

Enhancement of Thermo-hydraulic Performance of Tube Heat Exchanger with Perforated Conical Ring Inserts

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Abstract—In this study the effect of perforated conical turbulators on heat transfer, friction factor and thermal performance of the heat exchanger is estimated through investigation. The different parameters used for the experimentation includes, fixed diameter ratios ($d/D = 0.5$), pitch ratios ($l/D = 1, 2, 3$) and number of perforation hole ($N = 0, 2 \& 4$). The experiments were conducted in the range of Reynolds number lying between 3,000 to 21,000 with a test section of 1.5 m length pipe made of copper with a hydraulic diameter (D) of 68 mm. It was observed that Perforation played a major role in controlling the friction factor.

Keywords— Conical turbulators; heat exchanger; thermo-hydraulic; perforation hole; pitch ratios.

I. INTRODUCTION

Heat exchanger is a device which can be utilized to transfer thermal power or entropy between two or more fluids or between a superb floor and a fluid, or between strong particulates and a fluid, at distinctive temperatures and in thermal contact. Heat transfer development by creating turbulence in the physical behaviour of fluid flow inside the heat exchanger tube has become a really interesting area for the investigators. Many techniques have been investigated on enhancement of heat transfer rate to reduce the size and cost of the involving equipment especially in heat exchangers. In the past few decades, several studies on passive and active techniques of heat transfer augmentation have been reported. Sarada [1] experimentally investigated the heat transfer improvement by using twisted tape varying width inserts. The three twist ratio (TR) 3, 4, 5 and width 10, 14, 18, 22, 26mm were used in the experiment. The Reynolds number was varied from 6000 to 13500. Chokphoemphun [2] examined heat transfer and flow friction characteristics in a circular tube inserted with single, double, triple quadruple twisted tapes, co and counter twisted tape for the turbulent regime under a uniform wall heat flux. The Reynolds number was in the range of 5300 to 24000. By using multiple twisted-tapes, the Nusselt number was in the range of 1.15-2.12 times while the friction factor was 1.94-4.06 times that of the plain tube. Eiamsa-ard [3] investigated thermo-hydraulic performance of heat exchanger tubes fitted with circular-rings and twisted tapes. The study analysed three different pitch ratios ($l/D = 1.0, 1.5, 2.0$ and twist ratio ($y/W = 3, 4, 5$ with air as working fluid for Reynolds number between 6000 and 20000. Promvong [4] worked on thermo-hydraulic performance of heat exchanger by using coil pitch ratios (CR) = 4, 6, 8 and twist ratios TR ($y/W = 4, 6$ with Reynolds number range 3000 to 18000. The

results showed that the combination of wire coil and twisted tape turbulators compared with smooth tube at constant power doubled thermal performance. Eiamsa [5] investigated the heat transfer by using single twist ratio TR ($y/W = 4$ and short length at entry test section using tape Length ratio (LR) = 0.29, 0.43, 0.57 and (1.0) for Reynolds number range 4000 to 20000. Matani [6] determined the heat transfer enhancement using (swirl flow generator), twist ratio ($y/W = 3.5, 2.66$ and 2.25 and wire coil (co-swirl flow generator) and pitch ratio, PR ($l/D = 1.17, 0.88$ for Reynolds numbers range between 5000 to 18,000. The enhancement devices of the tube and the co-swirl show a considerable improvement of heat transfer rate and thermal performance as can pair to smooth tube, depending upon on twist ratios. Bhuiya [7] determined the double counter twisted tape four different twist ratio ($y/W = 1.95, 3.85, 5.92, 7.775$ for air as working fluid and Reynolds number range from 6950 to 50,050. Results showed that thermal performance factor improves upto 1.34 times when compared to plain tube. Mohammad and Danesh [8] observed that the friction factor increases significantly at high mass fraction and reduces with increase in flow velocity. Finally the authors concluded that there was a friction loss of up to 8 times and the split dimple fin results in 60% to 175 % gain of heat transfer than the plate fin.

