Transitional flow inside enhanced tubes for fully developed and developing flow with different types of inlet disturbances: Part I – Adiabatic pressure drops

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1. Introduction

It is accepted in literature that transition from laminar to turbulent flow inside tubes occurs at a Reynolds number of approximately 2300. Although this is an accepted value, transition in reality occurs in the range of Reynolds numbers between 2300 and 10,000 [1]. It is normally advised when designing heat exchangers to remain outside these limits due to the uncertainty and flow instability of this region. Large pressure variations are also encountered in this region since the pressure gradient required to move the fluid in laminar and turbulent flow could vary by an order of magnitude.

As early as 1883, Reynolds [2] showed that transition occurred at a critical value, which depended on the surrounding disturbances. For a tube, this value is a function of the fluid velocity, tube diameter and viscosity, also now known as the Reynolds number. Reynolds described the onset of turbulence always occurring at considerable distance from the entrance, with the turbulence moving towards the inlet as the velocity was increased. Reynolds also found that just above the critical value, turbulence would occur in flashes at a fixed point down the length of the tube. These flashes are also known as turbulent bursts, and were also visually observed by Lindgren [3]. Lindgren found that the transition occurred in a gradual manner with fluctuating bursts of turbulence.

The frequencies of these bursts were found to be a function of the fluid velocity and the distance from the inlet. It was also shown that the critical Reynolds number increased with an increase in distance from onset, with visual observations confirming this.

Kalinin and Yarkho [4] found with heat transfer experiments that the wall temperatures fluctuate in the transition region. Ede [5] also detected the fluctuations although it was blamed on a technical issue.

Inlet profiles were found to have a profound influence on the transition Reynolds number. Nagendra [6] found that the greater the disturbance, the earlier transition occurs. Ghajar and Madon [7] performed an extensive study into the effect of three different types of inlets on the critical Reynolds number during isothermal fully developed flow. The three inlets tested were a square-edged (sudden contraction), a re-entrant (tube protruding square-edged inlet) and a bellmouth inlet (smooth, gradual contraction). It was found that transition from laminar to turbulent flow occurred at Reynolds numbers of 1980–2600 for the re-entrant, 2070–2840 for the square-edged and 2125–3200 for the bellmouth inlet. A study performed by Smith [8] indicated that transition occurred in the inlet length of the tube and not in the fully developed Poiseuille region. This, combined with the work of Ghajar and Madon, shows that the inlet acts as a disturbance to the flow, which they also concluded. Results published in later work by Tam and Ghajar [1] showed transition to occur at different Reynolds numbers (much higher) than previous results. The transition for the re-entrant inlet started and ended at 2900–3500, for

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Abstract

Due to tube enhancements being used to achieve higher process efficiencies, heat exchangers are starting to operate in the transition region of flow. The paucity of data, however, has the implication that no correlation exists for enhanced tube transition flow. This article, being the first of a two-part paper, presents adiabatic friction factor data for four enhanced tubes for fully developed and developing flow in the transition region. Three inlets were used for developing flows, namely square-edged, re-entrant and bellmouth inlets. It was found that, as in the case of smooth tubes, transition was affected by the type of inlet used, with transition being delayed the most for the smoothest inlet. Correlations were developed to predict the fully developed critical Reynolds numbers and friction factors in the transition region. The correlations predicted the critical Reynolds numbers on average to within 1% with a root mean square deviation of less than 8%, while transition friction factors were predicted with a mean absolute error of 6.6%, predicting 89% of the data to within a 15% error.
the square-edged at 3100–3700 and for the bellmouth at 5100–6100. Once again, though, it is clear that the inlet disturbance influences the critical point where transition occurs.

Transition is also affected by the type of tube augmentation. Nunner [9] found that transition was accelerated by the severity of the augmentation. Nunner inserted different types of circular rings at different distances along the length of the tube. For the same diameter tubes, laminar heat transfer results of the augmented tubes were not significantly higher than those of the smooth tube. Obot et al. [10] analysed the results of previous research performed on transition flow. Specifically by reanalysing the results of Nunner [9] and Koch [11], it was found that the main contributing factor concerning transition was the roughness height.

Extensive augmentation work in the turbulent regime was performed between Reynolds numbers of 2000 and 150,000 mostly with water, although some data were on air and glycol–water mixtures. The augmentation techniques used were internally finned tubes [12], square-helix ridging [13], others with multi-helix ridging [14], micro-finned tubes [15–17], V-nozzle turbulators [18] and finned inserts [19].

