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TURBULENT REGIME FRICTION FACTORS FOR U-TYPE WAVY TUBES

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ABSTRACT

Pressure drop in both straight section and bend section for water flowing in small diameter tubes having U-type wavy configuration are measured. The curvature ratio (*2R/D*) spans from 6.43 to 13.03 and the dimensionless spacer length (*L/D*) spans from 3.16 to 6.41. The test is done in the turbulent regime. The bend friction factor is found to decrease with increase in curvature ratio (*2R/D*) and dimensionless spacer length (*L/D*). The Fanning friction factor in the straight section is found to match closely with the well-known Blasius equation for the turbulent regime. In this paper, a simple correlation for turbulent regime friction factors for U-type wavy tubes is developed based on the experimental data considering the effect of Dean number (D_n), dimensionless spacer length (*L/D*) and number of bends (*n*). Predicted friction factors are in good agreement with the experimental data with a mean standard deviation of only 4%.

Keywords: Friction factor, U-Type wavy tubes, turbulent regime.

1. INTRODUCTION

U-Type wavy tubes are utilized in a great number of heat transfer equipment and flow transmitting devices. It provides vigorous mixing of fluid provided by the alternating bends. At present, the wavy tubes are used in shell and tube heat exchangers for domestic and industrial water heating systems and in plate solar collectors. Helically coiled tubes are largely used for heat exchangers, such as steam generators and superheaters in nuclear power plants and continuous chemical reactors in chemical industries. The use of small diameter tubes in HVAC&R (Heating, Ventilation, Air-conditioning and Refrigeration) appliances is very popular, because of less refrigeration storage, better air side heat transfer and smaller air side drag. The U-type wavy tubes in aircooled heat exchangers usually have a straight section (spacer) between two consecutive 180^o bends.

Tae and Cho [1] reported that the condensation and evaporation heat transfer of R-22 and R-407C refrigerants in a straight section downstream a micro fin tube with a U-bend is 33% higher than that of a straight section upstream the U-bend. It is attributed to the disturbance caused by the secondary flow in the U-bend carried into the downstream straight section after the U-bend. The frictional resistance and, hence, the pressure drop associated with the induced turbulence of the secondary flow also increases. The increased pressure drop in the consecutive U-type wavy tubes may significantly affect the refrigeration distribution in the refrigerant flow system. For this reason, the frictional performance of U-type wavy tubes with consecutive 180^o return bends is very important for the design of heat exchangers.

The first theoretical study of this phenomenon was made by Dean in 1927. He pointed out that the dynamic similarity of the flow depends on a non-dimensional parameter, D_n called the "Dean Number".

$$D_n = \frac{2rV}{\nu}\sqrt{\frac{r}{R}} = \mathrm{Re}\sqrt{\frac{r}{R}} \tag{1}$$

Where '*Re*' is the Reynolds number, '*V*' is the mean velocity along the tube, '*ν*' is the coefficient of kinematic viscosity, and '*2r*' is the diameter of the tube which is bent into an arc of radius '*R*'. The ratio of bend friction factor to straight friction factor depends only on 'D_n', as long as the motion is laminar. But that is no longer the case when turbulence sets in. At a higher Dean Number a centrifugal instability near the concave outer wall of the tube generates an additional pair of vortices called "*Dean Vortices*".

When a fluid flows through a stationary curved tube, a pressure gradient across the tube is required to balance the centrifugal force arising from the curvature. The pressure in the portion of the wall further from the center of curvature of the tube is greater than that in the portion which is nearer to it. If the tube is considered to be lying on a horizontal plane, the fluid near the top and the bottom is moving more slowly than the fluid in the central portion and requires a smaller pressure gradient to balance the centrifugal force. In consequence, a secondary flow is set up in which the fluid near the top and the bottom moves inward and the fluid in the middle moves outward. The secondary flow is superposed on the main stream, so that the resultant flow is helical at the top and bottom halves of the tube. As a result of the secondary flow, the region of maximum velocity in the main stream is shifted towards the outer part of the wall, the total frictional loss of energy near the wall of the tube increases, and the flow experiences more resistance in passing through the tube. This increase in resistance is small when the curvature of the tube is small. If the curvature is significant the axial velocity distribution is entirely altered by the secondary flow and a considerable resistance is observed. [2]