Gupta and Bhatt [9] demonstrated that the heat transfer rate increases when the temperature difference of the particle and environment increases. The study also proved that whenever the heat transfer increases, the convective heat transfer also gets increased. Gau and Lee [10] achieved experiments to study slot air jet impingement cooling flow and the heat transfer along triangular rib-roughened walls. The experiments were conducted using nozzle to plate spacing ratio from 2 to 16, width to rib height ratio from 1.17 to 6.67, rib pitch to height ratio from 2 to 4 and Reynolds number ranging from 2500 to 11000. Saleh and Abdel [11] investigated both experimentally and numerically the effect of dimple on effectiveness of heat exchanger. The value of parameters considered were dimple depth ($d/D = 0.2, 0.3, 0.4$, and Reynolds number in a range of 50-3000.

II. DETERMINATION OF PERFORMANCE ATTRIBUTES OF HEAT EXCHANGER

This study considers a multifold objective of determining the relationship and affect among Nusselt number, friction factor and thermo-hydraulic performance on different flow

and geometrical parameters in heat exchanger tube with conical insert [12]. To realize the desired objectives of the study an experimental setup was designed and fabricated and measuring instruments were calibrated using standard calibrating machines before installing it on the experimental setup [13]. The material with requirement used for the fabrication of experimental setup is given below in Table I.

TABLE I. Materials with specifications

Device	Specification
Centrifugal blower	3HP Motor
GI Flanges	68 mm diameter with internal thread
Fasteners	Nuts and bolts (12mm and 80mm) diameter, length.
Copper Pipe	68 mm diameter, 6000 mm length.
Variac transformer	Up to 5 Amperes and 300 volts supply
Ammeter and volt meter	Digital
Heating element	Nicrome wire
Aluminium oxide	Neutral
Anbond 666T Plus	High temperature binder
Thermocouple	T-Type

Orifice plate	35 mm diameter
U tube manometer	300 mm
Gate valve	68 mm diameter
Glass wool fiber	Insulating tape
Temperature Scanner	24 channel

III. INSERT GEOMETRY AND PARAMETERS

In this study emphasis is given on the effect of perforation in the conical ring. Perforated conical ring is used in this work along with solid conical ring [14]. The parameters used in the experimentation are mentioned in table II. The schematic view of insert geometry with different representation is shown in Fig. 1, Fig. 2 and Fig. 3 respectively.

TABLE II. Parameters considered in the study

Name of Parameter	Specification	Range
Pitch ratio (PR)	l/D ratio	1, 2 & 3
Diameter Ratio (DR)	d/D ratio	0.5
Number of hole (N)	N	0, 2 & 4
Reynolds Number (Re)	Flow parameter	3,000 - 21,000

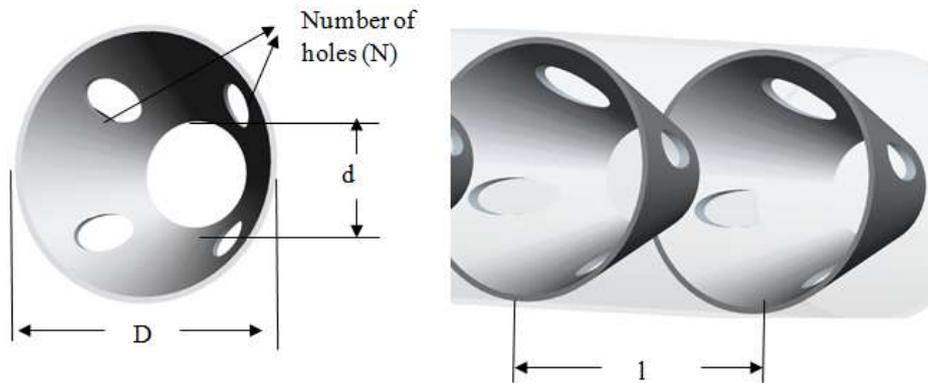


Fig. 1. Schematic view of insert geometry.



