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Turbulent Hydraulic Friction in Spirally Corrugated Tubes for Condenser Applications

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A modern method to augment the heat transfer in shell-and-tube heat exchangers is to replace the smooth tubes by spirally corrugated tubes (SCT). The applications include multistage flash desalination plants, power station turbine condensers, marine condensers, etc. The heat transfer augmentation is a result of the turbulence promoters (helical ridging) at the inside of the tube. They are produced by a cold rolling operation that creates a corrugated tube having internal helical ridging and external helical grooving. The promoters increase both the intube heat transfer coefficient and the hydraulic friction factor. If there is a requirement for a cost effectiveness of a particular heat exchanger design the pressure losses in the tubes could affect considerably the choice of a particular heat transfer surface.

When a single phase fluid flows turbulently near a wall with artificial wall roughness the structure of the flow near the wall is quite complicated. The flow in SCT is characterized by the interaction between the turbulent core and the wall region where flow separation and vortexes exist. The intensities of the latter effects determine the heat transfer enhancement. Flow visualization studies [1, 4], velocity profiles and turbulent fluctuation measurements [2, 3] indicate that the magnitude of the interaction between the core and the wall region depends on the lead angle θ and the relative height e/D_i, fig. 1. If $\theta > 60°$ the flow near the wall crosses the ribs with separation. The form drag is a major component of the flow resistance, like the case of a flow across transverse ribs ($\theta = 90°$). When $\theta < 45°$, it is shown [1, 4] that the flow near the wall is directed along the ribs and the effect of the spiral motion is dominating. In the range $45° < \theta < 60°$ a transition from a swirling-dominated flow to a cross-over flow takes place and the momentum and heat transfer are influenced by this flow mode.

A brief analysis of many experimental results [5-12] exhibits both quantitative and qualitative differences between the conclusions of the authers. These differences come from three main sources:

a) incomplete knowledge of the influence of the geometrical parameters of the promoter, in particular the shape of the ridge, on the momentum transport;

b) the search of empirical equations correlating data derived from two different flow regimes — cross-over flow ($\theta > 60°$) and swirling flow ($\theta < 45°$);

c) the correlation of data for promoters with relative height, $e/D_i > 0.06$ — in this case the law of the wall is no longer valid [13, 14] and the flow depends on the form of the channel.

Purpose of this brief is to analize the hydraulic characteristics of 25 SCT (23 single-start and 2 three-start tubes) in relation to some geometrical parameters which determine the shape of the helical ridges and influence the hydraulic resistance coefficient. All tubes have lead angles $\theta > 60°$ because as shown in [1], the heat transfer charactericticś of the tubes are higher.

Despite the fact that many experimental works are published, the lack of sufficient knowledge about the flow mechanism in the presence of helical ridges makes impossible the analytical determination of the friction factor and the heat transfer coefficient. One way to overcome this difficulty is to apply the similarity law concept introduced by Nikuradse [13]. This concept was successfully applied to correlate the friction results for intube flow with sand-grain wall roughness [14], used presently by many authors to describe their experimental results. The model is based on the idea of the existence of two distinct regions of the flow:

a) inner region where the law of the wall is applicable and the velocity distribution depends only on local conditions like y, τ_w, υ and e, and represented by

$$U^+ = \left(\frac{U}{U_*}\right) = \Phi_1\left(\frac{yu_*}{\nu}\right) = \Phi_1(y^+); \tag{1}$$

b) outer region where the velocity defect law is effective; this region is assumed to be insensitive to both roughness of the surface and viscosity of the fluid flowing inside the tube and the velocity distribution can be represented by

$$\left\{\frac{U_{\max} - U}{u_*}\right\} = \Phi_2(y/e). \tag{2}$$

In eqs. (1), (2) $u_* = (\tau_w/\rho)^{0.5}$, m/s, is the friction velocity τ_w, Pa, is the wall shear stress; ρ, kg/m^3, is the fluid density; ν, m^2/s, is the kinematic viscosity; y, m, is the radial distance from the wall; e, m, is the height of the turbulent promoter.

