Thermal and Thermohydraulic Behavior of Protruded Circular Tube

Prashant Kumar, Sunil Chamoli, Manoj Kumar

Abstract: The present study inculcates the heat transfer and fluid flow behavior in protruded circular tube of a heat exchanger. The experiments were carried out to collect the data for Nusselt number and friction factor in the investigated range of geometrical and flow parameters viz. Reynolds number(Re) : 6000-35000, span wise spacing (y/d) :10 and stream wise spacing (x/d): 10-40, relative protrusion height (e/d) :1. The experimental results show a significant effect of the presence of protrusion on the heat transfer rate and friction loss over the conventional smooth heat exchanger tube for constant mass flow rate. The enhancement in Nusselt number and friction factor of protruded circular tube over smooth tube was found of the order of 3.43 and 5.54 times, respectively. The best thermo hydraulic performance factor of 2.31 is found for the tube with protruded geometry of span wise spacing (y/d=10) and stream wise spacing (x/d=10).

Keywords: Nusselt number, thermo hydraulic performance, fluid flow, heat exchanger.

NOMENCLATURE
Re = Reynolds number

Nu = Nusselt number
Pr = Prandtl number
Qu= Useful Heat Transfer (W)
ṁ = Mass flow rate (kg/s)
h = Heat transfer coefficient (W/m2K)
V= Air flow velocity (m/s)
k = Thermal conductivity of air (W/m K)
Cp = Specific heat (J/kg K)
e = Height of protrusion (mm)
d = Diameter of protrusion (mm)
D = Hydraulic diameter of pipe (mm)
Ti = Inlet temperature (K)
To = Outlet temperature (K)
Tfm= Mean fluid temperature (K)
Tpm = Mean pipe temperature (K)
Ao= Area of orifice plate (m2)
Ap = Area of pipe (m2)
Cd= Coefficient of discharge
β= Open area ratio
ΔP = Pressure drop (Pa)
ΔPo = Pressure drop across orifice plate (Pa)
x/d = Stream wise spacing
y/d = Span wise spacing

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I. INTRODUCTION

Heat exchanger is a device facilitating transfer of heat between two or more fluids or from one medium to other medium.In modern applications viz. air conditioning, refrigeration system, radiators, and chemical process, the heat exchanger is used to enhance the heat transfer rate from one medium to other. In general, there are two methods of heat transfer enhancement in heat exchanger. First is the active method that requires the extra external power sources such as fluid vibration, jet impingement and injection. The other one is the passive method that requires no other power sources it means that there is no need of any kinds of external forces. Several investigations have been carried out to study the effect of turbulators for example dimpled or grooved tube, twisted tapes and protruded tube. **Xiao et. al. [1]** reported that the thermal performance parameters were higher in a heat exchanger with a dimpled bottom and smooth top than in a heat exchanger with a dimpled bottom and protrusions on top in laminar region, and proposed friction factor ratio and Nusselt number ratio correlations for a heat exchanger with dimpled bottom and smooth top. **Sahu and Bhagoria [2]** experimentally investigated the effect of 90˚ broken ribs as roughness elements and found that thermal efficiency lies in between 51% to 83.5%. **Terekhov et. al. [3]** stated that the heat transfer enhancement of the dimple is mainly due to auto oscillations generated by the dimple under turbulent flow regime, which depends on the depth and radius of the dimple. **Karwa et. al. [4]** investigated the effect of rib chamfered angle (ϕ), duct aspect ratio on heat transfer and friction factor using integral chamfered ribs. The experimental data show that the chamfered angle of 15˚ gives highest Nusselt number as well as friction factor. **Ligrani et. al. [5]** discusses flow structure and local Nusselt number variations in a channel with dimples and protrusions on opposite channel walls. Instantaneous flow visualization images and surveys of time-averaged flow structure show that the protrusions result in added vortical, secondary flow structures and flow mixing. As a result, local friction factors and Nusselt numbers are augmented compared to a channel with no protrusions on the top wall. **Mahmood et. al. [6]** indicates that important Nusselt number variations are observed as the array of protrusions is changed with respect to the locations of the dimples. With protrusions, form drag and channel friction are increased. As a result, thermal performance parameters are then generally slightly lower when protrusions and dimples are employed, compared to a channel with a smooth dimple arrangement. **Burgess and Ligrani [7]** proposed a Nusselt number ratio correlation as a function of the dimple print diameter and dimple depth in a heat exchanger with dimple on the bottom and a smooth top. **Kore and Sane [8]** concluded that the dimple surface with uniform heat flux have relatively low heat transfer coefficient on the leading edge of the dimple and high on the trailing edge and the flat area immediately downstream of the dimple. **Chen et. al. [9]** numerically studied the flow and heat transfer features in fully developed and found that the heat transfer rate distributes asymmetrically on protrusions when the protrusion height ratio (h/D) is large, the associated enhanced heat transfer is attributed to the asymmetric flow structure and vortex inside the wake behind the protrusion. **Bhushan and Singh [10]** studied the influence of Reynolds number on heat transfer coefficient distribution on the surface having staggered array of the protrusion geometry. The enhancement in heat transfer rate was about 2.5 times than smooth surface value over a range of Reynolds number. **Afanasyev et. al. [11]** experimentally studied the heat transfer enhancement mechanism for flows in a dimpled channel with various different shapes. Enhancements in heat transfer was found to be about of 30 to 40%, with pressure losses that are not increased appreciably relative to a smooth surface are presented. **Elyyan and Tafti [12]** numerically investigated the flow characteristics and Nusselt number distribution in a heat exchanger with a dimple bottom and protrusion on top using a large-eddy simulation, and reported that heat transfer augmentation was higher in the turbulent region due to