For augmentation in the laminar flow regime, most of the experiments were conducted with twisted tape inserts [20–23] with only a few on micro-finned tubes [22]. Reynolds numbers ranged between 15 and 30,000 with water and oil being the main fluids. These experiments were also mostly conducted on the heating of the fluid with only a few performing heating and cooling experiments.

In the transition regime, the experiments were performed on the heating of the fluid, except for those conducted by Manglik and Bergles [21], who also investigated flow in the transition region by means of the cooling of the fluid. Most of the augmentation techniques involved the inserts of tapes [21] and wire coils [24,25]. No helical finned-type tubes have yet been tested in this region. The fluids used were mixtures of water and ethylene/propylene glycol.

Due to the efficiency needs of the future, though, more surface area is added to heat exchangers, with the implication that flow rates per tube drop. These efficiency requirements also imply reduced compressor and pumping power, with the overall trend being that many heat exchangers will start to operate in the transition region of flow. Predictive methods for enhanced tubes in the transition region, however, are unavailable.

The aim of this paper is to present adiabatic friction factor results for helical finned tubes in the transition region for fully developed and developing flow. The data are to be compared with existing laminar and turbulent correlations for enhanced tubes and a new correlation in the transition region for enhanced tubes will be presented. This paper is part of a two-part paper and focusses on adiabatic pressure drops, while the second paper [26] focusses on heat transfer.

2. Experimental set-up

The experimental test section, being one of the components in the experimental test facility, as shown in Fig. 1, consisted of a tube-in-tube heat exchanger in a counterflow configuration. Water

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was used as the working fluid for both streams. For this paper, the annulus loop was not used and all references thereto are exclusively for use with Part II [26]. The test fluid was pumped through the system with two electronically controlled positive displacement pumps. The two pumps were installed in parallel and were used in accordance with the flow rate requirements. The pumps were of similar capacity, each having a maximum flow rate of 270 l/h.

The cold water loop consisted of a 1000 l reservoir, which was connected to a water chiller having a cooling capacity of 15 kW. The temperature inside the reservoir was maintained at 20 °C to prevent the risk of condensate forming on the test section if lower temperatures were used. The water was circulated through the system via an electronically controlled positive displacement pump, which had a maximum flow rate of 2670 l/h. This flow rate ensured that the variation in longitudinal wall temperature was never greater than 3 °C over the whole length of the tube during laminar flow. However, 90% of the laminar experiments were conducted with a wall temperature drop of less than 1 °C. The wall temperature during turbulent flow could not be maintained constant at turbulent Reynolds numbers, although this had a negligible effect on the results in this regime.

Implicit with the use of positive displacement pumps are flow pulsations introduced into the system. This has an unfavourable effect on the stability of the flow, which is crucial when studying transitional flow. To decrease pulsations, a 70 l accumulator was installed before the flow meters and the test section. The accumulators were fitted with rubber bladders filled with air, which dampened these fluctuations resulting in a much more constant pressure at the inlet of the test section. A test was conducted to determine the effect of the accumulators on the flow rate. It was found that without them the mass flow rate fluctuations varied between 0.3% and 0.6%, while with the accumulators the mass flow fluctuations never varied more than 0.05%.

From the accumulators, the water flowed through a set of Coriolis flow meters, which measured the mass flow rate. Two flow meters of different capacities were used according to the flow rate requirements. After the flow meters, the fluid flows to the experimental test section and then back into the reservoirs. Prior to the test section, the fluid first flows through a calming section to stabilise and remove any turbulence in the flow.

The calming section (Fig. 2) was based on work conducted by Ghajar and Tam [27] and consisted of a 5° diffuser, which increased from a diameter of 15–140 mm. This angle was chosen to prevent flow separation from the diffuser wall. Seventy millimetres after the diffuser were three screens separated 105 mm apart. These screens had an open-area ratio (OAR) of 0.31 (60 holes each with a diameter of 10 mm). The OAR is the ratio of the area occupied by the holes to the total area that the whole screen occupies. Located 155 mm from these screens, was a honeycomb, which had an OAR of 0.92 and a length of 100 mm. Prior to and after the honeycomb, was a wire mesh with the wires having a diameter of 0.8 mm and the OAR being 0.54. Another fine wire mesh was inserted prior to the inlet nozzles situated 170 mm from the honeycomb. This mesh had a wire diameter of 0.3 mm and an OAR of 0.17.

Bleed valves were positioned at the top of the calming section at several axial positions. This allowed the bleeding of any air that might have entered the section.

Three different inlets could be housed on the calming section, namely a square-edged, a re-entrant and a bellmouth inlet. These inlets are also shown in Fig. 2 as items a, b and c, respectively. The calming section was designed such that the inlets could easily be interchanged. Fig. 3 shows a basic schematic diagram of each inlet with respect to the test section, from which point heat transfer was initiated.