The combination of the law of the wall and the velocity defect law gives the velocity distribution equation for the turbulence-dominated part of the wall region, which is given by

$$U^+ = 2.5 \ln(y/e) + R(e^+), \tag{3}$$

where $e^+ = eu_*/\nu$ is the dimensionless height of the turbulent promoter. Once the function $R(e^+)$ is known, the friction coefficient can be obtained from

$$R(e^+) = (2/f)^{0.5} + 2.5 \ln(2e/D_i) + 3.75, \tag{4}$$

where f is the Fanning friction factor and D_i, m, is the internal diameter of the tube.

The model based on eqs. (1)-(4) has been used by all authors investigating SCT [1, 4, 8, 11, 12]. It has generally been assumed that the momentum transfer roughness function, $R(e^+)$, depends both on the geometry of the roughened surface and the physical properties of the fluid

$$R(e^+) = \Phi_3(e/D_i,\ p/e,\ \theta/90,\ Re), \tag{5}$$

where p, m is the pitch (the distance between two adjacent ridges) and $Re = U_m . D_i/\nu$ is the Reynolds number. It has to be noted that tubes with $\theta > 60$ only are to be considered in what follows. Although the simplex p/e is essential it does not reflect the influence of the shape of the promoter on the flow between two adjacent ridges.

The internal wall profile of a corrugated tube having internal helical ridging is shown on Fig. 1. For the tubes studied here the shape of the turbulent promoter profile is very similar to the one of the tubes manufactured by UOI Inc., Wolverine Division, USA, called KORODENSE, [8] and IMI Yorkshire Imperial Alloys, Leeds, U. K. [15], called ROPED, while the promoters of the tubes studied in [1, 11, 12] have semicircular profiles. For this reason we would like to pay attention to the following:

In addition to p, e and D_i, usually used to determine $R(e^+)$ from eq. (5), we assume that another two geometrical parameters will also influence the flow behaviour in the space between two adjacent ridges, namely, t — the axial width of the ridge cap and s — the radial height of the ridge cap. These two parameters are defined as follows: s is the distance between the crest of the ridge and the inflection point where the concave and convex portions of the (symmetric) ridge join each other smoothly and have a common tangent, whereas t is the distance between the inflection points of the two slopes of the ridge. These parameters have been mentioned, for the first time, in [16] where an attempt has been made to use them for an in-tube heat transfer coefficient correlation but since then they have not appeared in the reports. Similar attempts to introduce the width of the semicircular promoter can be encountered in [11, 12] but without any final development.

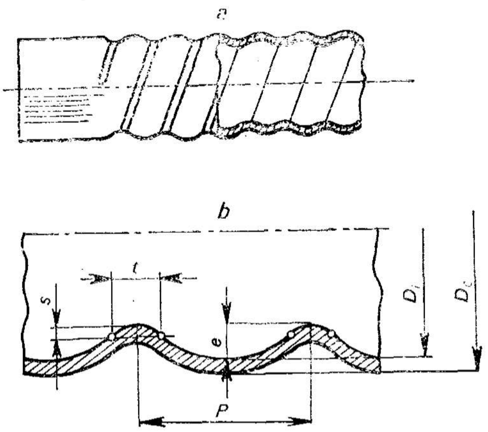

Fig. 1. Characteristic parameters of a spirally corrugated tube

The internal and external diameters D_i and D_o are defined as in [16] and settled later [17] as a general approach.

Having in mind the physical nature of the flow between two adjacent promoters, we suggest two geometrical complexes to determine its intensity — these are $(p-t)/e$ and s/e. The former characterizes the evolution of the flow separating from the promoter while the latter defines the degree of separation of the flow and the swirls after the promoter. Multiplying those two simplexes we introduce a new one

(6)
$$\Phi_* = \frac{(p-t) \cdot s}{e^2}$$

to replace the simplex p/e in eq. (5) and to take into account the phenomena mentioned above. Now the roughness function $R(e^+)$ can be sought in the form of a power law

(7)
$$R = a_o (e/D_i)^{a_1} \Phi_*^{a_2} \theta_*^{a_3} \mathrm{Re}^{a_4},$$

where $\theta_* = \theta/90$. This was done by means of the experimental study of 25 SCT divided into two groups. (The geometrical parameters of these tubes will be published elsewhere.) The SCT 11-17 and 21-27 were manufactured in pairs at identical technological conditions but tubes 11-17 were processed by a roll having a radius $r = 1.10^{-3}$ m, while tubes 21-27 — by a roll with a radius $r = 15.10^{-3}$ m. The tubes 11, 12 and 22, 13 and 23, 14 and 24, 15 and 25, 16 and 26, 17 and 27, have equal e/D_i, p/e, and θ_* values in the groups.