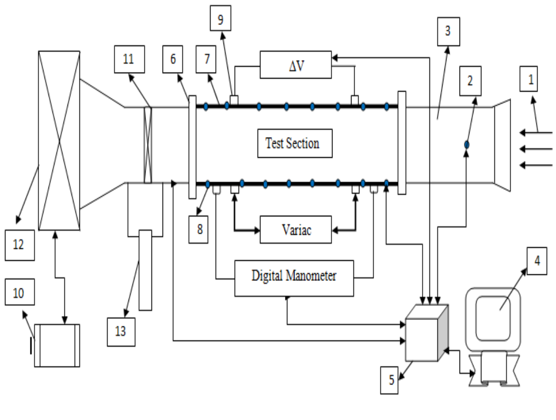

Fig. 1 Schematic view of experimental setup

1. Air inlet flow, 2. Inlet temperature, 3. Calming tube, 4. Laptop, 5. Data logger, 6. Flange, 7. Insulation, 8. Thermocouples 9. Current, 10. Power source, 11. Orifice plate, 12. Blower, 13. U-Tube Manometer

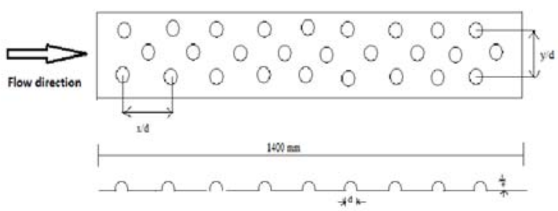

Fig. 2 Schematic view of geometry of protrusion

oscillatory flow. This paper presents an experimental study of fluid flow and heat transfer enhancement in protruded surface heat exchanger tube. The main aim of the present study is to enhance the heat transfer rate and thermo hydraulic performance factor by using the protruded surface heat exchanger tube for Reynolds number range of 6000-35000. Using the experimental results, correlations for heat transfer, friction factor and thermo hydraulic performance factor are also developed.

II. EXPERIMENTAL DESCRIPTION

A. Experimental setup

A schematic diagram of the experimental setup is given in Fig. 1. The experimental setup consists of suction blower with the capacity of 3 KW, orifice plate to measure the flow rate, and the test section. In the heating section, a galvanized iron tube having an inside diameter (D) of 68 mm, and outside diameter (D_o) of 72 mm, and length (L) of 1400 mm is wound with electrical heating element covered with the paste of alumina oxide (Al_2O_3) and Ana bond 666 T-plus. The outer surface of the test tube was well insulated to minimize the convective heat transfer loss to the atmosphere, and necessary precautions were taken to prevent leakage from the systems. The terminal of the wires is connected to a variac transformer, which is used to control the AC current passing through the heating wire and to keep the current less than 3 ampere.

