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Effect of Heat Transfer in a Circular Tube by Using Different Surface Texture

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Abstract: ***Experimental investigation has been carried out to study the effect of thermal enhancement factor,heat transfer and friction factor in a smooth tube and two different test tube having internal threads of pitch (p=1.78mm, p=3.57mm), with water as a working fluid. For experimentation purpose Reynolds number were varied in the range of 4000 to 9000. The copper tube (OD=38mm, ID=28mm, t=5mm) was subjected to constant and uniform heat flux. The experimental data obtained from test tube having different surface texture i.e. by internal threading of different pitches (p=1.78mm, p=3.57mm) were compared with smooth circular tube. The effect of different surface texture of inside tube i.e. by threading with varying pitch on thermal enhancement factor,heat transfer and friction factorwere presented. The heat transfer rate for tubes having internal threads was found to be much higher than smooth circular tube for a given Reynolds number. So simply by changing the internal surface texture the performance of circular tube improved.***

Keywords: Augmentation, Heat exchanger tube, internal threads, Passive heat transfer.

1. Introduction

There are numerous techniques to embellish the heat transfer, such as fins, dimples, additives, etc. A great deal of research effort has been devoted to developing apparatus and performing experiments to define the conditions under which an enhancement technique will improve heat transfer. Heat transfer enhancement technology has been widely applied to heat exchanger applications in refrigeration, automobile, process industries etc. The aim of enhanced heat transfer is to encourage or accommodate high heat fluxes. This result in reduction of heat exchanger size, which generally leads to lower capital cost. So there is need to increase the thermal performance of heat exchangers, thereby effecting energy, material and cost savings have led to development & use of many techniques termed as Heat transfer Augmentation. These techniques are also called as Intensification or Heat Transfer Enhancement. Augmentation techniques increase convective heat transfer by reducing the thermal resistance in a heat exchanger. Use of Heat transfer augmentation techniques lead to increase in heat transfer coefficient but at the penalty of increase in pressure drop. So, while designing a heat exchanger using any of these methods, analysis of heat transfer rate and pressure drop has to be done. Apart from this, issues like long-term performance and detailed economic scrutiny of heat exchanger has to be studied. To achieve maximum high heat transfer rate in an existing or new heat exchanger while taking care of the increased pumping power, several methods have been proposed in recent years.

Generally, heat transfer augmentation techniques are classified in three broad categories. Active method, Passive method, Compound method; In active method some external power input are required for the enhancement of heat transfer for example surface vibration, fluid vibration, suction or injection etc. whereas passive method generally uses surface or geometrical modifications to the flow channel by incorporating inserts or additional devices for example inserts extra components, rough surfaces, additives for fluids etc. The passive methods are based on this principle, by employing several techniques to generate the swirl in the bulk of the fluids and disturb the actual boundary layer so as to increase effective surface area, residence time and heat transfer coefficient in existing system. Lastly compound method is a combination of both the active and passive methods; the compound method involves complex design and has limited applications.

2. Literature Review

Number of investigations has been carried out by using various inserts and tube geometries for heat transfer augmentation. Investigations of various researchers and their findings are as follows.