The Fanning friction factors measured are shown on fig. 2. It is clearly seen that tubes 11, 12, 22, 14, 24 and 17, 27 have different friction factors despite the fact that

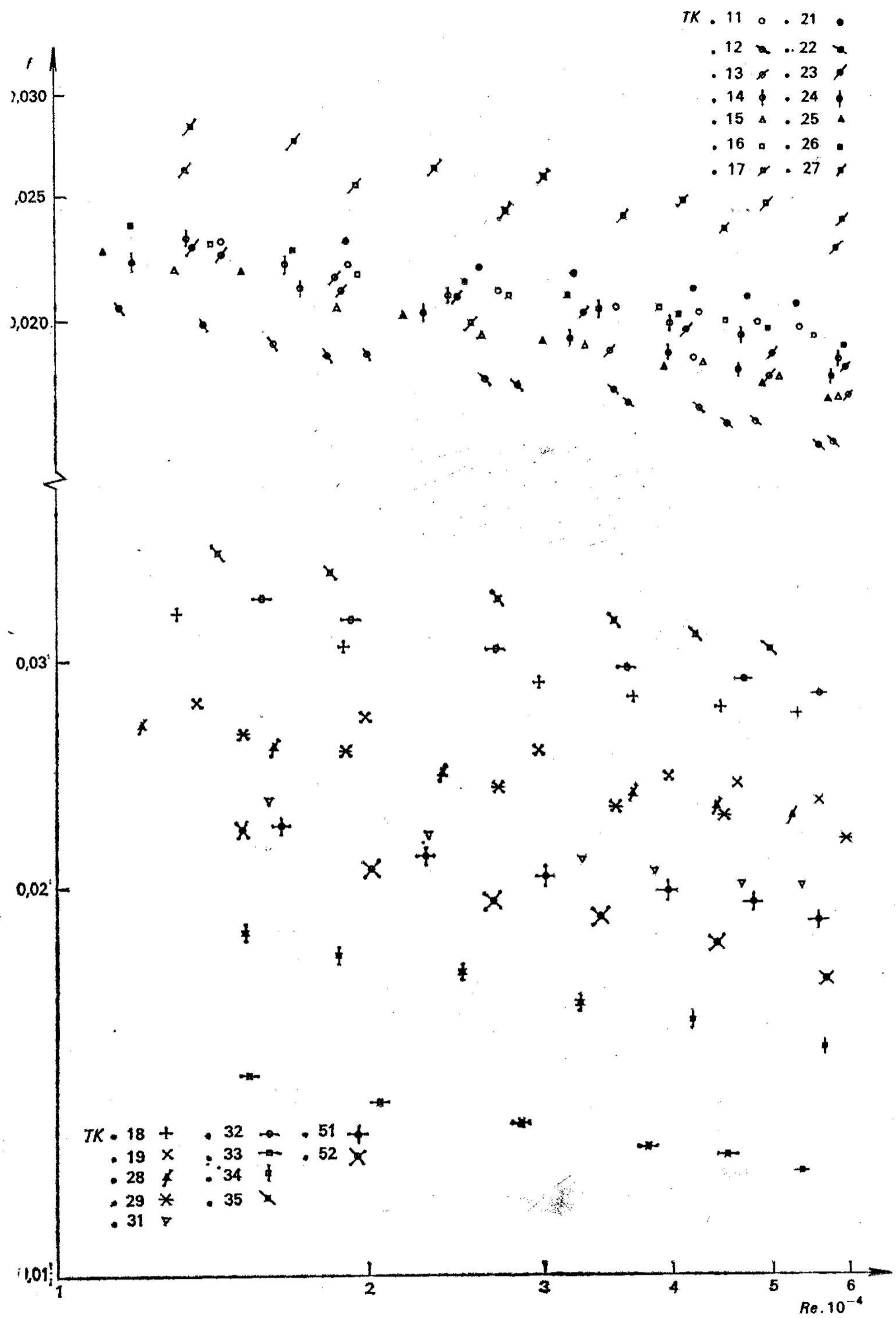

Fig. 2. Variations of the friction factor f with Reynolds number Re

their e/D_i, p/e and θ_* values are equal. Similar observations are reported by Withers [8] for tubes 1 and 2. The second group of tubes was manufactured to investigate the validity of the hypothesis, eq. (7), by varying the values of the corresponding parameters in wider ranges. The friction factors measured are shown on Fig. 2. The experiments reported concern tubes having single-start helical ridging mainly but we have included two three-start tubes to obtain some additional information. Fig. 2 indicates that for the ranges

$$0.017 < e/D_i < 0.047\,; \qquad 0.760 < \theta_* < 0.950\,;$$

$$1.40 < \Phi_* < 5.90\,; \qquad 10^4 < \mathbf{Re} < 6\,.\,10^4\,;$$

there are no tubes indicating a fully rough flow regime behavior and the friction factor continues to decrease with increasing the Reynolds number. On the basis of 150 experimental points a least squares data fit of eq. (7) was obtained in the form