The length of the fully developed inlet (Fig. 3) was determined in terms of the suggestion by Durst et al. [28], which required a minimum length of 120-tube diameters. To ensure this minimum was met, the length of the inlet was chosen as 160-tube diameters. The inner diameter of this section was the same as that of the test section.

Flow rates were controlled by means of frequency drives, which were connected to the positive displacement pumps. In turn, the frequency drives were connected to a personal computer via a data-acquisition system from which the frequencies could be set. The computer would give a voltage output, which the drives converted to a representative frequency. The finest voltage increments were in the order of 0.02 V, which in terms of Reynolds number were in the order of ∆Re = 20.

The test section (Fig. 4) consisted of a counterflow, tube-in-tube heat exchanger. All the test sections were manufactured from hard-drawn copper tubes, which were insulated with 25 mm thick insulation having a thermal conductivity of 0.034 W/mK. The total length of each test section was between 5.13 and 5.415 m. The tubes tested had nominal outside diameters from 15.806 to 15.889 mm and 19.062 to 19.160 mm. These tubes will be referred to as the 15.8 and 19.1 mm tubes, respectively. The geometric properties of the tubes are given in Table 1. A cross-section of one of the tubes is given in Fig. 5. The heat transfer and pressure drop lengths (Fig. 4) varied between 5.081–5.406 m and 5.130–5.415 m, respectively. More detail can be found in Olivier [29].

A 20.7 mm inside diameter annulus was used for the 15.8 mm diameter tubes and a 26.3 mm inside diameter for the 19.1 mm tubes. The respective hydraulic diameters were 4.81 and 7.21 mm. The annulus outer diameters were chosen such that the annulus hydraulic diameter was small, ensuring high flow velocities and thus turbulent flow within the annulus, keeping its

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**Fig. 2.** Schematic view of the calming section with three different inlet configurations: (a) square-edged; (b) re-entrant and (c) bellmouth (OAR = Open-Area Ratio).

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**Fig. 3.** The four different inlets viewed relative to the test section.

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thermal resistance to a minimum. To prevent sagging and the outer tube touching the inner tube, capillary tube was wound around the outer surface of the inner tube at a constant pitch of approximately 60°. This further promoted mixing inside the annulus and further reduced its thermal resistance.

T-type thermocouples with a nominal wire diameter of 0.2 mm and a limit of precision of 0.1 °C were used for temperature measurements. A total of 53 thermocouples were used, with 12 of them assigned to measure the inner tube and annulus in- and outlet temperatures. Thirty-six thermocouples were placed around the periphery of the inner tube's outer wall at nine axial stations, while the remaining five were placed on the annulus outer wall. All these thermocouples were calibrated with a Pt-100 temperature probe, which itself was calibrated to within 0.01 °C.

Differential pressure measurements were made possible by means of two pressure taps inserted at the inlet and outlet of the inner tube. To ensure the pressure taps did not influence the pressure readings, their diameter was kept below 10% of the inner diameter of the tested tube [30], while great care was taken to remove the burrs formed by the drilling process. The pressure transducers were calibrated with a water manometer having an uncertainty of 0.17%.

A full experimental uncertainty analysis was performed on the system by the method suggested by Kline and McClintock [31], which can be found in Olivier [29]. The range of experimental parameters and their accompanying uncertainties are given in Table 2. The uncertainties listed not only include those due to propagation of error, but also the random errors obtained from experimental measurements. The pressure drop and friction

### Table 1

Geometric properties of the tubes tested.

<table>
<thead>
<tr>
<th>Tube</th>
<th>$D_e$ [mm]</th>
<th>$D_{/}/D_e$ [mm]</th>
<th>$D_h$ [mm]</th>
<th>$L$ [m]</th>
<th>$e$ [mm]</th>
<th>$\gamma$ [°]</th>
<th>$\beta$ [°]</th>
<th>$N$ [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_1$</td>
<td>15.889</td>
<td>14.482</td>
<td>14.482</td>
<td>5.415</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$R_2$</td>
<td>19.062</td>
<td>17.651</td>
<td>17.651</td>
<td>5.140</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$E_1$</td>
<td>15.806</td>
<td>14.648</td>
<td>11.291</td>
<td>5.140</td>
<td>0.399</td>
<td>46.97</td>
<td>18</td>
<td>25</td>
</tr>
<tr>
<td>$E_2$</td>
<td>15.859</td>
<td>14.557</td>
<td>10.201</td>
<td>5.140</td>
<td>0.395</td>
<td>43.93</td>
<td>27</td>
<td>35</td>
</tr>
<tr>
<td>$E_3$</td>
<td>19.160</td>
<td>17.658</td>
<td>13.400</td>
<td>5.138</td>
<td>0.480</td>
<td>38.49</td>
<td>18</td>
<td>25</td>
</tr>
<tr>
<td>$E_4$</td>
<td>19.089</td>
<td>17.816</td>
<td>13.111</td>
<td>5.130</td>
<td>0.467</td>
<td>41.92</td>
<td>27</td>
<td>35</td>
</tr>
</tbody>
</table>