Fig. 2. Schematic view of perforated conical rings with N= 0, 2, 4 respectively.

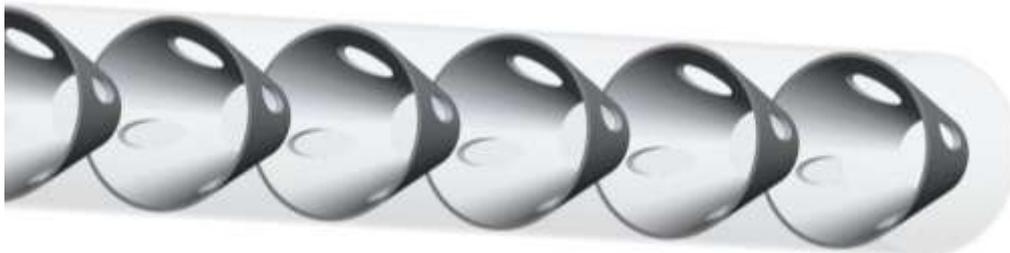


Fig. 3. Schematic view of tube assembly with inserts with N= 4 & PR=1.

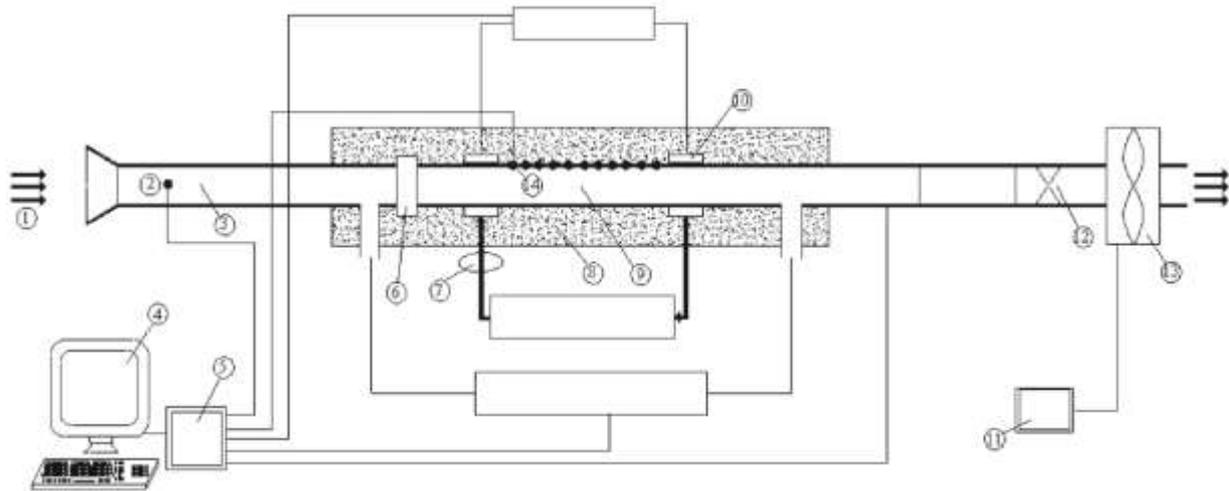


Fig. 4. Schematic view of experimental setup.

