

EXPERIMENTAL ENHANCEMENT OF HEAT TRANSFER THROUGH DOUBLE PIPE PARALLEL FLOW HEAT EXCHANGER USING DISCRETE WIRE ROUGHNESS

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ABSTRACT: This is an experimental analysis; here we use different Roughness in parallel flow heat exchanger for the heat transfer enhancement at same pump power. Mass flow rate is control by using valve from 0.1 to 0.6 kg/s. The present work is based on the heat transfer augmentation technique. In the present work an experimental and numerical analysis of augmentation of laminar and turbulent flow in horizontal concentric pipe in parallel flow heat exchanger with inserts pipe of different pitch to rib height ratio (P/e) varies from 5-15 respectively. water is to be acted as working fluid here. This experiment is done by taking Reynolds number range from 3000-20000 regimes. Using different pitch for breaking laminar sub layer and increasing turbulence and heat transfer rate with increasing Reynolds number. In the ribbed pipe showed that longitudinal swirl flow were generated inside the pipe. By using this roughness in pipe we find that The maximum enhancement in Nusselt number is found to be 3.12 and that of friction factor is obtained to be 2.59 corresponding to (P/e) of 10. The maximum value of Thermal hydraulic performance for (P/e) 10 is varies from 1.5-2.25.

Key words: Heat exchanger, Heat transfer enhancement, Roughness, turbulence, friction factor, pressure Drop.

1. INTRODUCTION

Heat exchangers are device that facilitate the exchange of heat between two fluids that are at different temperature while keeping them from mixing with each other. heat exchanger are commonly used in practice in a wide range of application, from heating and air conditioning systems in a household ,to chemical processing and power production in large plant. Recently there are so many experiment has been done to develop enhancement technique to reduce the size and cost of heat exchangers in order to improve the overall heat transfer performance of heat exchangers. Basically the augmentation technique refers to the improvement of thermal performance of heat exchanger. Enhancement technique is to be classified into three different ways:

- (A) Passive Techniques
- (B) Active Techniques
- (C) Compound techniques

(A) Passive Techniques: This method does not need any external power input and the additional power needed to enhance the heat transfer is taken from the available power in the system. They promote higher heat transfer coefficient by disturbing or the existing flow behavior which also leads to increase in pressure drop. In case of extended surface, the effective heat transfer area on the side of extended surface is increased. Passive technique holds the advantage over the active technique as they do not require any direct input of external power. Heat transfer by these techniques can be achieved by using:

1. Treated surface
2. Rough surface
3. Extended surface
4. Swirl flow devices
5. Coiled tubes
6. Surface tension devices

(B) Active techniques: These techniques are more complex in use and design point of view. This method requires some external source of power input which improves and modifies the heat transfer rate. This technique has not shown much potential owing to complexity in design. This technique not used on wide area due to difficult to provide external power input in many cases. Active technique is as follow:

1. Mechanical aids
2. Surface vibration
3. Fluid vibration
4. Electrostatics field
5. Injection

(C) Compound techniques: A compound method is a hybrid method in which both active and passive methods are used in combination. The compound method involves the complex designs and hence it has limited applications.

Nomenclature

Nu	Nusselt number	Re	Reynolds number
\dot{m}	Mass flow rate (kg/s)	U	Overall heat transfer coefficient
h	heat transfer coefficient	e	Rib height, mm

f	Friction factor	D_i	Inner Diameter
D_o	Outer Diameter	k	Thermal conductivity (W/m-K)
V	Mean velocity, m/s	Δh	Head loss, cm
A_s	Surface area, m^2		
Greek symbols			
μ	Dynamic viscosity of fluid, Ns/m^2	ρ	Density of fluid, kg/m^3
η	Thermal enhancement factor		

2. LITERATURE REVIEW

There are several experimental works done on heat transfer enhancement by using different kind of roughness.

A. Hasanpour, M. Farhadi(2015) [1]: In this paper, heat transfer and friction factor are experimentally studied in a double pipe heat exchanger which has an inner corrugated tube filled with various categories of twisted tapes from conventional to modified types which include perforated, V-cut and U-cut types. The twist ratios are 3, 5 and 7. The hole diameter ratio are 0.11 12 and 0.33, the width and depth ratio of the cuts vary from 0.3 to 0.6 and the Reynolds number is changed from 5000 to 15000 of turbulent regime. The results of the main parameters on heat transfer and pressure drop show that the Nusselt number and friction factor for all cases of twisted tape corrugated tube are more than the empty corrugated tube. Also the Nusselt number and friction factor for corrugated tube equipped with modified twisted tapes are higher than typical tapes except those of perforated types which lead to lower Nusselt number and friction factor.