Naphon [1] experimentally investigated the effect of coil-wire insert on heat transfer enhancement and pressure drop of the horizontal concentric tube has been studied. It was observed that as the Reynolds number increases heat transfer rate decreases. Further Naphon et al. [2] in 2011 changes the geometry of insert, it was found that the swirl flow is generated as fluid flowing through the plain tube with twisted wire brush insert the presence of swirl flow the convective heat transfer obtained is higher. Eiamsa-ard et al. [3] experimentally investigated the heat transfer characteristics in a tube fitted with helical screw-tape with/without core-rod inserts. It was observed that the heat transfer rate obtained by using the tape without core-rod is found to be better than that by one with core-rod around 25–60% while the friction is around 50% lower. Eiamsa-ard et al. [4] in 2008 used double pipe heat exchanger with different arrangement of louvered strip inserts, experimental results showed that the forward louvered strip arrangements can promote the heat transfer rate by approximately 150% to 284%, while the backward arrangements could improve the heat transfer by approximately 133% to 264%. It was observed that Louvered strip insertions can be used efficiently to augment heat transfer rate because the turbulence intensity induced could enhance the heat transfer. Further Eiamsa-ard et al. [5] in 2010 changed the geometry of insert in a heat exchanger tube and observed that the presence of novel alternate C–CC twisted-tapes, the periodic change of swirl direction and also the strong collision of the recombined streams behind the changing location, lead to superior chaotic mixing and to better heat transfer, compared with the typical twisted-tape.Murugesan et al. [6] experimentally investigated the heat transfer and pressure drop characteristics in a circular tube fitted with and without V-cut twisted tape insert. It was observed that the V-cut twisted tape offered a higher heat transfer rate, friction factor and also thermal performance factor compared to the plain twisted tape. The results obtained were same as that of [5].Thianponga et al. [7] experimentally investigated the effect of perforated twisted-tapes with parallel wings on heat transfer enhancement in a heat exchanger tube. The result showed that as compared to plain twisted tube, PTT increases heat transfer rate up to 208%, and compared with [5,6] showed good result. Fan et al. [8] numerically and computationally studied the heat transfer and flow characteristics in a circular tube fitted with louvered strip inserts. It was observed that the Nusslet number is augmented by around 4 times that of the smooth tube also compared with [5] it showed better result; hence it confirms that the louvered strip has a good effect of heat transfer enhancement. Promvonge et al. [9] experimentally investigated the heat transfer augmentation in a helical-ribbed tube with double twisted tape inserts. It was found that the compound enhancement devices of helical-ribbed tube and the twin twisted tapes show a considerable improvement of heat transfer rate and thermal performance relative to the smooth tube and the helical-ribbed tube acting alone. Mohammed et al. [10] numerically and computationally studied the heat transfer enhancement of Nano fluids in a double pipe heat exchanger with louvered strip insert. The result obtained by [8] was same that is the Nusslet number is augmented by around 4 times for the louvered strip insert than that of the smooth tube. Bhuiya et al. [11] experimentally investigated the thermal characteristics in a heat exchanger tube fitted with triple twisted tape inserts. It was found that the Nusslet number was obtained 1.73 to 3.85 times higher than those of the plain tube values for the tube with triple twisted tape inserts. The friction factor for the tube with triple twisted tape inserts was achieved 1.91 to 4.2 times higher than those of the plain tube values at the comparable Reynolds number.

Zhang et al. [12] numerically and computationally studied on heat transfer and friction factor characteristics of a tube fitted with helical screw-tape without core-rod insert. The results obtained were same as that obtained by [3] that is heat transfer rate increase around 25-60% while friction is around 50% lower. TuWenbin et al. [13] experimental studied on heat transfer and friction factor characteristics of turbulent flow through a circular tube with small pipe inserts. It was observed that the tube fitted with small pipe inserts can achieve a high heat transfer with a lower increase in friction factor. The friction factor is less than most of those recorded in other literature and is only 1.24–1.87 times that of the smooth tube. Esmaeilzadeh et al. [14] experimental studied on heat transfer and friction factor characteristics of g-Al_2O_3/water through circular tube with twisted tape inserts with different thicknesses. It was observed that the highest enhancement is achieved at maximum thickness and volume concentration. Nano fluids have better heat transfer performance when utilized with thicker twisted tapes. More the thickness of twisted tape is more the increase of friction factor is. Ultimately, the convective heat transfer enhancement outweighs the effect of friction factor increase, leading enhanced thermal performance.Shrirao et al. [15] experimentally carried out heat transfer analysis in a circular tube with different types of internal threads of constant pitch; experimental results showed that heat transfer along with the friction factor increases more in case of tube having buttress threads compared to knuckle and acme threads in a tube.