IV. DESIGNING OF EXPERIMENTAL SETUP

The schematic diagram of experimental setup is shown in Figure 4. The experimental facility includes a centrifugal blower with 3HP electric motor, entrance for uniform entry of the air, a flow meter to measure the volumetric flow rate, a 2000 mm calming tube for developing flow hydro dynamically and the heat transfer test tube with twisted tape with solid rings inserts [15]. The study considers a copper test tube in 1.5 meter test section and applies a uniform heat flux to the test tube by heating element of Nicrome wire. The electrical output will be controlled by a variac transformer up to 5A and 300V supply which provides a steady heat flux beside the entire length of test section. The outer surface of test tube was made up of insulated glass wool fiber to reduce the convective heat loss to the surroundings [16]. The surface temperatures of the tube wall were measured by 15 T-type thermocouples which will be positioned on the limited wall of the tube and calibrated before being used. The inner and outer temperatures of bulk air were also measured by T-type thermocouples at certain points [17]. The heated air in the test tube was removed from the experimental field to the atmosphere by a piping system during the experiments. The experiment was carried out by varying the flow rate and electrical power [18-20]. The temperature of heated test tube during the experiment and data of volumetric flow rate, pressure drop and temperature of the bulk air electrical output was recorded. The components highlighted in the above figure are listed in Table III [21]. The experimental setup of the proposed system is shown in Figure 5.

TABLE III. Components used in the setup

S.No	Component	S.No	Component
1	Inflow	8	Insulation
2	Inlet temperature measurement point	9	Test tube
3	Calming tube	10	Current
4	Computer	11	Inverter
5	Data logger	12	Orifice
6	Coupling unit	13	Blower
7	Current transformer	14	Thermocouples



Fig. 5. Experimental setup.

V. RESULTS

Experimental data was recorded in terms of temperature, pressure, and flow rate and then calculations were done by a sequence of data reduction steps and with the help of mathematical formulae and systematic mathematical calculations [22-23]. The values of attributes considered are shown in Table IV. The graphs obtained for validation curve for Nusselt number and friction factor are shown with the help of Fig. 6 and Fig. 7 below.

TABLE IV. Attributes considered for calculation.

A_o	Cross section area of orifice	0.000962 m ²
C_d	Coefficient of discharge for orifice meter	0.6
C_p	Specific heat of air at constant pressure	1006.444 J/kg K
D	Hydraulic diameter of the pipe	0.068 m
h	Convective heat transfer coefficient	90.8465 W/m ² K
K	Thermal conductivity of air	0.02629 W/m K
L	Length of the pipe	1.5 m
\dot{m}	Mass flow rate of fluid	0.02179 kg/s
ΔP	Pressure drop across test section	214 Pa
ΔP_o	Pressure drop across the orifice plate	588.011 Pa
T_i	Fluid inlet temperature	18.09 K
T_o	Fluid outlet temperature	38.83 K

μ	Dynamic viscosity	1.85e-05 kg/m s
η	Thermal performance factor	1.332014
Re	Reynolds number	22839.37
Pr	Prandtl number	0.707508
Nu	Nusselt number	61.45649

The experimentation was carried out using different geometrical and flow parameters [24-26]. The effect of each parameter on heat transfer, friction factor and thermal performance factor of heat exchanger tube are shown with the help Figures below. The Fig. 8 and Figure 9 shows variation of Nusselt number with respect to flow parameter i.e. Reynolds number for different insert geometries. The Fig. 10 and Fig. 11 shows a variation of friction factor with respect to flow parameter i.e. Reynolds number for different insert geometries.

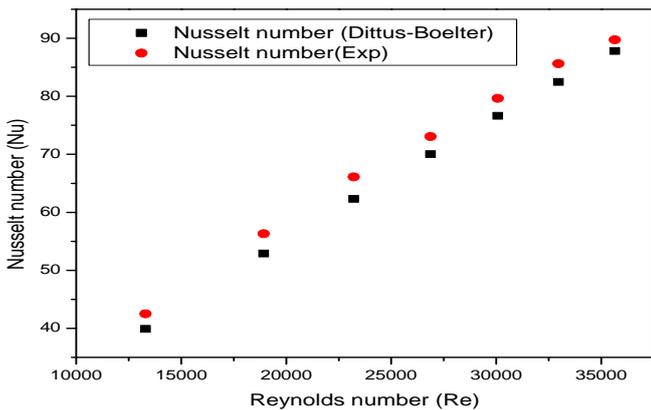


Fig. 6. Validation curve for Nusselt number (Nu).

