U tube manometer was with Carbon Tetra-chloride (CCL_4) as a manometric fluid was used to measure pressure drop of test section. After reaching the steady state pressure drop and temperature readings were taken for a plain tube in the Reynolds number range of 8000 to 32000. The experiment was repeated for test section (copper tube) fitted with cup shape inserts.

T type thermocouples were used for temperature measurement and reading was recorded in a data logger. For accuracy in measurement of temperatures, two thermocouples were placed at inlet and exit. Copper tube surface wall temperature was measured by six thermocouples and average was taken for data analysis.

III. DATA COLLECTION AND ANALYSIS [1]

The heat gained by cold water in copper tube (Q_c) is calculated as follows:

$$Q_c = \dot{m}_c C_{p,w}(T_{c,out} - T_{c,in}) \tag{1}$$

The heat rejected by hot water in steel tube (Q_h) is calculated as follows:

$$Q_h = \dot{m}_h C_{p,w}(T_{h,in} - T_{h,out}) \tag{2}$$

At thermal equilibrium conditions, the difference between rejected and heat gained by hot and cold water was found in the range of 2 to 9% due to various heat losses from test section. Hence average value of heat transfer rate is considered for heat transfer analysis.

$$Q_{avg} = \frac{Q_c + Q_h}{2} \tag{3}$$

Considering a constant wall surface temperature ($\tilde{T}_s$) and no thermal resistance in copper tube test section, the average inner side heat transfer coefficient (h_i) is obtained using [8]

$$Q_{avg} = h_i A_i \Delta T_{lm} \tag{4}$$

Where,

$$\Delta T_{lm} = \frac{(\tilde{T}_s - T_{c,out}) - (\tilde{T}_s - T_{c,in})}{ln[(\tilde{T}_s - T_{c,out})/(\tilde{T}_s - T_{c,in})]} \tag{5}$$

And

$$A_i = \pi D_{i,} L \tag{6}$$

The average of temperatures (T_s) recorded by six thermocouples mounted on copper tube outer surface is considered as tube wall surface temperature ($\tilde{T}_s$):

$$\tilde{T}_s = \sum T_s/6 \tag{7}$$

The average Nusselt number (Nu) is calculated as follows:

$$Nu = \frac{h_i D_h}{k} \tag{8}$$

The Reynolds number at inlet of test section for different flow rates is obtained as follows:

$$Re = \rho V D/\mu \tag{9}$$

The friction factor coefficient (f) is obtained as:

$$f = \frac{\Delta P}{\left(\frac{L1}{D_i}\right)\left(\rho \frac{V^2}{2}\right)} \tag{10}$$

Where V is mean velocity of cold water.

Average performance ratio $R3(Nu_a/Nu_c)$[9],is calculated for predicting heat transfer performance of given insert for constant area consuming same pumping power for plain tube and inserted tube. For assuring this, new Nusselt number based on equivalent Reynolds number.

$$f_a \times Re_a^3 = f_p \times Re_c^3 \to Re_c^{2.75} = f_a \times \frac{Re_a^3}{0.079} \tag{11}$$

where (f_a) is friction factor of inserted tube at given inserted tube Reynolds number, Re_a. The Nusselt number for equivalent plain tube Reynolds number, Re_c is calculated using Dittus Boelter equation

$$Nu_c = 0.023 Re_c^{0.8} Pr^{0.4}$$

(12)

Average performance ratio is calculated as follows:

$$R3 = \frac{Nu_a}{Nu_c} \quad (13)$$

IV. RESULTS AND DISCUSSIONS

4.1. Validation of plain tube [1]

Average Nusselt number (*Nu*) and friction factor (*f*) are calculated for plain tube validation compared with established equations as given below:

$$Nu = 0.023Re_p^{0.8}Pr^{0.4} \text{(DittusBoelter)} \quad (14)$$

$$Nu = \frac{(f/8)(Re-1000)\,Pr}{1+12.7(f/8)^{1/2}(Pr^{2/3}-1)} \text{ (Gnielinski)} \quad (15)$$

$$f = 0.079Re^{-0.25} \text{(Blasius)} \quad (16)$$

Results are comparison is shown in figs.3 and 4. It is seen that the variation is not more than 10% for average Nusselt number, *Nu* and 8% for friction factor *f*. Hence plain tube validated.