* $R_1$ and $R_2$ are reference smooth tubes. $E_1$ to $E_4$ refer to the enhanced tubes.
factors have such high values of uncertainty, due to the fluctuations in the transition region.

3. Results

The Darcy–Weisbach friction factor was determined from the overall pressure drop, $\Delta p$, as

$$f = \frac{2\Delta p}{\rho u^2 L}$$

(1)

The inside diameter was used for the smooth tubes, while the envelope diameter, as recommended by Marner et al. [32], was used for the enhanced tubes.

3.1. Validation

The validation friction factor data consisted of a total of 427 data sets with 100 data points per set (logged at a frequency of 1.5 Hz), giving a total of 42,700 data points. The data consisted of both the 15.8 and 19.1 mm smooth tubes, referred to as the reference tubes $R_1$ and $R_2$ (Table 1). The data sets consisted of both increasing and decreasing increments of the Reynolds number, which spanned from $Re = 500–13,000$, covering a good portion of the laminar and turbulent flow regimes. A fully developed laminar velocity profile was enforced with the aid of a bellmouth inlet (Fig. 3) and the entrance length being long enough to ensure hydrodynamic fully developed flow. Fig. 6 shows the friction factor results for all the flow regimes, from laminar to turbulent, for increasing and decreasing Reynolds number increments. Included in the figure are the laminar, isothermal equation (Poiseuille flow, $f = 64/Re$), and the Blasius equation for turbulent flow ($f = 0.316Re^{-0.25}$). Furthermore, the data obtained from Senecal and Rothfus [33], Nunner [9], Koch [11], Patel and Head [34], Ghajar and Madon [7], García et al. [24] and the correlation of Churchill [35] for fully developed flow are also given. The data of Ghajar and Madon are based on the bellmouth inlet in the fully developed region of the tube.

Comparing the laminar results with the isothermal laminar (Poiseuille) and turbulent equations (Blasius), the data are underpredicted on average by 1.5% with a maximum deviation of 4.5%. Comparing the increasing and decreasing Reynolds number data, it was found that the two deviate from each other on average by 0.5% with a maximum deviation of 4.3%. The current experimental data and those of others are in excellent agreement with regard to transition. This is even further confirmed by the correlation of Churchill [35], which is in excellent agreement with the data. This adds a great amount of confidence in the current measurement methodology and validates the experimental system for friction factor measurements for enhanced tubes.

3.2. Enhanced tubes: fully developed flow

The friction factors for the enhanced tubes are shown in Fig. 7. Also included are the data for the smooth tube ($R_1$) for fully developed flow. A few things can be noticed from this figure. First, there is an upwards shift in friction factor values in the turbulent as well as in the laminar flow regimes compared with the smooth tube results. Second, transition occurs earlier than for the smooth tube. Third, the results of the enhanced tubes $E_4$ and $E_3$ and the tubes $E_1$ and $E_2$ lie on top of each other after transition to turbulence has occurred. This is because the relative roughness and helix angle of the tubes are the same, even though they have different diameters. Fourth, there appears to be a “secondary transition” between Reynolds numbers of 3000 and 10,000 where there is a smooth second increase in the friction factors. The upward shift in friction factor is understandable. This is due to the increase in roughness the fins exhibit, which, in turn, increases the amount of resistance to flow. These results are in conjunction with those of other authors, for example, Vicente et al. [36] and García et al. [24], who investigated laminar-to-turbulent flow inside dimpled tubes.

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and tubes with wire inserts. The earlier transition can also be attributed to this increase in roughness.

The secondary transition, however, would most probably be due to the effective rotation the fins bring about the fluid. Results where different types of roughness for tubes were compared did not show this secondary effect, where these roughness were in the form of ring inserts or dimpled tubes [9,11,37]. The shape of the curve in this region can be explained by means of the effectiveness the fins have in rotating the fluid. At the lower Reynolds numbers, the fins are ineffective in rotating the fluid and only as the velocity is increased, do they become more effective. Similar effects are seen in the results of Brongnaux et al. [17] and Jensen and Vlakancic [38], with micro-fin and high-fin tubes. Only after a Reynolds number of approximately 10,000 does the secondary transition stop and the friction factors continue along the standard roughness-height-to-diameter ratio lines depicted in the Moody chart. It should be noted that the relative roughness of the two diameter tubes for both the 18° and 27° enhanced tubes are the same. Thus, the higher friction factor values of the 27° tubes are then purely due to the helix angle.