Nianben Zheng, Peng Liu, Feng Shan(2016) [2]: This paper investigate the effects of rib arrangements on the flow pattern and heat transfer in an internally ribbed heat exchanger tube. Details of the flow structures in the tube with parallel type ribs (P-type ribs) and V shape type ribs (V-type ribs) were presented and analyzed, respectively. The average Nusselt number and friction factor in the V-type ribbed tubes were about 57-76% and 86-94% higher than those in the P-type ribbed tube, respectively.

Amar Raj Singh Suria, Anil Kumar(2017) [3]: This work presents, an experimental study on Nusselt number and friction factor of heat exchanger circular tube fitted with multiple square perforated with square wing twisted tape inserts. The experimental determination encompassed the geometrical parameters namely, width depth ratio (W_d / WT) of 0.042–0.167, perforation width ratio (a / WT) of 0.250, twist ratio (TL / WT) of 2.5, and number of twisted tapes (NT) of 4.0. The effect of multiple square perforated twisted tape with square wing has been investigated for the range of Reynolds number (Re) varied from 5000 to 27,000. The maximum enhancement in Nu and fr is observed to be 6.96 and 8.34 times of that of the plain circular tube, respectively. Correlations of Nu , fr and η are established in term of Re and geometrical parameters of wings twisted tape which can be used to predict the values of Nu , fr and η with considerably good accuracy.

Mohamed Meziane(2017) [4]: In this paper, the design and thermo-hydraulic performance of a double pipe heat exchanger with helical baffles in the annulus side, are investigated numerically. The results obtained for a helically baffled annulus side provide enhanced heat transfer performance and high-pressure drop compared to the simple double-pipe exchangers. Thermal performance and high-pressure drop is an increasing function of baffle spacing and Re . In addition, empirical correlations expressing the results were developed based on curve fitting.

YonghuaYo, Fahui Zhang (2015)[5] To improve the shell side thermo-hydraulic performance of Rod baffle heat exchangers, round rods with arc cuts are used to support staggered tubes in the current investigation, and numerical computation on the turbulent heat transfer enhancement is conducted within two rods (OTWTR) are about 41.9% and 63.8% larger than the counterparts of non-staggered ones respectively, while the pressure loss is doubled. Moreover, it is observed that similar to the pressure drop.

Pawan Singh Kathait (2014)[6] Heat exchangers are used basically in thermal system receiving or rejecting heat with its surroundings. Thermal performance of a system is dependent on the heat exchangers. The ability to transfer heat which is governed by distinct fluid flow characteristics. The present study investigates the effect of discrete corrugated rib roughened tube on heat transfer and frictional losses under varied fluid flow rates. Heat exchanger tubes having corrugated rib roughness with different number of gaps have been tested for pitch to rib height ratio (P/e) of 6-14 by operating under a wide range of flow Reynolds number (Re) from 7500 to 50,000. The maximum enhancements in Nusselt number and friction factor are found to be 2.73 and 2.78 corresponding.

Prasad and Saini [7] [1988], studied the effect of roughness and flow parameters such as relative roughness height (e/D) and relative roughness pitch (p/e) on heat transfer and friction factor. The type and orientation of roughness geometry used have been shown in Fig. 3.4. They developed expressions for the heat transfer and friction factor for a fully turbulent flow. It was observed that maximum heat transfer occurred in the vicinity of reattachment points and reattachment of free shear layer does not occur if relative roughness pitch (p/e) is less than about 8 to 10. Optimal thermo-hydraulic performance is achieved for roughness height slightly higher than the transition sub layer thickness. For relative roughness height (e/D) value of 0.033 and relative roughness pitch (p/e) value of 10, maximum enhancement in Nusselt number and friction factor was reported to be 2.38 and 4.25 times respectively over smooth duct.