3. Experimentation

3.1 Experimental Set-up description

Figure 1 shows photograph of experimental set-up. Test section consist of copper tube (O.D=38mm, I.D=28mm, t=5mm) of length 800mm. Six k type thermocouples were soldered at six equally spaced point which were separated by 134 mm distance and two thermocouples were placed at inlet and outlet stream to measure stream inlet and outlet temperature. The copper tube was wrapped by mica heaters of 1500 Watts capacity in order to maintain constant heat flux. The mica heater wrapped on test section was surrounded by glass wool insulation and after that steel cover was placed. The required heat input was given through Dimmerstat. A U-tube manometer was used to measure the pressure drop across the tube; water was used as a manometric fluid. The distance between two pressure tapping was 900mm. Same procedure was repeated for two test section having internal threading of pitch (p=1.78mm, p=3.57mm). A British Standard Whitworth (B.S.W) thread was used for threading throughout the length of copper tube i.e. L=800mm.

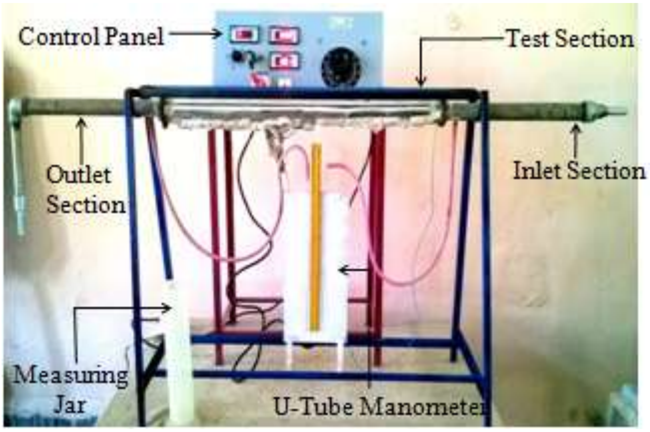

Figure 1: Photograph of experimental set-up

3.2 Experimental Procedure

Inlet section of set up is connected to the gate valve of water tank which takes water and pumped through test section. The flow rate of water is controlled by gate valve and was measured manually using stop watch and measuring jar. The flow rate varied using gate valve for different values of Reynolds number and kept constant during experimentation. After switching on the heater power the sufficient time was given to attain the steady state condition. In each run data were taken for water flow rate, water inlet, water outlet and tube outer surface temperature and pressure drop readings.

- Open the supply valve and adjust the flow by means of gate valve and to some desired difference in the manometer level.
- Start the heating of test section with the help of dimmerstat and adjust desired heat input with the help of voltmeter and ammeter.
- Take readings of thermocouples at an interval of 10 minutes, until steady state is reached.
- Wait for steady state and take reading of all thermocouples at steady state.
- Note down heater input.
- Above Same procedure was repeated for remaining two test section having internal threads of pitches (p=1.78mm, p=3.57mm).

4. Data Reduction

Heat transfer rate by heater to water was calculated by measuring heat added to the water.
Heat added to water was calculated by,

$$Q = mc_p(T_{out} - T_{in}) \tag{1}$$

Heat transfer coefficient was calculated from,

$$h = \frac{q}{(T_{wi} - T_b)} \tag{2}$$

And heat flux was obtained from,

$$q = \frac{Q}{A} \tag{3}$$

Where,

$$A = \pi d_i L \tag{4}$$

The bulk temperature was obtained from the average of water inlet and outlet temperatures,

$$T_b = \frac{(T_{in} + T_{out})}{2} \tag{5}$$

Tube inner surface temperature was calculated from one dimensional radial conduction equation,

$$T_{wi} = T_{wo} - Q\frac{ln\left(\frac{d_o}{d_i}\right)}{2\pi k_w L} \tag{6}$$

Tube outer surface temperature was calculated from the average of six local tube outer surface temperature,

$$T_{wo} = \sum_{i=1}^{6} T_{wo,i}/6 \tag{7}$$

Experimentally Nusselt number was calculated from,

$$Nu = \frac{hd_i}{k} \tag{8}$$

Theoretical Nusselt number was calculated from Gnielinski, 1976, correlation,

$$Nu_{th} = \frac{\left(\frac{f}{8}\right)(Re - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \tag{9}$$