Fig. 8 depicts the relative friction factor standard deviation for both the 18° and 27° enhanced tubes. This was calculated by taking the measured friction factor’s standard deviation of 100 data points per set and dividing it by the average value of these 100 data points. This figure, when used in conjunction with Fig. 7, shows an increase in pressure fluctuations in the region where the friction factors start to deviate from the laminar and turbulent predicted values. The fluctuations also peak midway between these two regions. The increase, peak and decrease of the fluctuations occur in a Reynolds number span of approximately 600. Transition from laminar to turbulent flow for the two different helix angles all commence at Reynolds numbers from 1900 to 2100, with the 27°, 19.1 mm tube (E4), showing the greatest delay in transition. Transition for this tube occurs at a Reynolds number of about 2070, while for the other tubes transition occurs at approximately 1870. This slight delay in transition is not due to inlet disturbances, as the same inlet was used for all the tubes. Furthermore, the fin-pitch-to-diameter ratios have values of 0.39 and 0.18 for the 18° and 27° enhanced tubes, respectively, showing that its effect is also negligible. The same can be said regarding the helix angle. Thus, the only geometrical aspect that would have an influence on transition is the fin-height-to-diameter ratio. Three of the tubes have a ratio of 0.027 with transition occurring at a Reynolds number of about 1870, while the fourth tube, being the 27° 19.1 mm tube, has a ratio of 0.022. Thus, it follows that only the roughness height has an influence on transition.

Similar conclusions were made by Vicente et al. [36,39], who performed tests on corrugated and dimpled tubes. Their correlations predicted the critical Reynolds numbers only in terms of the roughness-height-to-diameter ratio.

The friction factor fluctuations peak at a Reynolds number of approximately 2200 and drop down again at approximately 2600 with increasing Reynolds numbers. It should be noted, though, that in the Reynolds number region of 3000–10,000, where the secondary transition commences, the fluctuations are stable. This shows that this region is not at all chaotic and is, from repeated experiments, predictable. It can further be stated that fully turbulent flow for these enhanced tubes is only reached at Reynolds numbers above 10,000. For the enhanced tubes, as with the smooth tubes, no hysteresis was noted. All the data presented in Figs. 7 and 8 are for increasing and decreasing Reynolds numbers.

3.3. Enhanced tubes: developing flow

Fig. 9 shows the adiabatic friction factor results for the enhanced tubes with developing flow by means of different inlets. Transition for the 15.8 mm tubes (E1 and E2, Fig. 9a) occurs at different Reynolds numbers, depending on the type of inlet used. The bellmouth inlet delays transition to Reynolds numbers higher than...
those for the square-edged and re-entrant inlets, while the square-edged inlet delays transition to Reynolds numbers higher than those for the re-entrant inlet. These trends are similar to those obtained for the smooth tube [1,2,6,40]. For the 19.1 mm tubes (E₃ and E₄, Fig. 9b), the delays in transition are less pronounced, with only the bellmouth inlet showing significant delays. The delay in transition for the 19.1 mm square-edged inlet is much less than obtained for its 15.8 mm counterpart. The 19.1 mm bellmouth inlet does, however, show a greater delay in transition when compared with the 15.8 mm bellmouth inlet, with the effect being very similar to the delay in transition between 15.8 and 19.1 mm smooth tubes with the bellmouth inlet [40].

Turbulent flow results are unaffected by inlet profile as shown by the current data and the fully developed enhanced tube data. Laminar results are also the same, both being slightly higher than the Poiseuille relation. This is also attributed to the fin height/surface roughness, with the helix angles having no effect. Although there is a boundary layer growth due to the inlet profiles, this is not evident in the laminar friction results. Olivier [29] as well as Ghajar and Madon [7] showed that this developing boundary layer was responsible for the increase in friction factors for the smooth tubes with various inlets when compared with the fully developed inlet, which also confirms the theoretical values for laminar developing flow [41]. This is not evident for the enhanced tubes, implying that the tube roughness has a greater effect on the wall shear stress than the effect of the boundary layer. Only the bellmouth inlets near the transition region are slightly affected by the boundary layer growth. It is also evident from the results that the helix angle has no effect on this increase in friction factor, as was found for the fully developed case.