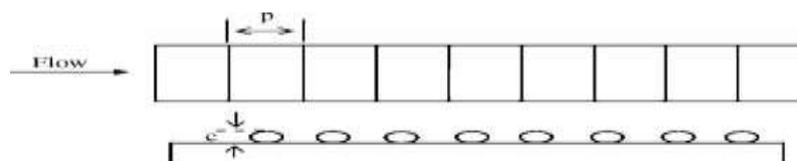


Figure. 1 Transverse continuous ribs

Verma and Prasad [8][2000], conducted an outdoor experimental investigating the thermos hydraulic optimization of the roughness and flow parameters for Reynolds number (Re) range of 5000– 20,000, relative roughness pitch (p/e) range of 10–40 and relative roughness height (e/D) range of 0.01–0.03. The optimal value of roughness Reynolds number ($e+$) was found to be 24 and corresponding to this value, optimal thermos hydraulic performance was reported to be 71%. Heat transfer enhancement factor was found to vary between 1.25 and 2.08 for the range of parameters investigated

3. EXPERIMENTAL SETUP & PROCEDURE

This experiment was done in an open loop experimental facility. The experimental setup used in this experiment is shown in a Fig 3-1 with sketch diagram. It consist different type of equipment's for the measurement like flow meter, heater, thermocouple, pump etc. In this experiment the water is used as a working fluid. Water is stored in a reservoir or in water tank for experiment. From the water tank water is passes through a pump (0.746 KW capacities) for maintain the constant pump power. From the pump water is divided in two sections. One line goes through heater (1kw) for heating water and another for cold water. We use parallel flow heat exchanger for experiment so hot and cold water is needed. The discharge of water at inlet and outlet is measured by using flow meter for mass flow rate. The diameter of inner pipe is to be 13mm (Di) and outer pipe diameter is to be 32mm (Do). The total length of pipe is to be 1800mm. all pipe used in this experiment is galvanized iron pipe. Here we use three different pitches to rib height ratio (P/e) of 5-15 on the inner pipe of heat exchanger for breaking the laminar sub layer and increase turbulence. There are four different points for the temperature measurement (1, 2, 3,4) as shown in fig 3-1. The temperature of hot and cold water at inlet and outlet are measured by using four thermocouple located along the test section. To prevent the water leakage from the end of the shell side, silicone is used on both shell caps.