Flow in a wavy tube approaching a consecutive bend is much more complex. Pressure losses in the flow are a result of the very complex velocity gradient distribution and friction at the tube wall and of the dissipation of energy of the vortex pairs produced by each consecutive bend. Popiel and Wojtkowiak [3] studied this type of flow in the laminar regime and proposed the following correlation for Darcy friction factor, f_D for bend.

$$\ln\left(\frac{f_D \mathrm{Re}}{64}\right) = 0.021796 + 0.0413356(\ln D_n)^2 \tag{2}$$

Later, Wojtkowiak studied the effects of the length of the straight section or the spacer (L/D) between the two consecutive 180° bends and proposed a new correlation as given below.

$$f_D = a + b \ln D_n + \frac{c}{D_n} \tag{3}$$

Where $a = 0.121433 - 0.00182313(L/D)$, $b = -0.010311 + 0.0001936\,(L/D)$ and $c = 16.68855 - 0.16757(L/D)$. However, Chen, Lai and Wang [4] reported that the two correlations mentioned above do not match with some new experimental data. They proposed a better correlation shown as equation (4), which is able to predict their own experimental data as well as the data of Popiel and Wojtkowiak [3] for Fanning friction factor, f_B for bend.

All the above mentioned studies of flows through U-type wavy tubes are in the range of $Re = 200 \sim 18000$, which covers mostly laminar and transition regimes of flow. None of them reported the friction factors in the case of fully turbulent flow as well as the effects of number of bends. Therefore, the purpose of the study is to investigate the effect of number of bends (n), curvature ratio ($2R/D$) and dimensionless spacer length (L/D) on single-phase turbulent flow inside U-type wavy tubes.

2. EXPERIMENTAL METHOD: TEST FACILITY

The schematic diagram of the setup is shown in Fig.1. The experimental setup consists of a water tank, a centrifugal pump, a gate-valve V-1 for delivery control, the test section, a U-tube mercury manometer and a measuring flask to measure the water flow. The test sections were made of copper tubes used for HVAC and Refrigeration purposes. They were carefully checked to ensure that they have uniform diameter along the length of the test section.

The test was carried out in two stages and two sets of test sections were prepared. To investigate the effect of curvature ratio ($2R/D$) and dimensionless spacer length (L/D), three different tube diameters (D) are chosen while the number of bends (n = 15), bend radius (R = 25.4 mm) and spacer length (L = 25 mm) are kept constant as shown in Table 1.

Table 1: Test section details for effects of *2R/D* and *L/D*

Tube diameter *D*, mm	**Curvature Ratio 2R/D**	**Dimensionless Spacer length L/D**
3.9 ± 0.01	13.026	6.41
4.85 ± 0.01	10.47	5.15
7.9 ± 0.01	6.43	3.16

Number of bends, n = 15, bend radius, R = 25.4 mm and spacer length, L = 25 mm.

To investigate the effect of number of bends, four different bend numbers (n = 9, 11, 13 and 15) are tested, while keeping other parameters constant, namely, tube diameter, D = 7.9 ± 0.01 mm, bend radius, R = 25.4 mm, spacer length, L = 25 mm, curvature ratio, $2R/D$ = 6.43 and dimensionless spacer length, L/D = 3.16.