Where from Petukhov, 1970,

$$f_{th} = (0.790 lnRe - 1.64)^{-2} \tag{10}$$

$$Re = \frac{\rho Umdi}{\mu} \tag{11}$$

$$Pr = \frac{\mu Cp}{k} \tag{12}$$

Mean water velocity was obtained from,

$$U_m = \frac{m}{A_f} \tag{13}$$

Flow area was obtained from,

$$A_f = \frac{\pi}{4} di^2 \tag{14}$$

Experimentally Friction factor, f can be calculated from,

$$f = \frac{\Delta p}{(L/d_i)(\rho U_m{}^2/2)} \tag{15}$$

5. Results and Discussion

Figure 1 shows the comparison between Nusselt number obtained experimentally, and theoretically by using Genielinski correlation for smooth tube. It was observed that the value of Nu (Experimental) is less than that of Nu (Theoretical). As the heat is transferred through convection mode, so while performing experimental and numerical calculations, it can be expected that Nu (Experimental) is less than that of Nu (Theoretical).

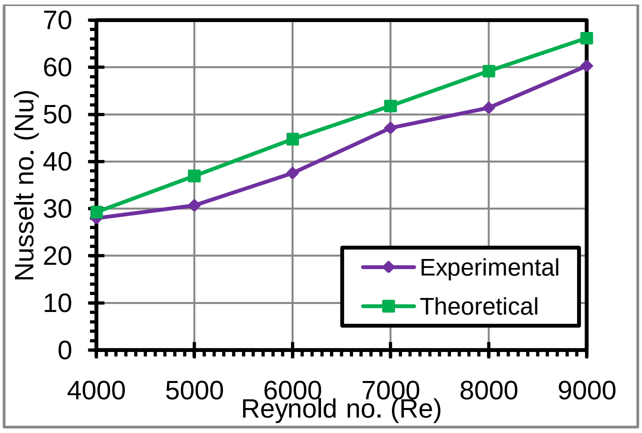

Figure 1: Comparison between Nusselt number obtained experimentally, theoretically by using Gnielinski correlation for smooth tube.

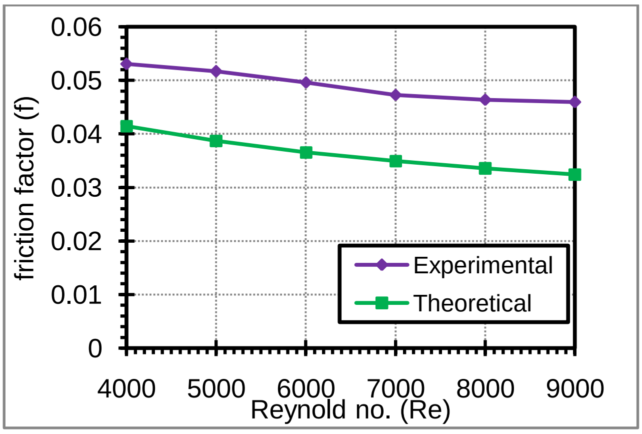

Figure 2: Comparison between friction factors obtained experimentally, theoretically by using Petukhov correlation for smooth tube.

Figure 2: shows the variation of friction factor with Reynolds number for smooth tube. From the graph obtained it is cleared that f (Experimental) is greater than that of f (Theoretical).