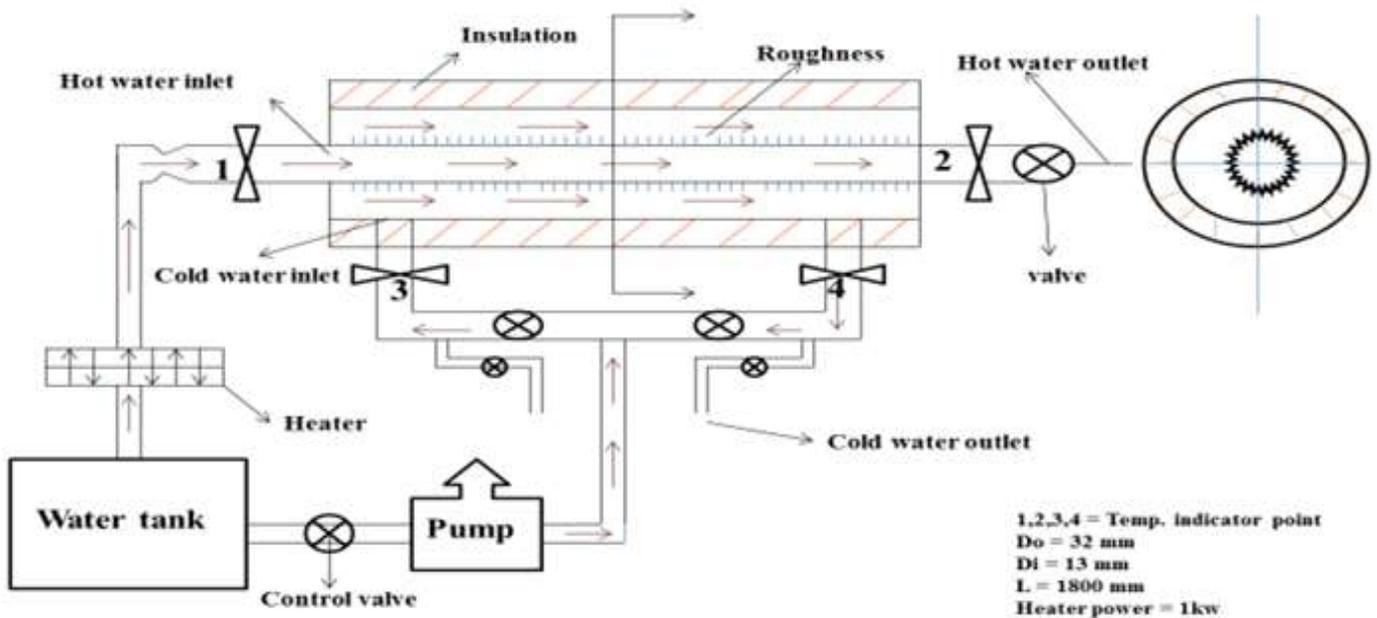


Figure. 2 Sketch Diagram of parallel flow heat exchanger



Figure. 3 Experimental setup of heat exchanger

4. DATA REDUCTION

The data reduction for the given parameter like Nusselt number (Nu), the friction factor (f), the Reynolds number (Re), and thermal enhancement factor (η) are to be used in present study. In this experiment water is used as the working fluid. The steady state heat transfer rate is equal to the heat transfer in a given experiment.

In which,

Where Q_c is the heat transferred to the cold fluid through the inner tube which is calculated?

$$Q_c = \dot{m} C_p (T_{co} - T_{ci})$$

Which \dot{m} , C_p , T_{ci} and T_{co} are the mass flow rate of cold fluid, specific heat of fluid, inlet and outlet cold fluid temperatures, respectively.

Also Q_h which is the heat transferred from hot fluid in the outer tube can be expressed as:

$$Q_h = \dot{m}C_p(T_{hi} - T_{ho})$$

Where \dot{m} the hot water is mass flow rate; T_{hi} and T_{ho} are the inlet and outlet hot fluid temperatures, respectively. It should be mentioned that by conducting a good insulation over the heat exchanger, both the values of Q_c and Q_h are close to each other.

The average Nusselt number and friction factor base on the inner diameter of corrugated tube can be written as below:

$$Nu = \frac{hD_h}{k}$$

where, $D_h = D_o - D_i$

Heat transfer through parallel flow heat exchanger by cold and hot fluid in pipe:

$$Q_{c\ water} = UA\Delta T_{LMTD} = UA \frac{\theta_1 - \theta_2}{\ln(\theta_1/\theta_2)}$$

Overall heat transfer coefficient can be calculated by:

$$U = \frac{Q_{c\ water}}{A \frac{\theta_1 - \theta_2}{\ln(\theta_1/\theta_2)}}$$

In a fully developed Pipe flow the friction factor (f) can be calculated by measuring the pressure drop across the test tube length as:

$$f = \frac{h_f}{\frac{L}{D_h} v^2 \times 2g}$$

The value of Nusselt number for smooth pipe is to be calculated by using Dittus – Boelter Equations:

$$Nu = 0.023Re^{0.8}Pr^{0.4}$$

For the smooth pipe, the friction factor in turbulent flow can be determined from the explicit first petukhov equation is given as:

$$f = (0.790 \ln(Re) - 1.64)^{-2}$$

Nusselt number and friction factor correlation for different types of pipe using roughness:

$$Nu = 0.36Re^{0.59} (P/e)^{0.26}$$

$$f = 0.48Re^{-0.29} (P/e)^{0.30}$$

$$Nu = 0.47Re^{0.58} (P/e)^{0.26}$$

$$f = 0.53Re^{-0.28} (P/e)^{0.32}$$

The thermos hydraulic performance parameter (η) proposed by Webb and Eckert for simultaneous assessment of thermal and hydraulic benefits of the rough pipe is defined as:

$$\eta = \frac{Nu_r/Nu_s}{(f_r/f_s)^{1/3}}$$

ROUGHENED PIPE PARAMETERS

S no.	Parameter	Range/Value
1.	Rib height (e)	2 mm
2.	Rib thickness(t)	1.5 mm
3.	Pipe diameter(D)	32 mm
4.	Pitch to rib height ratio(P/e)	5-15
5.	Relative roughness height(e/D)	0.0625

5. RESULT AND DISCUSSION

Experiments are conducted to generate the heat transfer and friction factor data for the flow through roughed pipe of different pitch to rib height ratio (P/e) 5-15. The Reynolds number varied from 3000 to 20000. In order to observe the effect of system and operating parameter on heat transfer and friction, the Nusselt number and friction factor plots are discussed for different roughness configurations. The heat transfer enhancement is obtained by using roughed pipe is calculated by ratio of Nusselt number of rough pipe and smooth pipe under similar operating condition. The frictional loss is to be obtained by using the ratio of friction factor for rough pipe to that of smooth pipe at similar operating condition. The effect of flow and roughness data on heat transfer and friction, the performance factor is examined for different pitch to rib height ratio (P/e) and the range of Reynolds number considered for the present experiment.