The temperature of the of the bulk air were measured at certain points with data logger in conjunction with the T-type thermocouples. Fourteen thermocouples were tapped on the wall of the test section, one thermocouple at inlet of the tube and the other one is fixed at exit section of the tube. The average pipe temperature was determined by means of calculations based on the reading of the T-type thermocouples

B Experimental procedure

In the apparatus, the inlet air at 16.23 ˚C from a 3 KW centrifugal blower was directed through an orifice plate and passed through the test section. The air flow rate was measured by the orifice meter. The air flow rates from the blower were adjusted by varying motor speed, placed before the exit section. An inlet and outlet temperature of the bulk air from the heat exchanger tube has been measured by T-type thermocouples. For every reading, it was compulsory to record the data of temperature, pressure drop, and mass flow rate at steady condition. Various characteristics viz. Nusselt number, Reynolds number, friction factor, mass flow rate were based on the mean pipe temperature and outlet temperature. The heat transfer rate of the heat exchanger tube has been measured by using the characteristics like inlet and outlet temperature, pressure drop across the test section and air flow rate.

Fig. 3 Photographic view of protruded tube (x/d=10 and y/d= 10)

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The uncertainty estimates obtained for protruded surface heat exchanger tube for the non-dimensional parameters are ± 8.67% for Reynolds number, ± 2.79% for Nusselt number, ± 8.67% for friction factor, ± 2.79% for heat transfer coefficient and ± 2.75% for useful heat gain.

The deviation between predicted and experimental values for Nusselt number, friction factor and thermo hydraulic performance factor are of the order of ± 10%, ± 15% and ± 12% respectively.

III DATA REDUCTION

The experimental data like pressure drop across orifice and test section, air and temperatures at different locations in the tube has recorded under Quasi-steady state conditions at different mass flow rates of air. The heat transfer rate of air flowing in the tube has been calculated by using the collected data. The following equations were used to determine the various results.

Pressure difference across the orifice meter obtained from the calibrated U–tube manometer is used for the determination of mass flow rate from the following relationship:

$$m = C_d \times A_o \times \left[\frac{2.\rho.(\Delta P)_o}{1-\beta^4}\right]^{0.5} \qquad (1)$$

Where $(\Delta P)_o = 9.81 \times (\Delta h)_o \times \rho_m \times \sin\theta$ (2)

Reynolds number corresponds to air velocity (*V*) can be expressed as:

$$\mathrm{Re} = \frac{\rho V D}{\mu} \qquad (3)$$

The friction factor is obtained from the measured values of pressure drop $(\Delta P)_d$ across the test section length using Darcy equation as,

$$f = \frac{(\Delta P)}{\left\{\left(L/D\right)\left(\rho V^2/2\right)\right\}} \qquad (4)$$

The average heat transfer coefficient was calculated from the heat transfer from the protruded surface tube to the air. The value of the heat transfer coefficient can be expressed as:

$$h = \frac{Q_u}{A_s\left(T_{pm} - T_{fm}\right)} \qquad (5)$$

The convective heat transfer coefficient (h) has been used to obtain the average Nusselt number as follows:

$$Nu = \frac{hD}{k} \qquad (6)$$

The value of useful heat gain was calculated from the following expressions:

$$Q_u = mC_p(T_o - T_i) \qquad (7)$$

The value of thermo hydraulic performance factor was calculated by applying the given expression:

$$\eta = \left[\frac{Nu/Nu_s}{(f/f_s)^{1/3}}\right] \qquad (8)$$

IV RESULT AND DISCUSSION

The average values of Nusselt number and friction factor were determined by using formulas. These values have been compared with the values completed from the standard correlation like Dittus-Boelter equation for Nusselt number and modified Blasius equation for friction factor for the smooth tube.

These equations are given below:

$$Nu = 0.023 \times \mathrm{Re}^{0.8} \times \mathrm{Pr}^{0.4} \qquad (9)$$

$$f = 0.316 \times \mathrm{Re}^{-0.25} \qquad (10)$$

The comparison of experimental and correlated values of Nusselt number and friction factor is shown in Fig. 4 and Fig. 5 respectively. The below comparison ensures that the accuracy of experimental results planned to be obtained from the present experimental data.

The average absolute deviation between the predicted and experimental values of Nusselt number for Dittus-Boelter friction factor for modified Blasius has been found 5.34% and 8.80% respectively. The result shows good agreement between experimental and analytical values and thus confirms the accuracy of experimental data collected with the experimental setup.