Figure 3 shows the variation of Nusselt number with Reynolds number in smooth tube and two different test tube having internal threading of pitch (p=1.78mm, p=3.57mm). It was observed that for all cases, Nusselt number increases with increasing Reynolds number. Also it was observed that for tube with internal threads the heat transfer rate was higher than those for smooth tube. From the graph it was cleared that the test tube having internal threads of pitch (p=3.57mm) the heat transfer rate is much higher than those from tube having internal threads of pitch (p=1.78mm), smooth tube and values obtained from theoretical correlation (Gnielinski). This is due to the fact that depth provided by test tube having internal threads of pitch (p=3.57mm) is more as compared to test tube having internal threads of pitch (p=1.78mm) and smooth tube, which increases the strength and turbulent intensity of water across the range of Reynolds number.

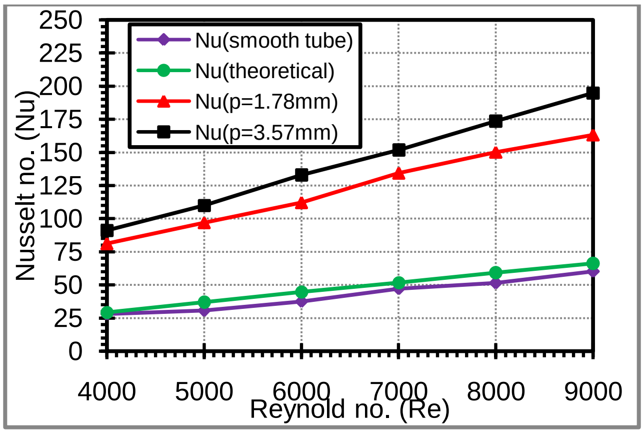

Figure 3: Variation of Nusselt number with Reynolds number

Figure 4 shows the variation of friction factor with Reynolds number for smooth tube, values obtained from theoretical correlation (Petukhov) and test tubes having internal threads of pitch (p=1.78mm, p=3.57mm). The friction factor for the test tube having internal threads is more than that of smooth tube. From the graph it was cleared that as the Reynolds number increases there is decrease in friction factor so we conclude that as velocity goes on increasing, friction factor is inversely proportional to the velocity. This shows that the turbulence formation advanced due to artificial turbulence exerted by internal threads. The friction factor of test tube having internal threads of pitch (p=3.57mm) is much higher than that of test tube having internal threads of pitch (1.78mm) and smooth tube. This due to fact that depth provided by test tube having pitch (p=3.75mm) is more as compared to test tube having pitch (p=1.78mm), which increases the swirl flow of water across the range of Reynolds number.Figure 5 shows the variation of thermal enhancement factor with Reynolds number. At the same Reynolds number the thermal enhancement factor for test tube having internal threads of pitch (p=3.57mm) was found to be greater than those for test tube having internal threads of pitch (p=1.78mm). The thermal enhancement factor for all internal threads tends to decrease with increasing Reynolds number.