The secondary transition is not influenced by the inlet geometry as the data lay on top of each other for the respective helix angles. After transition, the friction factors follow the secondary transition’s profile, as seen from the square-edged and re-entrant data.

Fig. 10 shows the fluctuation in friction factors for all the inlets and enhanced tubes. This confirms the results from Fig. 9 with regard to where transition starts and ends. For the re-entrant inlet, the start of transition varies between 1985 and 2080 for the different tubes, showing that it is unaffected by the tube diameter and fin helix angle. Only the end of transition appears to be affected by the helix angle, with transition ending at approximately 3000

Table 3
Adiabatic friction factor correlations for enhanced tubes.

Carnavos [12]

\[
f_s = 0.184 \text{Re}^{0.12} (D_t/D_h)^2 (A_t/A_r)^{0.5} (\sec \beta)^{0.75}
\]

\[
0 < \beta < 30^\circ, \quad 10,000 < \text{Re} < 100,000
\]

Ravigururajan and Bergles [42]

\[
f_s/f_s = \left\{1 + \left[29.1 \text{Re}^2 \left(e/D_t\right)^{a_2} (p_t/D_t)^{a_3} (\beta/90)^{40} (1 + 2.94 \cos (\gamma/2N))\right]^{15/16}\right\}^{16/15}
\]

\[
0.01 < e/D_t < 0.2, \quad 0.1 < p_t/D_t < 7.0, \quad 0.3 < \beta/90 < 1.0, \quad 5000 < \text{Re} < 250,000
\]

\[
a_1 = 0.67 - 0.06p_t/D_t - 0.49 \beta/90
\]

\[
a_2 = 1.37 - 0.15p_t/D_t
\]

\[
a_3 = -1.66 \times 10^{-7} \text{Re} - 0.33 \beta/90
\]

\[
a_4 = 4.59 + 4.11 \times 10^{-6} \text{Re} - 0.15p_t/D_t
\]

Jensen and Vlakancic [38]

\[
f_s/f_s = (l_{sw}/D_t)^{1.27} (A_t/A_r)^{1.75} - 0.0151 (l_{sw}/D_t)^{1.27} (A_t/A_r)^{1.75} - 1 \exp(-\text{Re}/6780)
\]

\[
2000 < \text{Re} < 80,000, \quad 0 < \beta < 45^\circ, \quad 0.0075 < e/D_t < 0.05
\]

\[
l_{sw}/D_t = \left[1 - 0.994 N \sin \beta/\pi 0.032 (2e/D_t)^{0.44} \times (\pi (N - s/D_t) \cos \beta)^{0.44}\right]
\]

\[
s = 4\sqrt{3} \tan(\gamma/2) = \text{average width of triangular fin}
\]

\[\]

\[\]

A Appears to be incorrectly given in his original paper, with this correlation being taken from [44].
for the 18° tubes and 2500 for the 27° tubes. This shows that the turbulence is higher for the greater of the two helix angles as Reynolds numbers increase.

For the square-edged inlet, transition for the 15.8 mm starts at a Reynolds number of approximately 2800 for the 18° tube, and 2500 for the 27° tube. For the 19.1 mm tube, transition starts at 2200 and 2050 for the respective helix angles. It would appear that in this instance, the helix angle and tube diameter have an effect on the delay of transition. The transition from laminar to turbulent flow for the square-edged inlet is very abrupt, spanning a relatively small range of Reynolds numbers.

The bellmouth inlet shows the greatest delay in transition, being in the Reynolds number range of 3900–4600 for the 15.8 mm enhanced tubes, and 5500–5800 for the 19.1 mm tubes. The transition for the bellmouth, though, occurs at lower Reynolds numbers than the transition for its smooth tube counterparts (7000 and 12,000 for the 15.8 and 19.1 mm smooth tubes, respectively). This shows that the roughness of the fins influences the stability of the boundary layer, and hence its ability in maintaining laminar flow at high Reynolds numbers.

4. Correlations

Since there is a definite increase in laminar and turbulent friction factors, the data are compared with the literature.