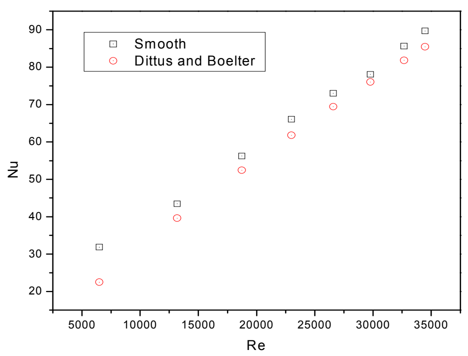

Fig. 4 Comparison of experimental and predicted value of Nusselt number .

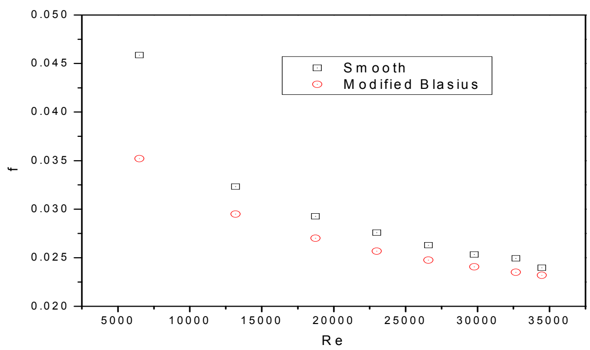

Fig.5 Comparison of experimental and predicted values of friction factor.

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A Effect of Nusselt number

Fig. 6 illustrates the effect of stream wise spacing (x/d) on Nusselt number with Reynolds number ranging from 6000 to 35,000 and at fixed relative roughness height (e/d = 1). As the gap size increases, the swirl component decreases hence reducing the turbulence in the area. The reduced turbulence reduces the heat transfer from surface to air. For this situation. Nusselt number is seen to decrease with increase in stream wise spacing (x/d). It has been ascertained that maximum value of Nusselt number is obtained for Stream wise spacing (x/d) value of 10. For all the protruded tubes, Nusselt number increases with increasing the Reynolds number, the maximum Nusselt number was obtained at the highest Reynolds number (Re = 35000).

Fig. 7 shows the variation of Nu/Nu_s with Reynolds number at different stream wise spacing (x/d= 10, 20, 30, 40) and at constant span wise spacing (y/d = 10) and e/d =1. It is seen that the ratio of Nu/Nu_s increases with increasing the Reynolds number. The Nusselt number ratio tends to extend with the increase of Reynolds number from 6000 to 35000.

B Effect of friction factor

It has been ascertained that friction factor decrease with increase the stream wise spacing (x/d) and attains a maximum value corresponding to stream wise spacing (x/d) value of 10. The minimum friction factor has been found for stream wise spacing (x/d) of 10 and lower friction factor has been found for stream wise spacing (x/d) of 40 at constant span wise spacing (y/d) of 10.

Fig. 9 shows the variation of f/f_s with Reynolds number at different stream wise spacing (x/d= 10, 20, 30, 40) and at constant span wise spacing (y/d = 10) and e/d =1. It is seen that the ratio of f/f_s decreases with increase in Reynolds number. The minimum value of f/f_s is obtained for the case of x/d=40 and y/d=10.

C Effect of thermo hydraulic performance factor

It has been discovered that maximum value of thermo hydraulic performance factor is found for stream wise spacing (x/d) of 10. For all the protrusion, thermo hydraulic performance factor increases with increasing Reynolds number, the maximum thermo hydraulic performance factor was obtained at the highest Reynolds number of 35000. It is obvious that applying protruded surface results in a significant increase in thermo hydraulic performance factor as compare to Reynolds number.

VI CONCLUSIONS

The experimental result of thermal and thermo hydraulic behavior with protruded circular tube has been reported. The higher heat transfer rate was found at span wise spacing (x/d) value of 10 and stream wise spacing value of 10 for all the Reynolds number. At the same Reynolds number, the thermohydraulic performance factor decreases with increasing the steamwise spacing of protrusion.

The experimental result shows the maximum enhancement in thermo hydraulic performance factor of the investigated system is up to 56% than that of smooth heat exchanger tube.