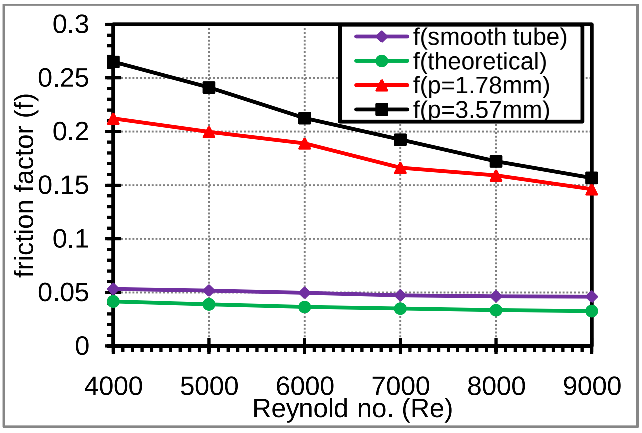

Figure 4: Variation of friction factor with Reynolds number

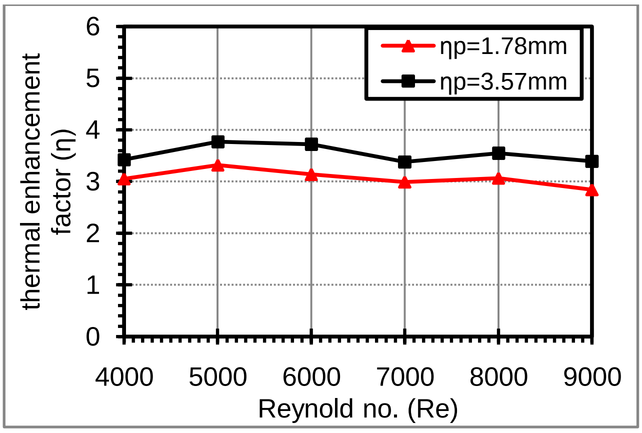

Figure 5: Variation of thermal enhancement with Reynolds number

6. Conclusions

An Experimental investigation have been carried out to study the effects of internal threads of different pitches (p=1.78mm, p=3.57mm) and smooth tube. Heat transfer coefficient, friction factor and thermal enhancement factor are analyzed by using passive heat transfer augmentation method. From the graph plotted in the above chapters following conclusions are made:

- The heat transfer rate increases in a test tube having internal threads as compared to smooth tube. The result shows that the heat transfer rate increases with increasing in Reynolds number.
- The test tube having internal threads of pitch (p=3.57mm) causes the maximum heat transfer rate. This is due to the fact that depth provided by test tube having internal threads of pitch (p=3.57mm) is more as compared to test tube having internal threads of pitch (p=1.78mm) and smooth tube.

- The friction factor increases for a test tube having internal threads of pitch (p=3.57mm) than those for test tube having internal threads of pitch (p=1.78mm) and smooth tube, due to more swirl flow exerted by a test tube having internal threads of pitch (p=3.57mm).
- As the pitch of internal threads increases it is found that there is increase in heat transfer rate but increase in friction factor is observed. So it can be concluded that maximum the pitch of internal threads, maximum will be the heat transfer rate but more friction losses will also be occurs.
- The enhancement of Nusselt number is much higher than that of enhancement in friction factor a for test tube having internal threads of pitch (p=1.78mm) and smooth tube that justify the usage of internal threads in a circular tube.
- The performance of circular tube can be improved by the use of internal threads. The cost involved for making internal threads is minimal compared to energy efficiency improvement provided by this technique.

Nomenclature

Symbol	*Meaning*	*Unit*
A	Area of the heated region of tube	(m^2)
A_f	Flow area	(m^2)
C_p	Specific heat of water at constant pressure	(J/kg.K)
d_i	Tube inner diameter	(m)
d_o	Tube outer diameter	(m)
t	Thickness of copper tube	(m)
p	Pitch of threads inside copper tube	(m)
h	Heat transfer coefficient	(W/m.K)
k	Thermal conductivity of water	(W/m.K)
k_w	Thermal conductivity of tube material	(W/m.K)
L	Effective tube length	(m)
m	Mass flow rate of water	(Kg/s)
Q	Heat transfer rate	(W)
q	Heat flux	(W/m^2)
T_b	Bulk temperature	(℃)
T_{in}	Water inlet temperature	(℃)
T_{out}	Water outlet temperature	(℃)
T_{wi}	Tube inner surface temperature	(℃)
T_{wo}	Tube outer surface temperature	(℃)
$T_{wo,i}$	Local tube outer surface temperature	(℃)
U_m	Mean velocity	(m/s)
N_u	Experimental Nusselt number	(-)
Nu_{th}	Nusselt number from Gnielinski correlation	(-)
f	Experimental friction factor	(-)
f_{th}	Friction factor calculated from Petukhov correlation	(-)
Δ_p	Drop in pressure	(N/m^2)
g	Acceleration due to gravity	(m/s^2)
Pr	Prandtl number	(-)
Re	Reynolds number	(-)
ρ	Density of water	(kg/m^3)
μ	Dynamic viscosity of water	(kg/m.s)

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