The variation of Nusselt Number along the duct with different value of Relative Roughness in the range of Reynolds number of 3000 – 20000. With increase in Reynolds number, Nusselt number also increases and attains its maximum value at Relative Roughness of 10 and Reynolds number 20000.

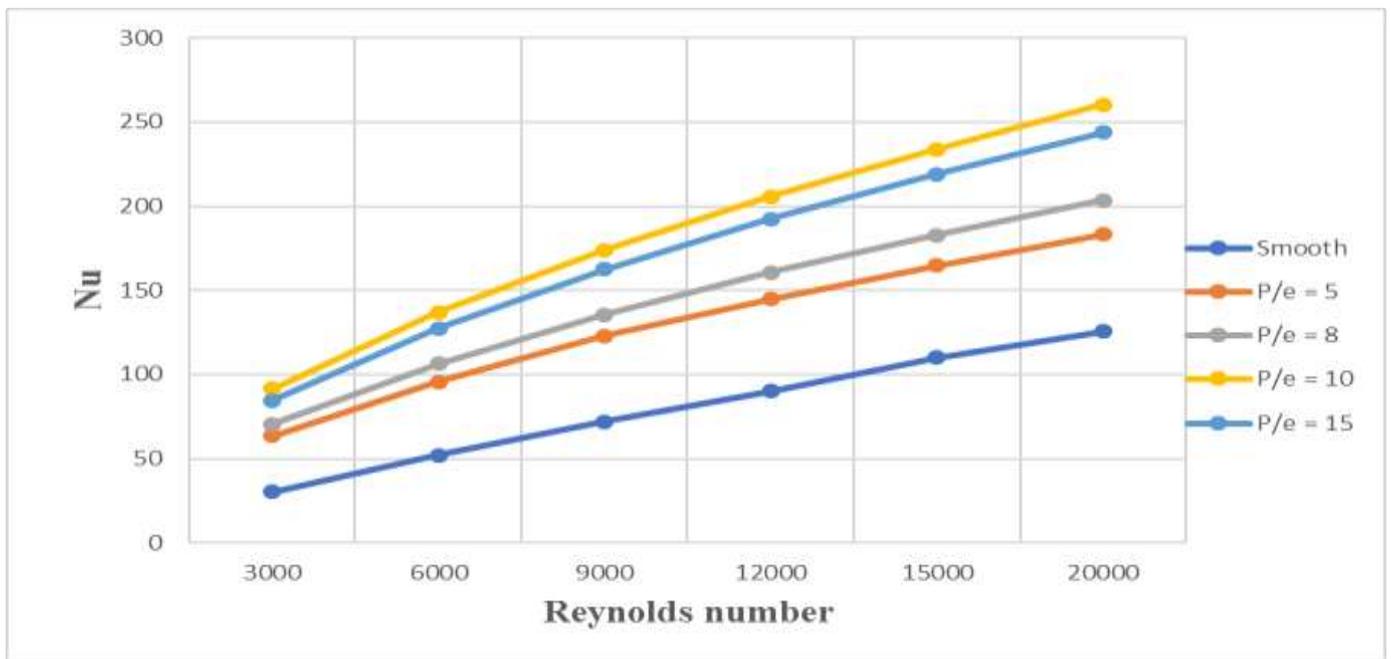


Figure. 4 Variation of Nusselt number and Reynolds number for Smooth and Rough pipe

The variation of Friction factor along the duct with different value of Relative Roughness in the range of Reynolds number of 3000 – 20000. With increase in Reynolds number, Friction factor decreases and attains its maximum value at Relative Roughness of 10.