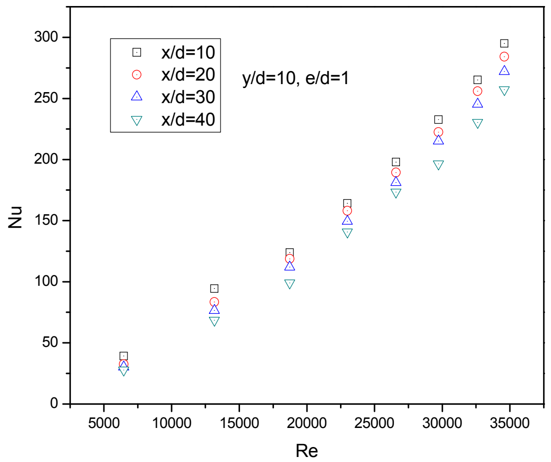

Fig. 6 Variation of Nusselt number with Reynolds number at y/d=10.

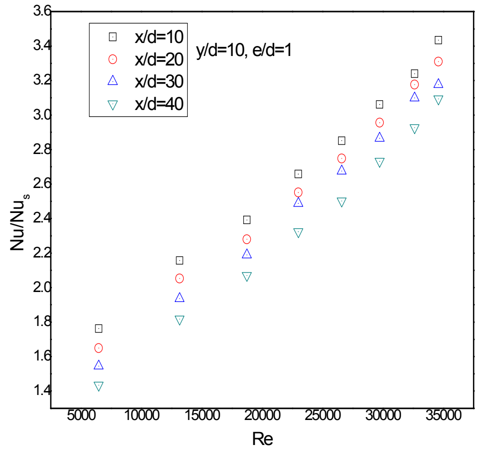

Fig. 7 Variation of Nu/Nus with Re at y/d=10

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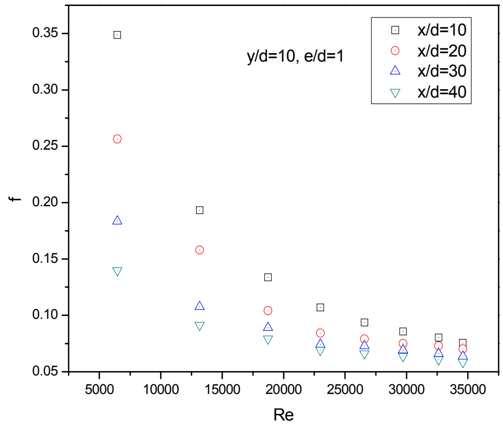

Fig. 8. Variation of Friction factor with Reynolds number for y/d=10 and e/d=1.

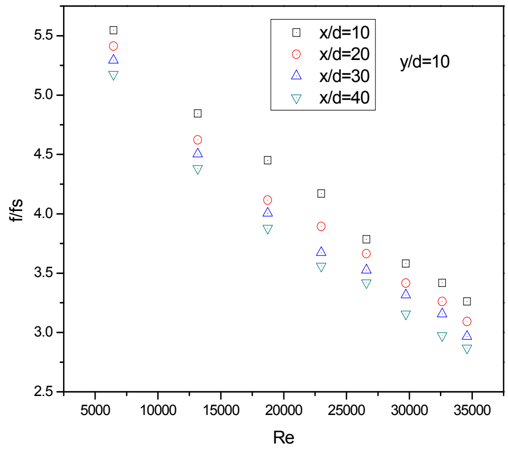

Fig. 9. Variation of f/f_s with Re for y/d=10, e/d=1.

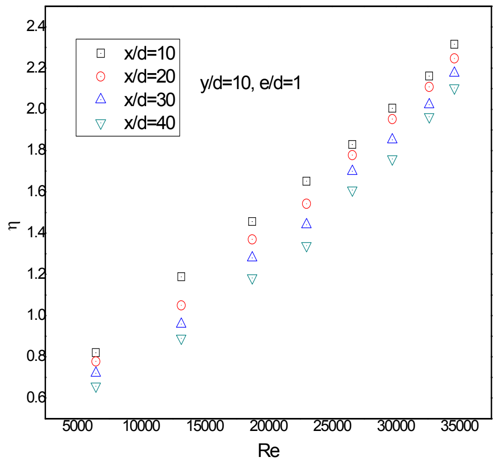

Fig. 10 Variation of thermo hydraulic performance factor with Re for y/d=10, e/d=1.

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