4.1. Laminar flow

For laminar flow, the increase in friction factor would mainly be due to the roughness height of the fins and not the fin helix angle or number of fins. This was observed by Vicente et al. [36], examining helical dimpled tubes. They found that it was only the dimple height that affects the friction factor, and not the density of the dimples. They proposed the following correlation for laminar flow:

\[
\frac{f_{Le}}{f_{Re}} = \frac{64}{Re} \left[ 1 + 123.2 \left( \frac{e}{D_x} \right)^{2.2} Re^{0.2} \right].
\] (2)

This correlation predicted the laminar friction factor data for all four enhanced tubes with an average mean deviation of 6% and an average root mean square (rms) deviation of 10%. This improved, though, by changing the constant, 123.2 to 88, predicting the data with an average mean deviation of 1.3% and an average rms deviation of 7.3%. It is thus proposed that Eq. (2) becomes

\[
\frac{f_{Le}}{f_{Re}} = \frac{64}{Re} \left[ 1 + 88 \left( \frac{e}{D_x} \right)^{2.2} Re^{0.2} \right].
\] (3)

4.2. Turbulent flow

For turbulent flow, the correlations of Carnavos [12], Ravigururajan and Bergles [42] and Jensen and Vlakancic [38] are compared with the experimental data. Table 3 lists the authors and their respective correlations as well as their limits of use. The reason the correlations of these three were chosen was the fact that these are the most recent correlations developed for low-fin enhanced tubes for single-phase flow. It is noted that these correlations are all functions of the geometric parameters of the tubes.

Fig. 11 shows the performance of these correlations against the experimental data for the four enhanced tubes, spanning a Reynolds number range of 3000–19,000. The Carnavos [12] correlation, originally developed from a wide range of enhanced tubes, had a mean absolute error of 10%, with 62% of the data being predicted to within 10%. The correlation deviates from the data for Reynolds numbers lower than 8000, which is below its applicable range of 10,000.

The correlation of Ravigururajan and Bergles [42] predicts the data with a mean absolute error of 6.2%, predicting 92% of the data to within 10%. The data deviate from the correlation in the lower turbulent region, Reynolds numbers lower than 5000. This is also
below the applicable range of this correlation. The correlation of
Jensen and Vlakancic [38] predicts the data with a mean absolute
error of 4.7% with 95% of the data being predicted to within 10%.
This is the only correlation that predicts the data for the whole tur-
bulent regime while noting that it is applicable for Reynolds num-
bers greater than 2000. This excellent agreement between data and
correlations also further validates the experimental system.

4.3. Critical Reynolds numbers

Since very few data regarding these values for enhanced tubes exist (including the present data), the data of Vicente et al. [39] were utilised in the development of correlations. The choice for using the helical corrugated tube data was due to these tubes having the closest resemblance to the current enhanced tubes for the available data. Other data by García et al. [25] for wire coil inserts in the transition region were found. However, the wire heights were much greater than those in the current study with their results showing not only very early transitions, but also a very gradual change from laminar to turbulent flow with no distinct critical value. This was due to the wires inducing a swirl flow in the laminar regime, with the trends being very similar to the twisted tape inserts investigated by Manglik and Bergles [21].

Data of fin-height-to-diameter ratio and critical Reynolds numbers are given in Fig. 12. Since pressure fluctuation data of other research were not available, the critical Reynolds numbers were taken as the minimum friction factor at transition. From the data, it was found that the curve that predicts the critical Reynolds the best is

\[ \text{Rec}_c = 222 \left( \frac{e}{D_e} \right)^{-0.58} \]  

which is valid for \( 0.022 < \frac{e}{D_e} < 0.057 \). To extend its range of validity to include the critical Reynolds number of the smooth tube with a fully developed inlet profile (\( \text{Rec}_c = 2200 \)), the method of Churchill [43] was incorporated. Thus, the limiting form for a fully developed smooth tube is

\[ \text{Rec}_c = 2200. \]  

Writing Eqs. (4) and (5) in the form of an \( n \)th-order asymptotic solution and rearranging gives

\[ \text{Rec}_{c}/2200 = \left[ 1 + \left( \frac{222(e/D_e)^{-0.58}}{2200} \right) \right]^{-1/n} \]  

which also has the form

\[ Y = \left[ 1 + Z^{-n} \right]^{-1/n}. \]  

The constant \( n \) can be evaluated for \( Z = 1 \), which gives the central value of \( e/D_e \), which in this case has a value of 0.0175. Thus, the value of \( n \) could be computed for \( Z = 1 \) if the value of \( Y(1) \) were known, which unfortunately it is not. Thus, by loosely fitting Eq. (6) for different values of \( n \) to the data, a value of \( n = 10 \) was chosen. Thus, the final correlation predicting the critical Reynolds number is given as

\[ \text{Rec}_c = 2200 \left[ 1 + 9.13 \times 10^0 \left( \frac{e}{D_e} \right)^{0.58} \right]^{-1/10}. \]  

This correlation predicts the current set of data on average to within 1% with an rms deviation of 8% and a maximum deviation of less than 18%.