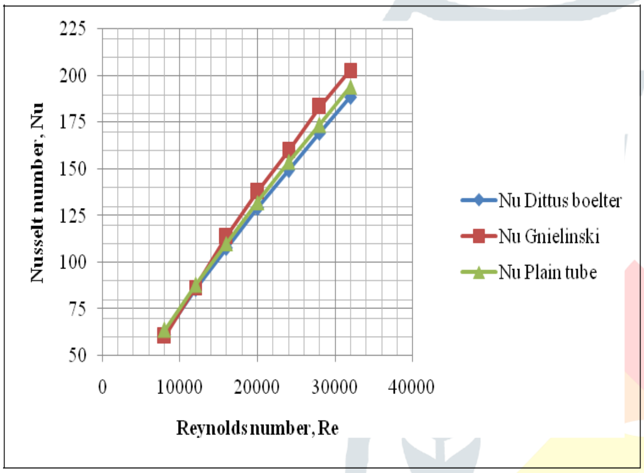

Fig.3. Plain tube Nusselt number

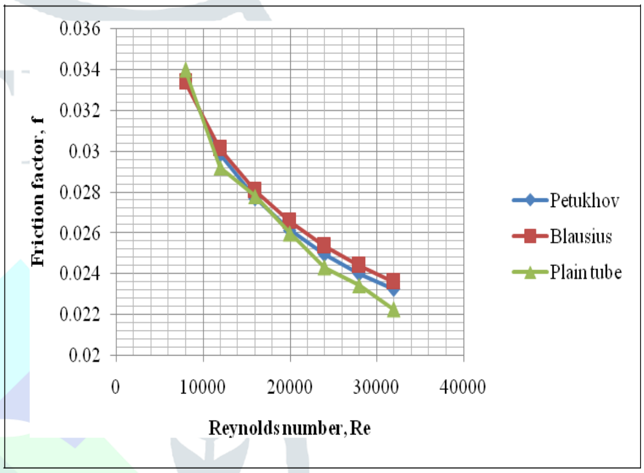

Fig. 4 Plain tube friction factor

4.2 Effect of pitch on heat transfer and pressure drop of test section:

The Nusselt number ratios for inserts are shown in fig.5. As Cup shape inserts create sudden obstacle and increase residence time of flow. It is also seen that no swirl is imparted to the flow, hence turbulence is not created. Heat transfer enhancement is obtained because of increased residence time of flow and waviness imparted to the flow near cup. It is observed that heat transfer rate is enhanced for lower pitch and reduces for higher pitch.

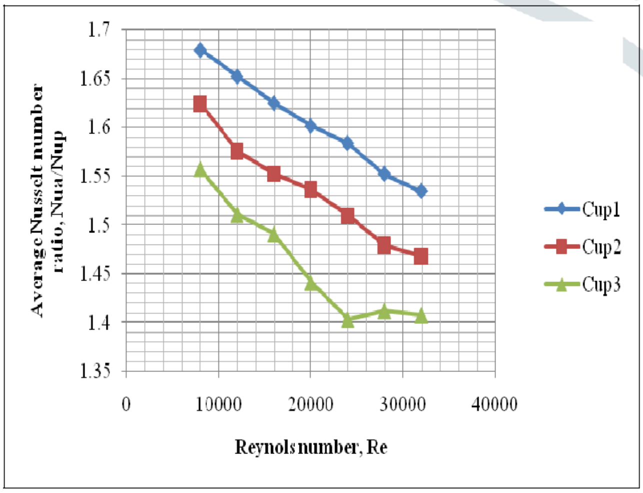

Fig.5 Average Nusselt number ratio Vs Reynolds number

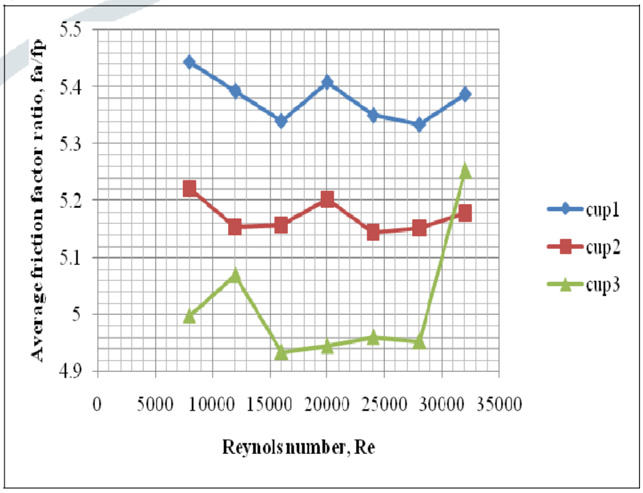

Fig.6 Average friction factor ratio Vs Reynolds number

Results show that the friction factors are increased by approximately 5 times for all cup shape geometries as shown in fig.6 due to sudden obstacle in the flow field. It is observed that friction factors ratios are higher for low pitch and vice versa. Low pitch increases number of obstacles in the flow field.

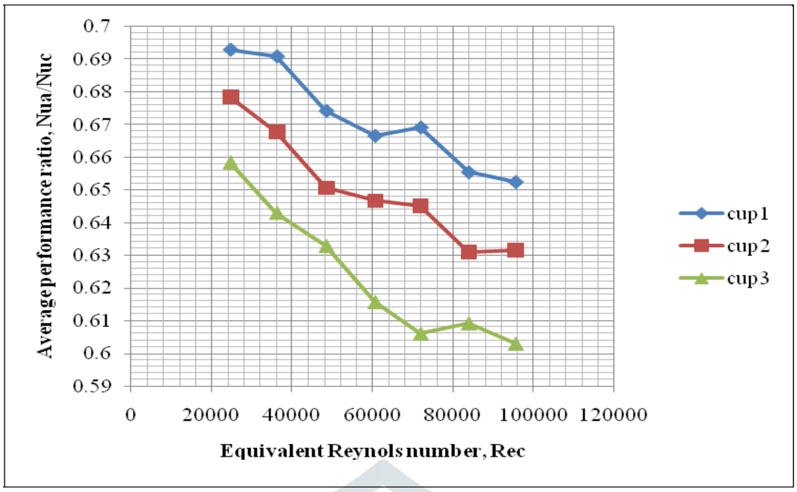

Fig. 7 Average performance ratio Vs Equivalent Reynolds number

Although lower pitch increases the heat transfer rate, friction factors are also significantly increased resulting in increased pumping power requirements. Hence equal pumping power criteria are applied to judge the performance of three inserts. Using equal pumping power criteria, equivalent Nusselt numbers are evaluated and performance ratios are compared as shown in fig.7. It is seen that performance ratios for all three geometries of cup shape insert are less than 1.

4.2 Conclusion:

Test results for three geometries of cup shape with threaded core rod inserts are reported in this study. Results reveal that, heat transfer rate increases at lower pitch of insert but pressure drop is also increased significantly. Based on equal pumping power criteria, performance ratios for all geometries were found to be less than unity. Hence tested geometries of inserts are not recommended for applications where power saving is a prime concern.

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