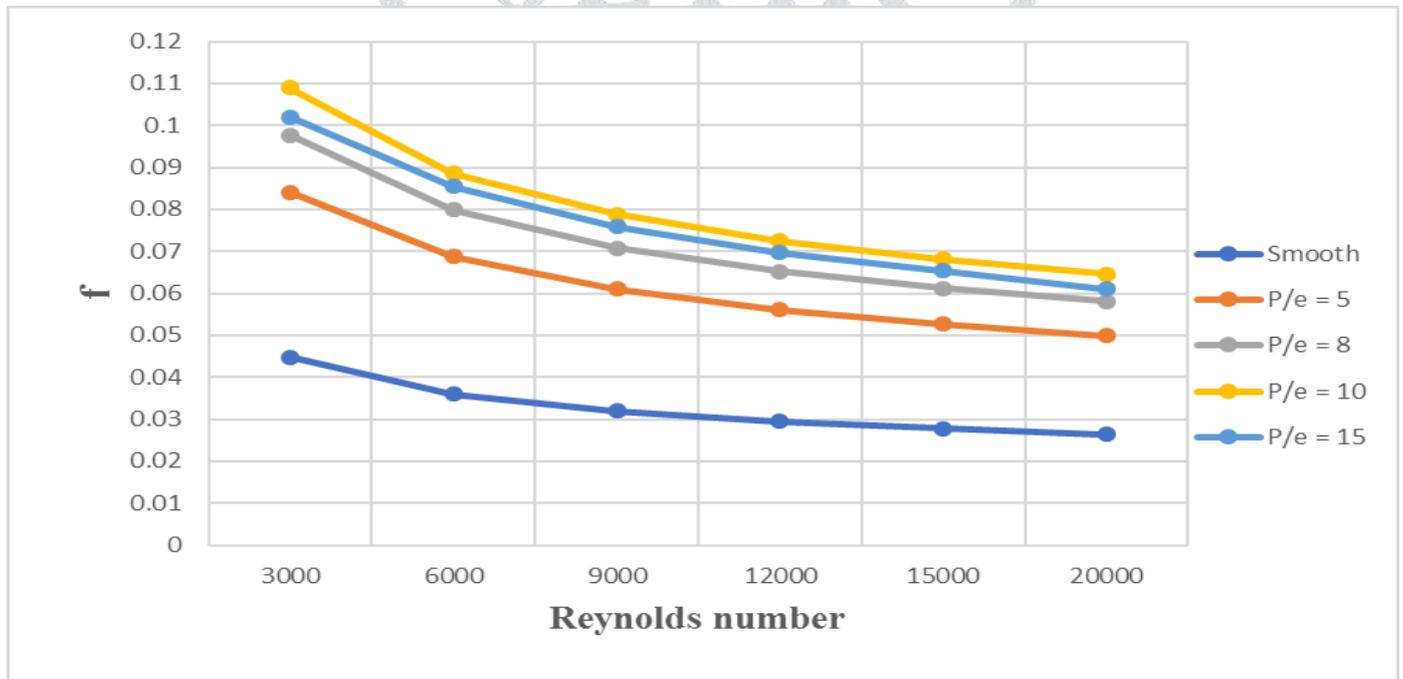


Figure. 5 Variation of Friction factor and Reynolds number for Smooth and Rough pipe

The Nusselt number ratio, Nu_r/Nu_s plotted against the Reynolds number is illustrated in Figure 6. From the figure, we see that Nusselt number ratio, is highest for $(P/e) = 10$ and then after pitch to rib height ratio 15,8 and the smallest among all is $(P/e) = 5$. The pitch to rib height ratio at 10 gives the highest Nusselt number ratio of about 3.127 and lowest nusselt number ratio is for pitch to rib height ratio 5 is 1.456.