In each of the test sections, the entry length is kept greater than $100D$ to ensure a fully developed flow before it reaches the first pressure tapping, P-1. At $100D$ downstream of the pressure tapping P-1 the second pressure tapping P-2 is provided. The fanning friction factor for the straight section is determined from the pressure drop between P-1 and P-2. Downstream of the bends the distorted flow condition persists for some distance beyond which the pressure gradient becomes steady and presumably the same as that of the upstream [5]. For this reason, the third pressure tapping, P-3 is located at a distance of about 130D downstream of the bend. The pressure drop between P-1 and P-3 is composed of the pressure drops both in the straight and bent sections.

3. BEND FRICTION FACTOR, f_B

The pressure drop due to a single bend is calculated by subtracting the straight section pressure drop from the measured total pressure drop. Then the equivalent bend friction factor, f_B is defined as:

$$f_B = \frac{\Delta P - 2 f \rho V^2 \frac{L_t}{D}}{2 \rho V^2 \frac{L_c}{D}} \tag{5}$$

Here, f is the Fanning friction factor for the straight

$$f_B = \frac{16}{\mathrm{Re}}\left[1 + 29 e^{\left(-\frac{2R}{D}\right)}\right] \times \exp\left\{0.07 + 0.04 \ln(Dn)^2 + \left(\frac{L}{D}\right)\left[0.36 - 0.035 \ln(\mathrm{Re})^{0.9} - 0.145\left(\frac{L}{D}\right)^{2.5} + 0.005\left(\frac{L}{D}\right)^3\right]\right\} \tag{4}$$

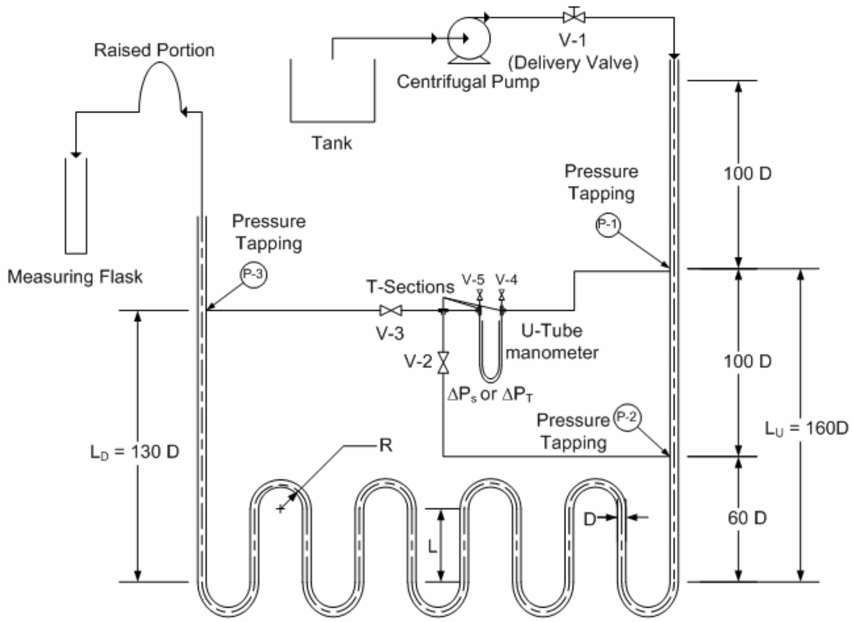

Fig 1: Schematic Diagram of the Setup

section, 'V' is the mean axial velocity and L_s = length of the total straight portion in the test section, i.e. straight portion in the upstream (L_U) + straight portion in spacer (L_L) + straight portion in the downstream (L_D). If 'n' is the number of bends and L is the length of a single spacer then, $L_L = (n-1)L$ and $L_c = n\pi R$.

An uncertainty analysis has been carried out as described by Moffat [6]. The maximum overall uncertainty in the measurement of bend friction factor f_B is ±7% for $Re \approx 20000$ and ±4.3% for $Re \approx 2000$.

