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Condensation of Steam on Corrugated (Roped) Tubes for Horizontal Condensers — the Heat Transfer Coefficient

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It is well known that during condensation of steam on smooth tubes a large thermal resistance occurs due to conduction of heat across the falling film and any attempt for thinning or disturbing that film will be beneficial to heat transfer. Despite the fact that surface condensers, utilizing corrugated tubes instead of smooth ones, yield an improvement in their thermal effectiveness due mainly to the enhanced internal heat transfer coefficient, it is interesting to know how the shape, size and frequency of the grooves affect the steam-side heat transfer coefficient.

In a recent paper [1] we discussed the idea that additional geometrical parameters can be used to describe the effect of the shape of the ridge on the hydraulic friction for a flow in spirally corrugated tubes suggesting a new dimensionless geometrical complex. This idea was used to develop a „mixing-length" model [2] to predict the friction factor and water-side heat transfer coefficients in spirally corrugated tubes which is an extension of the model [3] treating internally sand-roughened tubes.

Numerous articles have been issued discussing the effect of the grooves (ridges) on the in-tube heat transfer augmentation, but few investigated their influence on the steam-side heat transfer enhancement [4-8]. Moreover, except [9]. the influence of the shape of the groove on the steam-side enhancement has not been studied. An analytical solution to the heat transfer problem for condensation of steam on the external surface of a horizontal corrugated (roped) is difficult to obtain and (as far as we are aware) has not been reported yet. This is not surprising as simultaneous heat and mass transfer is involved in a three-dimensional flow system containing a free surface influenced by surface-tension forces. Moreover, the large scattering of experimental data does not stimulate efforts for development of rigorous mathematical models.

Purpose of this brief is to obtain a simple correlation for the steam-side heat transfer enhancement. It will be derived from a physical model for the influence of the shape of the groove on the steam-side heat transfer coefficient. This correlation is expected to be useful for design calculations. The shape of the groove will be described by the same geometrical parameters utilized earlier [1, 2] to model the influence of the shape of the ridge on the friction factor and in-tube heat transfer coefficient. Thus a complete model requiring at input the standard in-tube Reynolds and

Prandtl numbers and the geometrical parameters of the ridge (groove) will be obtained to predict theoretically the Fanning friction factor and heat transfer enhancement at both sides of the tube.

As shown by Gregorig [10] when vapour condenses on a corrugated tube surface tension forces are introduced in the liquid film and act in two ways to improve the heat transfer: first by reducing the thickness of the film on the convex parts of the groove and its thermal resistance and second, by shedding liquid more quickly from the concave parts of the tube surface. As a result there will be a net gain in the overall heat transferred if the increase of the heat transfer over the convex portions is more than the reduction in heat transfer over the concave surface. This depends on the capabilities of the grooves to drain with enhanced condensate rate. Therefore the factors likely to be important in assessing the performance of a particular groove shape and geometry are:

a) the relationship between surface tension and gravity forces causing film thinning or thickening;

b) the ability of the grooves to drain and shed the condensate.

Following [7] the relationship "surface tension vs. gravity forces" can be expressed as the Weber number We, defined by

$$\mathrm{We} = \frac{\rho P/\rho x}{\rho g}. \tag{1}$$

Assuming (ideally) that the shape of the groove surface is a part of a circular arc, Fig. 1, (for the tubes under study, brass 28×1, such assumption can be made since this configuration is obtained by cold rolling operation which embosses an internal projection known as ridge, in registration with an external groove and is attained without thinning the tube wall at the focus of the corrugation) the surface tension, forces generated in the liquid film create a pressure drop

$$\partial P = \sigma\{1/R_1 + 1/R_2\}. \tag{2}$$

To take into account the fraction of the surface grooved, i. e. the groove frequency we shall consider this pressure drop over a distance $\partial x = p$ (one pitch). Thus eq. (1) yeilds

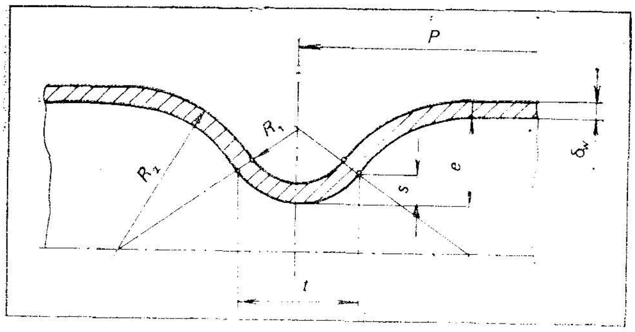

Fig. 1. Characteristic parameters of a corrugated tube

$$\mathrm{We} = \frac{\sigma}{\rho g d p}\{1/R_1 + 1/R_2\}. \tag{3}$$

In equation (3) σ, kg/ s^2 is the surface tension force coefficient; ρ, kg/m^3 is the density of the condensing film; g, m/s^2 is the gravity acceleration; p, m is the pitch of corrugation and R_1, R_2, m are the radii of the concave and convex parts of the groove. By simple geometry (Fig. 1) the radii R_1 and R_2 can be expressed as