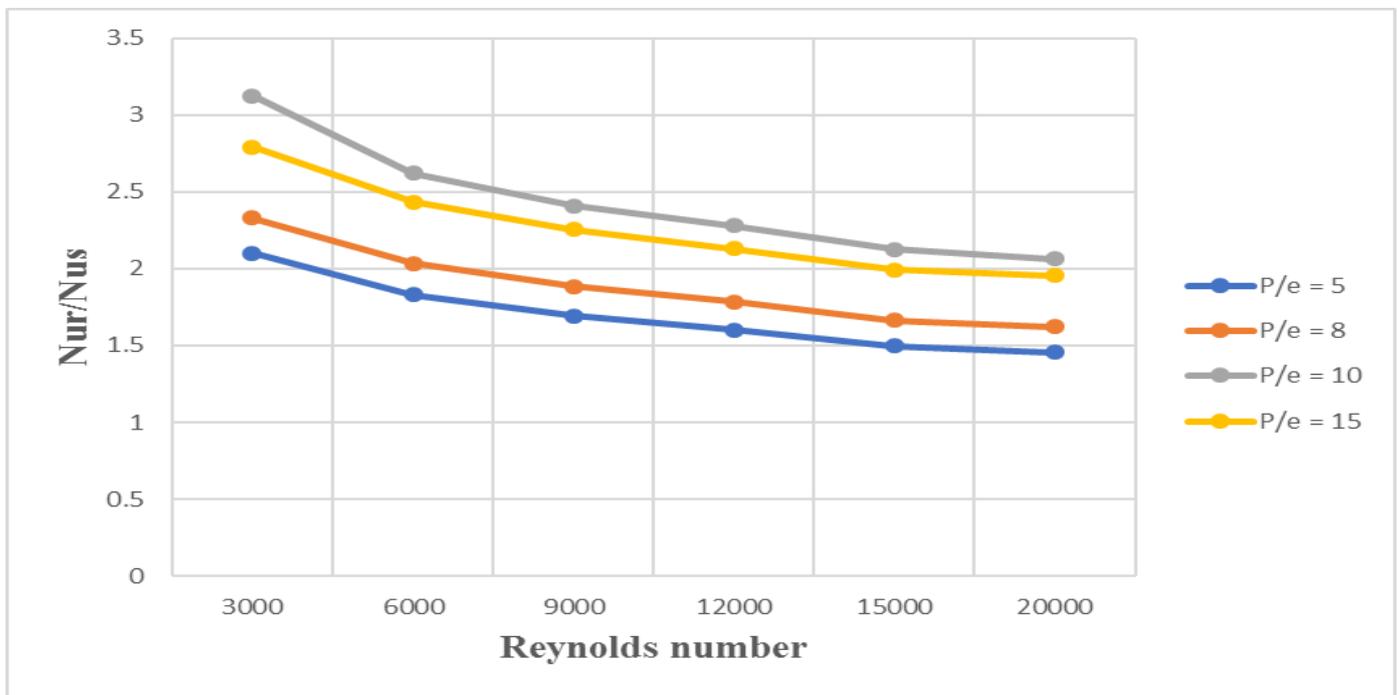


Figure. 6 variation of Nusselt number enhancement ratio with the Reynolds number for Roughened pipe with different (P/e) values

The insignificant variation of friction factor ratio with the Reynolds number can be seen in Fig.7. The maximum and minimum augmentations in friction factor are observed for pitch to rib-height ratio (P/e) of 10 and 5 respectively. It is believed that heat transfer enhancements are accompanied with the rise in frictional losses when turbulence is created in the boundary layer region. We see that Friction factor ratio, is highest for (P/e) = 10 and then after pitch to rib height ratio 15, 8 and the smallest among all is (P/e) = 5. The pitch to rib height ratio at 10 gives the highest Nusselt number ratio of about 2.593 and lowest nusselt number ratio is for pitch to rib height ratio 5 is 1.842.

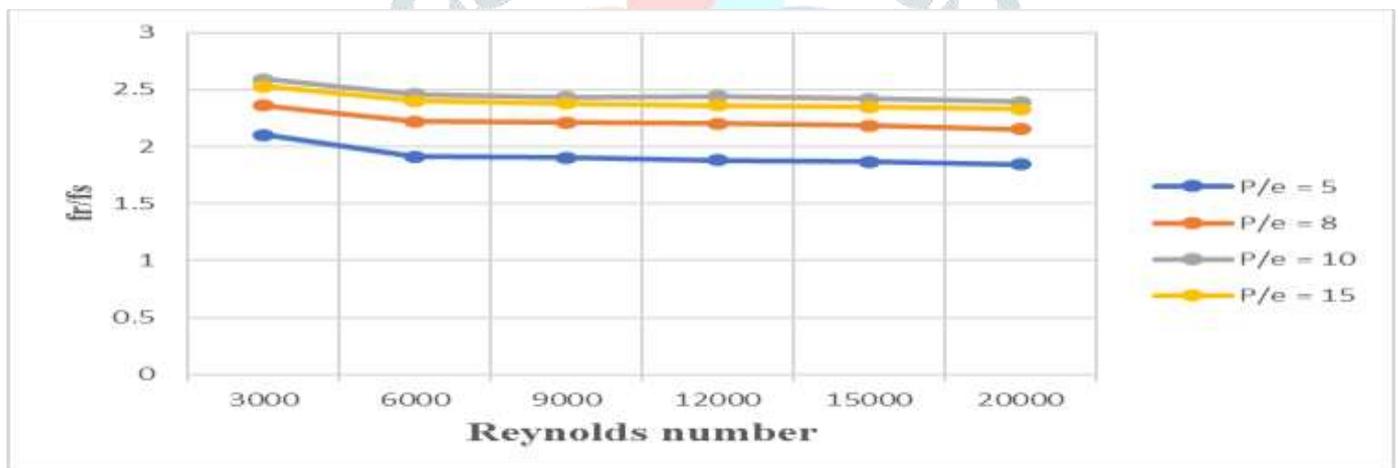


Figure. 7 Variation of Friction factor enhancement ratio with the Reynold number for roughened pipe with different (P/e) values