4. RESULTS AND DISCUSSION

In Fig-2, the experimentally determined values of bend friction factor, f_B and Fanning friction factor for straight sections, f are plotted against Reynolds number, Re. Three different diameters, namely, D = 3.9, 4.85 and 7.9 mm were used for the plot of f. In case of the bent sections, the number of bends was 15 for all cases whereas the values of $2R/D$ were 13.03, 10.47 and 6.43 and the values of L/D were 6.41, 5.15 and 3.16. The solid line is the plot of the well-known Blasius equation given below for turbulent flow through a hydrodynamically smooth tube.

$$f = \frac{0.079}{Re^{0.25}} \tag{6}$$

It is evident that the experimentally Fanning friction factor, f for the straight sections agrees favorably with the Blasius line. This good agreement illustrates the accuracy of the experimental setup and procedure. The graph shows that as $2R/D$ and L/D decrease, f_B increases. As the curvature ratio ($2R/D$) decreases, the centrifugal force increases. For this reason the streamline of the maximum flow velocity is shifted towards the outer wall and the bend friction factor, f_B increases. As the dimensionless spacer length (L/D) increases, the increase in spacer length relaxes the strength of the vortical motion, thus the flow becomes more uniform downstream the bends and as a result friction factor f_B becomes smaller.

The transition from laminar to turbulent region of f_B is relatively smooth, whereas an abrupt change is seen in the slope of f at the transition of laminar to turbulent flow in straight sections of the tubes. From the smooth laminar to turbulent transition, it is concluded that in the elongated transition region the laminar secondary flow and the developing additional dean-vortices are gradually replaced by the turbulent secondary flow.

It is seen in Fig. 3 that for same curvature ratio ($2R/D$) and same dimensionless spacer length (L/D), f_B increases slightly with the increase of number of bends (n). The ratio of f_B/f for 15 number of bends are shown in Fig. 4

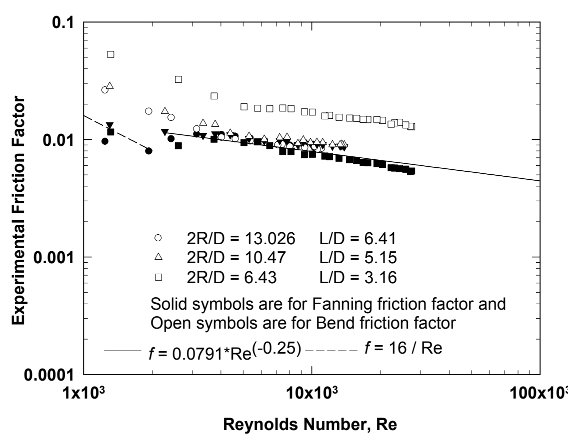

Fig 2: Fanning Friction factor for straight tubes and Bend Friction factor for wavy tubes against Re.

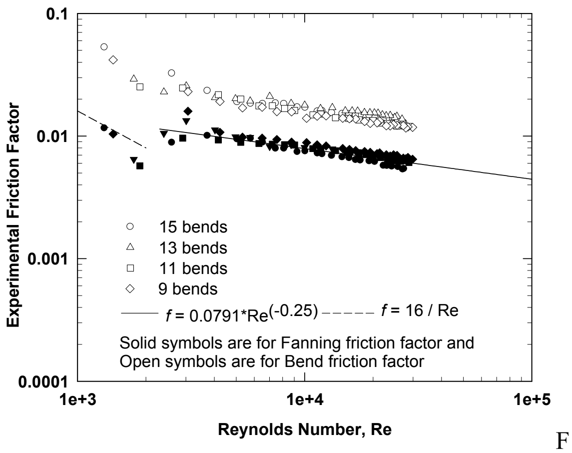

Fig 3: Fanning Friction factor for straight tubes and Bend friction factor for wavy tubes against Re for D = 7.9 mm, 2R/D = 6.43, L/D = 3.16