$$R = 0.416\,.\,\mathbf{Re}^{0.1}(e/D_i)^{-0.42}\,\theta_*^{-1.94}\,\Phi_*^{0.08} \tag{8}$$

Eq. (8) was obtained by QR decomposition [18] of the data matrix [19]. All calculations were done on a PC IBM- XT computer (relative machine precision $1.17\,.\,10^{-7}$). The correlation, eq. (8), and the experimental points are shown on fig. 3. All points lie within the $\pm 10\%$ band.

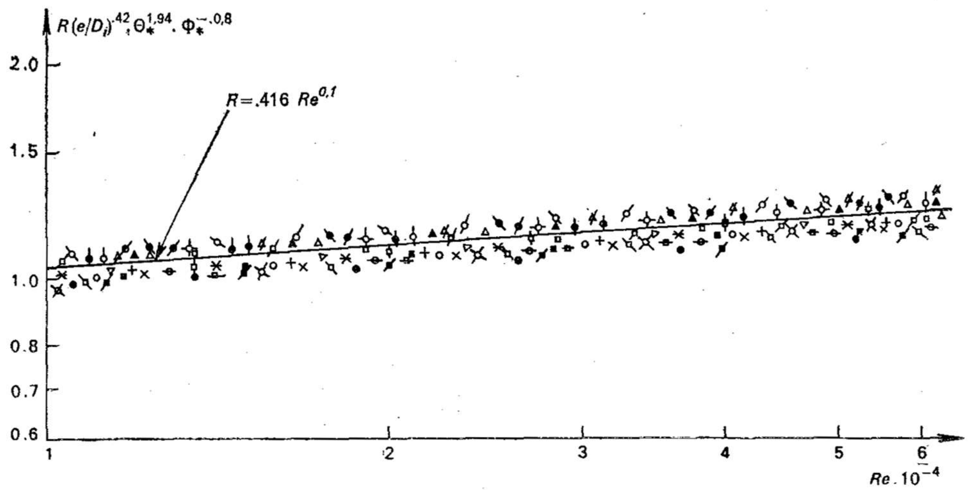

Fig. 3. Variations of $Re(e/D_i)^{0,42}\theta_*^{1,94}\Phi^{*-0,08}$ with Reynolds numbers

Using eqs. (4) and (8) the friction factor can be calculated from

$$(2/f)^{0.5} = 2.5\ln(D_i/2e) - 3.75 + 0.416\,\mathbf{Re}^{0.1}\,(e/D_i)^{-0.42}\theta_*^{-1.94}\,\Phi_*^{0.08}. \tag{9}$$

The friction factors corresponding to 346 experimental points were calculated by means of eq. (9). The comparison between experimental and calculated values showed relative errors more than 10% for 6 points only.

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