(4) $$R_1=0.5s\{1+0.25(t/s)^2\}-\delta_w,$$
$$R_2=0.5(e\text{-}s)\{1+0.25(t/s)^2\}+\delta_w,$$

where e, m is the height of the ridge; s, m is the cap height of the ridge; t, m is the cap width of the ridge; δ_w, m is the wall thickness. The parameters s and t 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 smoothly and have a common tangent, whereas t is the distance between the inflection points of the two slopes of the ridge. These parameters can be measured accurately and were used in [1, 2] to express the influence of the shape of the ridge on the friction factor coefficient and in-tube heat transfer coefficient. The same geometrical parameters can be utilized now to model the influence of the shape of the groove on the steam-side enhancement.

The groove capability to shed a larger condensate rate will obviously depend on the groove lead angle γ, through the groove length between the upper and the lower part of the tube. Thus $\cos\gamma$, the ratio of the tube circumference to groove length per urn, will be a parameter related to E_o. Grooves with small lead angles (nearly vertical grooves for horizontal tubes) will presumably be most effective for drainage For this reason

(5) $$E_o=E_o\,(\mathrm{We},\ \cos\gamma).$$

To obtain data for steam-side and in-tube heat transfer coefficients of corrugated tubes an experimental program, including 25 tubes, was undertaken. A comprehensive description of the test program and geometrical parameters of all tubes studied can be found in [11]. Here, the results for the steam-side heat transfer coefficients will be discussed in brief. Since the tube wall temperature was measured, the steam-side theat transfer coefficients were determined from

(6) $$Q=h_o\,A_o\,(T_s-T_w),$$

where h_o, W/(m^2.K) is the heat transfer coefficient; Q, W is the heat flux; A_o, m^2 is the heat transfer surface; T_s and T_w, K are the steam and average tube wall temperatures. Smooth tube heat transfer coefficients obtained from the experiments were compared to those resulting from Nusselt's equation

(7) $$\frac{h_o}{\left\{\frac{\lambda_f^3\rho^2 g}{\mu_f^2}\right\}^{1/3}}=h_o^+=1.51\ (4\Gamma/\mu_f)^{-1/3}.$$

Here Γ, kg/(s. m) is the condensate mass flow rate per unit length of the tube; λ_f, W/(mK), and μ_f, kg/(ms) are the thermal conductivity, and dynamic viscosity of the condensing film. This comparison revealed that the heat transfer coefficients calculated from eq. (7) underestimate with up to 40% those measured in the experiments. But if one takes into account the fact that generally in the research practice results higher up to 35% than those predicted from eq. (7) have been reported, and that the changes of the complex $4\Gamma/\mu_f$ were within quite narrow limits (31.3-39.5), these results can be considered acceptable. Moreover eq. (7) has been derived assuming several simplifications, among others a zero steam velocity. The steam-side heat transfer coefficient h_o was determined simultaneously with the internal heat transfer coefficient h_i, which depends on the Reynolds and Prandtl numbers (Re and Pr). For this reason the variations of h_o^+ in terms of Re are shown in Fig. 2 ($\mathrm{Re}=u_m.D_i/\nu$, where u_m, m/s is the mean fluid velocity; D_i, m is the internal diameter of the tube and ν, m^2/s is the kinematic viscosity of the fluid). From Fig. 2 one can determine the magnitude