for the tube diameters of 3.9 mm, 4.85 mm and 7.9 mm. The ratio of f_B/f considerably increases with decrease of curvature ratio. This increase is mostly caused by the secondary flow, which increases the disturbance in flow in the curved tubes. In the transition regime, the ratio of f_B/f for 7.9 mm diameter tube becomes as high as 4.5 and then decreases to a value slightly greater than 2.0 in the turbulent regime. For the 3.9 mm and 4.85 mm diameter tubes, the value of f_B/f can be as high as 3.0 in the transition regime and then decreases to a value which is slightly greater than 1.0 in the turbulent regime. At Reynolds number 4000 and above the f_B/f values are approximately equal to 1.0, which implies that the f_B data are very close to the line of Blasius equation. That is the effect of curvature ratio and dimensionless spacer length is diminished. According to Popiel and Wojtkowiak [8], the effect of laminar secondary flow and the developing Dean vortices on the pressure losses is gradually replaced by the turbulent secondary flow as the Reynolds number is increased. Larger spacer length may relax the strength of vortical motion of the swirled flow passing the return bend. Therefore when *L/D* increases the overall flow in the recovery region becomes more uniform leading to a lower friction factor in the system.

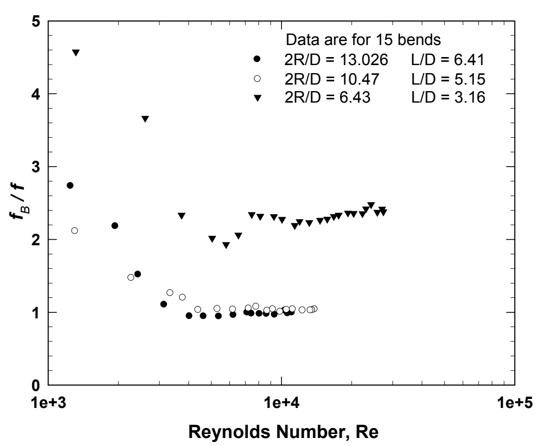

Fig 4: The ratio of f_B/f vs. Re

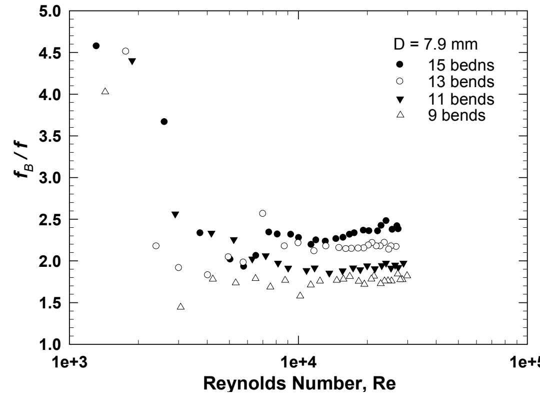

Fig 5: The ratio of f_B/f vs. Re

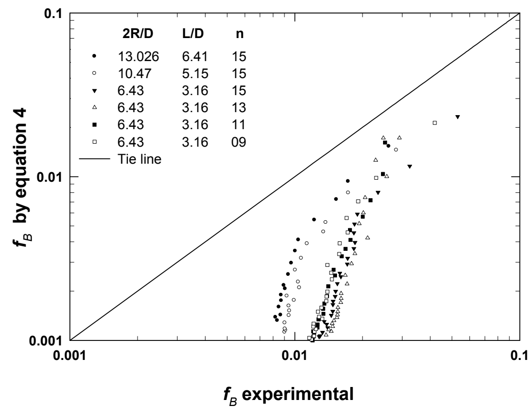

Fig 6: Comparison of the experimental data and the predictions by equation-4.