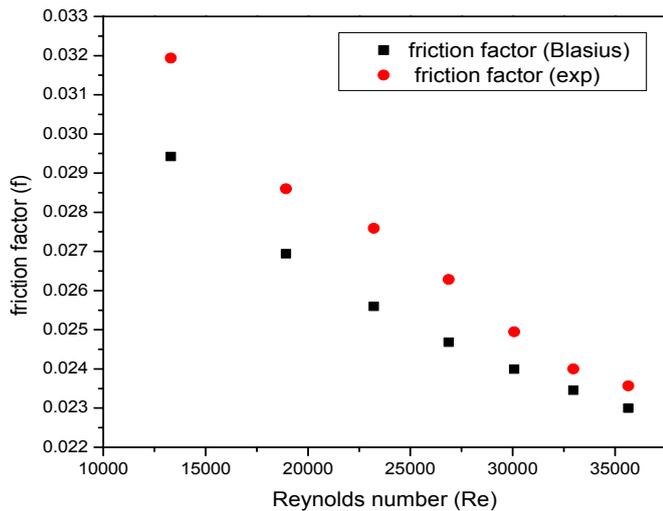


Fig. 7. Validation curve for friction factor (f).

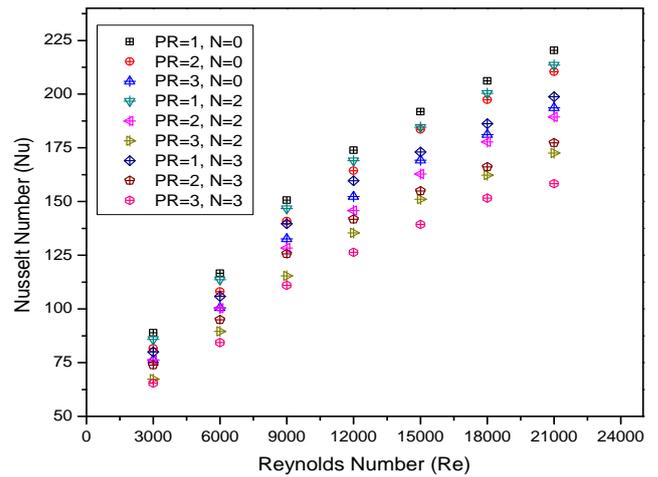


Fig. 8. Nusselt number (Nu) versus Reynolds number (Re).

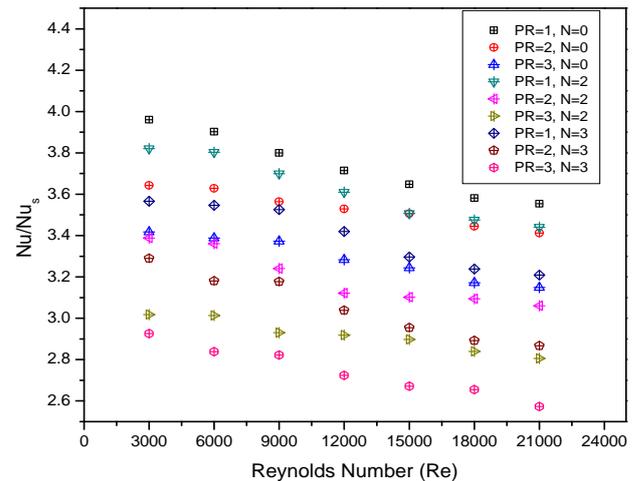


Fig. 9. Nusselt number variation with respect to smooth tube (Nu/Nu_s)