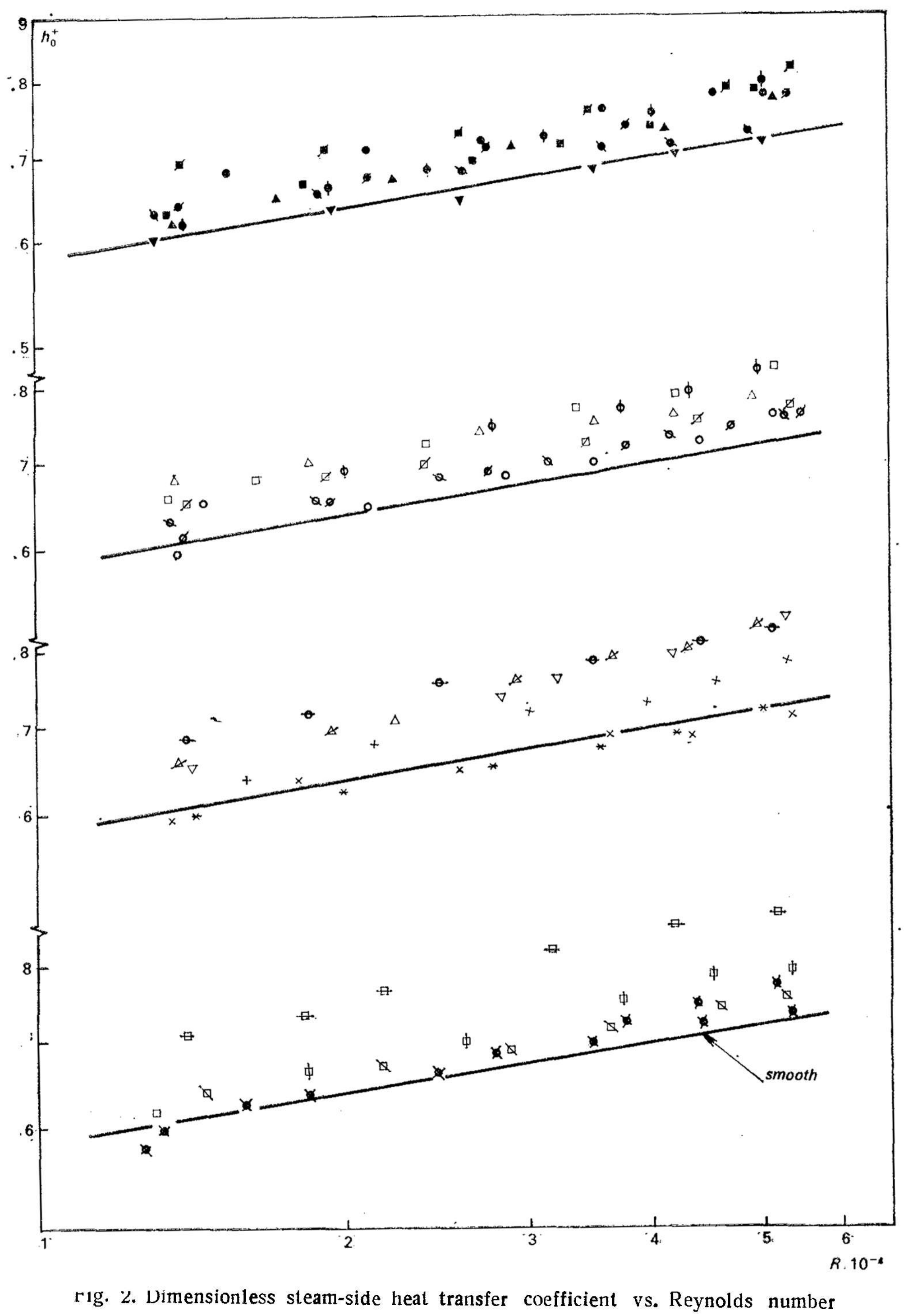

Fig. 2. Dimensionless steam-side heat transfer coefficient vs. Reynolds number

of the dimensionless steam-side heat transfer coefficient of each corrugated tube and compare it with the one of the smooth tube. Thus the enhancement factor E_o can be defined as

(8) $$E_o = h^+_{o,r}/h^+_{o,s},$$

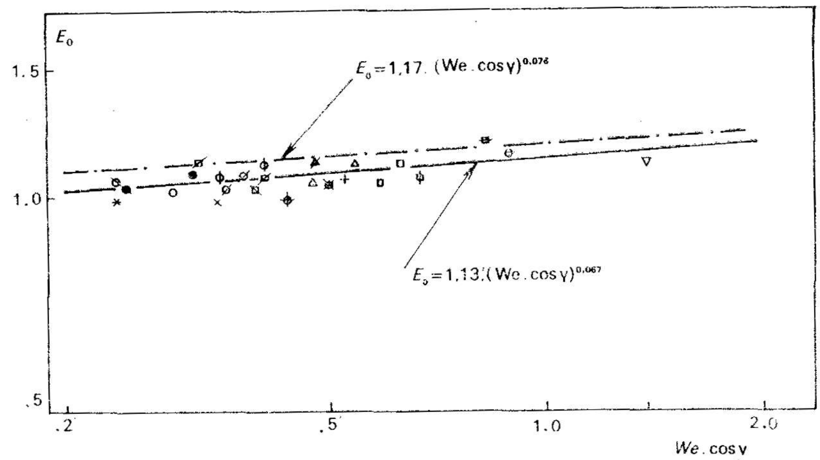

Fig. 3. Steam-side enhancement factor vs. (We. cosγ)

where the actual roped surface result is being divided by the actual smooth surface result in identical circumstances. (The subscripts r and s mean roped and smooth surface.) A simple relation for eq. (5) was found by linear regression analysis and the result is shown in Fig. 3. The experimental data could be fitted adequately by the correlation

(9) $$E_o = 1.13(\mathrm{We} \cdot \cos\gamma)^{0.067}.$$

For comparison Cathpole's relation [9] is given in Fig. 3 which yields slightly higher values of E_o. This comparison must be considered with a stipulation since the Weber number has been determined differently in this paper. Equation (9) shows that when We and cosγ increase, E_o also increases. An increase of We and cosγ might gain deeper grooves and smaller pitch between them. The larger condensate rate in the deeper grooves might nullify the benefit of reduced thermal resistance on the convex parts of the tube unless the surface has good drainage characteristics. In this case it seems to be more attractive when the corrugated tube surface has higher drainage capabilities than improved surface tension forces. This requires shallow and closely spaced grooves. The roped tubes manufactured by YIM Alloys, Leeds [8] apper very attractive from this point of view.

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