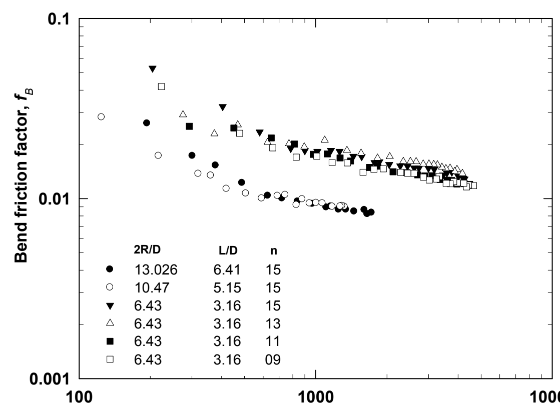

Fig 7: Bend Friction Factor, f_B vs D_n

For the same curvature ratio (*2R/D*) and spacer length (*L/D*) the ratio f_B/f increases with the increase of number of bends as shown in Fig. 5. The prediction of bend friction factor f_B by Eq. 4, and the present experimental data are shown in Fig. 6. The correlation is valid for Reynolds number 300 to 20000 and it is seen that, the present data which are below Reynolds number 20000, match with the trend of the correlation. But the quantitative agreement could not be achieved perhaps because the correlation did not consider the effect of

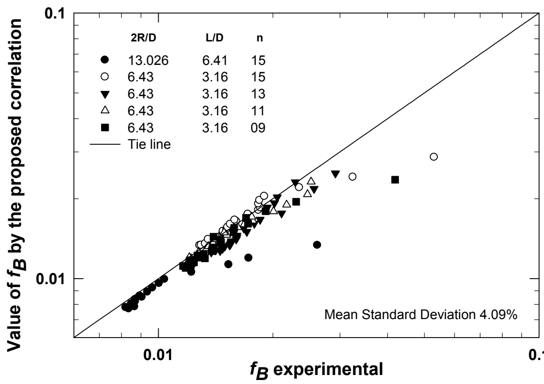

Fig 8: Comparison between the predictions of the proposed correlation and the experimental data

number of bends. In the turbulent regime, the correlation cannot even predict the trend of the present data.

In Fig. 7, bend friction factor, f_B is plotted against Dean number, D_n which shows that for Dean Number > 700, the data maintain a linear relation. Based on this observation, a correlation is proposed below which includes the effect of Dean number (D_n), dimensionless spacer length (L/D) and number of bends (n). The correlation is given below.

$$\log f_B = -0.252\log(D_n) - 1.1\log\left(\frac{L}{D}\right) + 1.3\times10^{-3} n^{1.7} - 0.54 \tag{7}$$

Figure 8 shows that the predictions of f_B by the proposed correlation are in good agreement with the present experimental data. The correlation is valid for

Dean number, D_n: 700-5000
Reynolds number, Re: 3500-30000
Curvature ratio, $2R/D$: 6.43-13.026
Dimensionless spacer length, L/D: 3.16-6.41
Number of bends, n: 9-15

5. CONCLUSIONS

The salient points of this study may be summarized as follows:

a. Ratio of f_B/f increases considerably with the decrease of curvature ratio ($2R/D$) and dimensionless spacer length (L/D).
b. Existing correlations for low Reynolds number cases are unable to predict the present experimental data because the correlations are valid only in the laminar regime and also the effect of number of bends were not considered.
c. The proposed correlation agrees favorably with the present data and considers the effect of Dean number (D_n), dimensionless spacer length (L/D) and number of bends. The correlation is valid in turbulent regime.

6. REFERENCES

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7. NOMENCLATURE

Symbol	Meaning	Unit
f	Fanning friction factor for straight smooth tubes	Dimensionless
f_D	Darcy friction factor	Dimensionless
f_B	Bend friction factor	Dimensionless
D	Diameter of copper tube	mm
R	Radius of bend	mm
L	Spacer length	mm
n	Number of bends	Dimensionless
ΔP_T	Total pressure drop	Pa
Re	Reynolds number	Dimensionless
V	Velocity	m/s
D_n	Dean number	Dimensionless
L_s	Length of straight portion of the test section	mm
L_c	Length of bent portion of the test section	mm