4.4. Transitional flow

Since the transition region is characterised as lying between the laminar and turbulent flow regime, it would only be fitting if this region were described by the same parameters used to predict these regimes. Thus, as with turbulent flow for enhanced tubes, the main parameters having an influence on this region will be similar. It is thus suggested that the correlation should have the form

\[ f_e = f(\text{Rec}_c, e, D_e, p_f, \beta). \]  

By rearranging the parameters, the best form of the correlation was found to be

\[ f_e = 4 \left( \frac{16}{\text{Rec}_c} \right)^{0.04} \left( \frac{\text{Re}_c}{\text{Rec}_c} \right)^{0.07} \left( \frac{\text{Re}_c}{\text{Rec}_c} \right)^{0.08} \left( \frac{p_f}{\text{pf} D_e} \right)^{-0.48} \left( \frac{e}{D_e} \right)^{0.06}. \]

\[ \left[ \frac{e}{D_e} \right]^{-0.58} \left( \frac{\text{pf} D_e}{p_f} \right)^{0.009} \left( \frac{\text{pf} D_e}{p_f} \right)^{0.37} \]  

The constants were found by means of a non-linear least-squares optimisation method. The final correlation for fully developed flow then becomes

\[ f_e = 4 \left( \frac{16}{\text{Rec}_c} \right)^{0.04} \left( \frac{\text{Re}_c}{\text{Rec}_c} \right)^{0.07} \left( \frac{\text{Re}_c}{\text{Rec}_c} \right)^{0.08} \left( \frac{p_f}{\text{pf} D_e} \right)^{-0.48} \left( \frac{e}{D_e} \right)^{0.06}. \]

This correlation is valid for \( \text{Rec}_c \leq \text{Re}_c, 18^\circ \leq \beta \leq 79^\circ, \ 6.14 \times 10^{-4} \leq \frac{e}{D_e} \leq 0.004, \ 6.48 \times 10^{-4} \leq \frac{\text{pf} D_e}{p_f} \leq 1.23 \) and \( 0.022 \leq e/D_e \leq 0.057 \).

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Fig. 12. Critical Reynolds numbers for enhanced tube fully developed friction factors.

Fig. 13. Predicted vs. experimental adiabatic transitional friction factors for enhanced tubes for fully developed flow.
Note that the upper limit for the Reynolds number has not been defined. This limit can be set as the intersect of Eq. (11) and an appropriate turbulent correlation. For the current experimental data, the model of Jensen and Vladančić [38] should be used. A comparison of the experimental data with Eq. (11) is shown in Fig. 13. The correlation has a mean absolute error of 3% and predicts 95% of the data to within 10%.

For developing flow, unfortunately, there was no obvious trend regarding the critical Reynolds numbers. For this reason, the constants in Eq. (10) are selected based on the inlet and type of enhanced tube. Table 4 lists these values. Fig. 14 shows the experimental values against the predicted values. The correlation predicts the transition friction factors with a mean absolute error of 6.6%, predicting 89% of the data to within 15%. It should be noted, though, that the greatest deviation from the correlation is the data for all the bellmouth inlets, due to their trends being different from the others. Removing the bellmouth data, the correlation then has a mean absolute error of 3.7%, predicting 93% of the data to within 10%.

5. Conclusion

This paper presented experimental adiabatic friction factor data for enhanced tubes for fully developed and developing flow in the transition Reynolds number region for different types of inlet disturbances. It was found that transition for enhanced tubes occurs at lower Reynolds numbers than for their smooth tube counterparts. Furthermore, transition, as in the case for smooth tubes, is also influenced by the type of inlet used. The smoother the inlet, the greater the critical Reynolds number. In this case, the greatest critical Reynolds numbers were obtained for the bellmouth inlet, with the re-entrant showing the lowest values.

Turbulent friction factors were much higher than those for smooth tubes, but were unaffected by the inlet used. A secondary transition is noticed for Reynolds numbers between 3000 and 10,000. It was found that this region was, unlike the transition region, very stable and predictable. Turbulent correlations in the literature were in excellent agreement with the data.

Laminar friction factors were generally higher than for their smooth tube counterparts. This was attributed to the relative roughness of the tube due to the fins. The effect of the developing boundary layer is diminished by the fins. A new correlation for the laminar region was proposed, predicting the data on average to within 1.3%.

Correlations were developed to predict the fully developed critical Reynolds numbers. This correlation predicted the data on average to within 1%. A correlation was also developed to predict friction factors in the transition region for enhanced tubes. This correlation predicted 89% of the data to within 15%, having a mean absolute error of 6.6%. This was improved by removing the bellmouth data, with the correlation then predicting 93% of the data to within 10% and having a mean absolute error of 3.7%.

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