The thermo-hydraulic performance factor (h) is determined under different system and operating conditions. Fig.8 shows the plot of thermo-hydraulic performance parameter (h) as function of Reynolds number for different pitch to rib-height ratio (P/e) at 10 roughened tubes. The thermo enhancement performance of roughened remains highest at the minimum value of Reynolds number irrespective of the value of pitch to rib-height ratio (P/e). As Reynolds number increases, the thermo-hydraulic performance depreciates and attains the minimum value corresponding to maximum value of Reynolds number. The value of thermo-Enhancement performance lies in the range of 1.529 to 2.251 corresponding to pitch to rib-height ratio (P/e) of 10 for roughness in the range of Reynolds number considered for the present investigation.

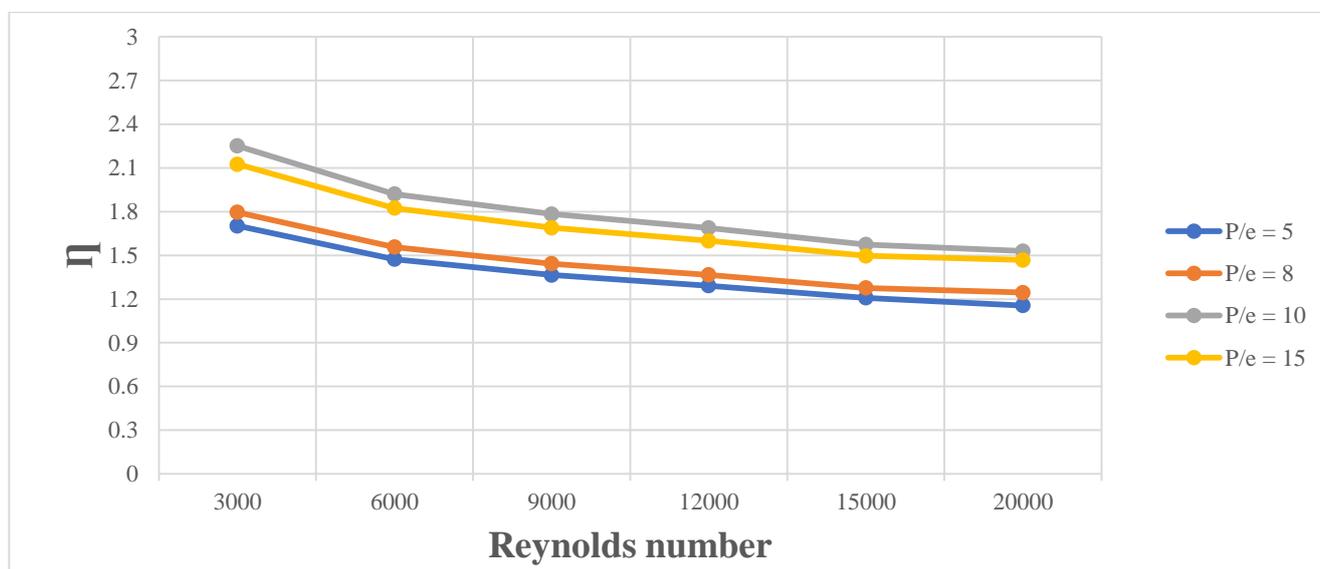


Figure. 8 Variation of Thermal enhancement performance parameter η with Reynolds number for rough pipe with different P/e value

6. Conclusions

The experimental investigation of heat transfer and friction factor characteristics of different pitch roughness with constant increasing pitch ratio sets, as a heat transfer augmentative devices. There are various characteristic of heat enhancement are shown in this experiment. Our experiment was conducted with water as a test fluid and Reynolds number range 3000 to 20000. The following conclusions are drawn from result of this investigation.

- In this experiment the heat transfer enhancement, friction factor and thermal performance factor behaviors in parallel flow heat exchanger with different pitch ratios are to be examined.
- The maximum enhancement in Nusselt number is found to be 3.127 and that of friction factor is obtained to be 2.593 corresponding to (P/e) of 10.
- The maximum value of Thermal hydraulic performance for (P/e) 10 is varies from 1.529 to 2.251.
- With increasing in Reynolds number, friction factor decreases and pressure drop is to be increases with Reynolds number.
- Value of Nusselt and friction factor enhancement ratio increases from (P/e) 5-10 and then decreases.

7. References

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