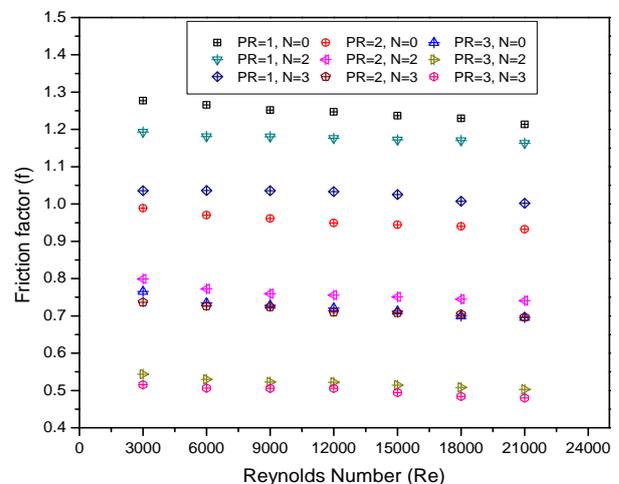


Fig. 10. Friction factor (f) versus Reynolds number (Re)

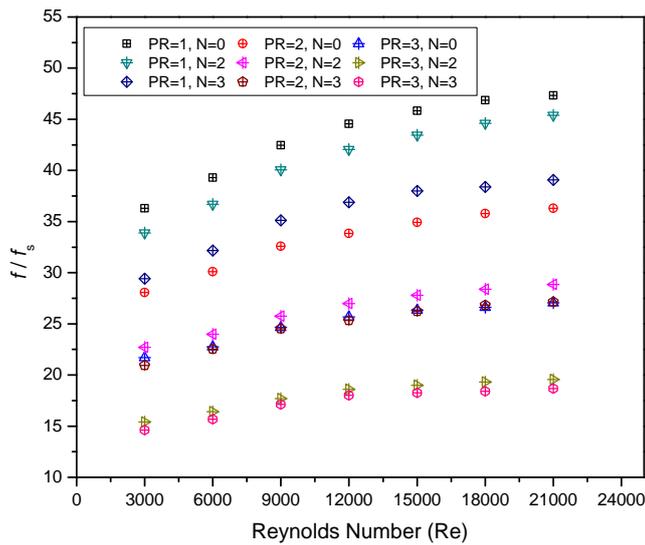


Fig. 11. Friction factor variation with respect to smooth tube (f/f_s)

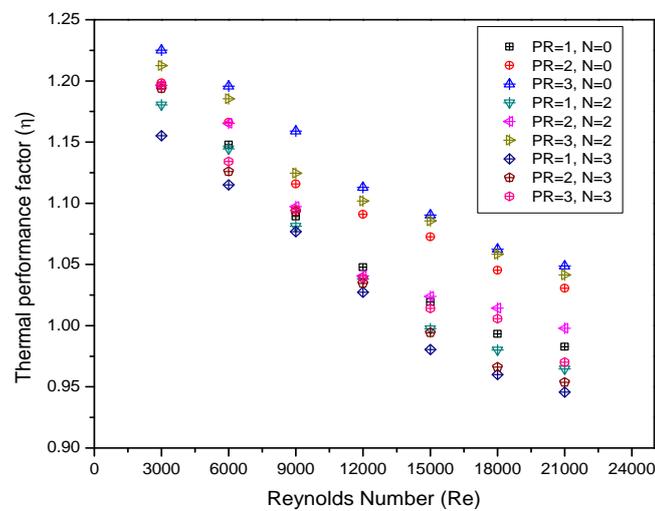


Fig. 12. Thermal performance factor (η) versus Reynolds number (Re).

VI. CONCLUSION

This research work was preoccupied with the objective of conducting depth investigations on heat transfer and friction factor characteristics of perforated conical ring as roughness components of heat exchanger tube. The experimental results has been presented for Nusselt number and friction factor as geometrical parameters able to certainly justify the influence of that parameters on heat transfer and friction factor characteristics. It was observed that Nusselt number of the perforated conical ring heat exchanger tube increases with increase in Reynolds number. Friction factor of the perforated conical ring heat exchanger tube decreases with increase in Reynolds number. Friction factor decreases with increase in stream wise spacing and attained minimum values for stream wise spacing value of $PR=3$. Also the friction factor decreases with increase in pitch ratio. Finally it was observed that the maximum enhancement in thermo hydraulic performance factor of the investigated system is 22% than that of smooth heat exchanger tube.

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