EFFECTS OF BLADE SHAPE ON HEAT TRANSFER AND RESISTANCE FOR A SPIRAL FLOW IN A TUBE

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Theoretical and experimental studies have been made on the effects of mass forces on heat transfer and hydraulic resistance in pipes with local turbulence induced by spiral blade systems having shapes defined by $u\bar{r}/(m - \bar{r}) = \text{const}$.

Heat transfer may be accelerated on employing a spiral flow; in such a flow, the mode of motion and the interactions with the walls are substantially affected by centrifugal forces, which can perturb the flow and produce circulation or else may stabilize it and suppress random radial perturbations produced by pressure forces. In the first case, the mass forces tend to increase the turbulence, while in the second case they have a conservative effect.

This distinction between the two types of effect in a flow is a particular case of the general problem of stability of motion, for which Rayleigh's method can be applied. In such motion, these forces will amplify radial perturbations if the following condition is met [1]:

$$\frac{1}{(ur)^2}\frac{d(ur)^2}{dr} + \frac{1}{\rho}\frac{d\rho}{dr} < 0. \tag{1}$$

We see from (1) that the effects are determined by the distribution of the circumferential velocities and of the temperatures (densities) in the cross section. If the spiral motion is local, one can obtain any required variation in the circumferential component by suitable blade design.

Studies have been made [3, 8] of the effects of uneven force distribution over the cross section on the heat transfer and hydraulic resistance for blade systems having shapes defined by $ur^n = \text{const}$; in that case, the heat transfer is accelerated on account of the increased speed of the liquid with respect to the wall, in addition to effects from secondary flows.

Consider a system whose blades have the form defined by

$$\frac{u\bar{r}}{m - \bar{r}} = \text{const}. \tag{2}$$

If the axial velocity has a uniform distribution ahead of the device, the variation in the angle of attack of the blade is given by

$$\operatorname{tg}\varphi = \frac{m - \bar{r}}{\bar{r}(2m - 1)} \operatorname{tg}\varphi_{\text{av}}. \tag{3}$$

We see from (3) that such a device has the distinctive feature that the angle φ is zero for $m = 1$, and hence the same applies for the circumferential component of the velocity, in which case the extent of improvement in the heat transfer will be determined by the magnitude and direction of the excess mass force.

From (2) we see that the first term in (1) is

$$\frac{1}{(ur)^2}\frac{d(ur)^2}{\mathrm{dr}} = \frac{2}{r - mR}. \tag{4}$$

The second term in (1) is governed by the distribution of the density over the radius, which is related to the

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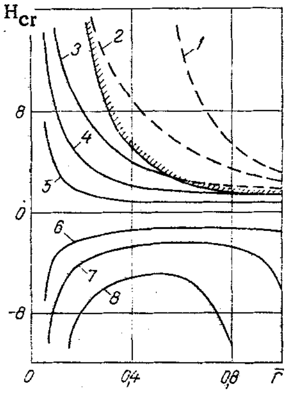


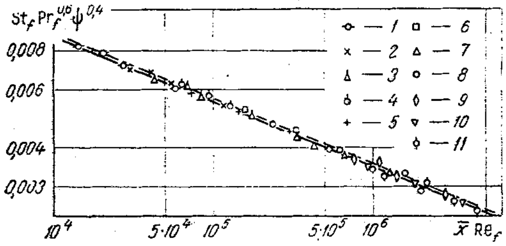


Fig. 1 **Fig. 2**

Fig. 1. H_{cr} as a function of $\bar{r}$ and m for blades with the shapes of (2): 1) m = 0; 2) 0.2; 3) 0.5; 4) 1; 5) 2; 6) −2; 7) −1; 8) −0.5.

Fig. 2. Comparison of measurements on heat transfer with calculations from (11): 1) $\bar{x}$ = 1; 2) 2; 3) 3; 4) 4; 5) 5; 6) 10; 7) 20; 8) 30; 9) 40; 10) 50; 11) 60. The solid line is derived from (11) and the dashed one, from the formula of [8].

temperature via the bulk thermal-expansion coefficient β as follows:

$$\rho = \frac{\rho_0}{1+\beta(\theta_0-\theta)}. \tag{5}$$

After the thermal-stabilization section, the radial temperature distribution can be represented by means of a polynomial of third degree [2]:

$$\theta = \theta_0\left[\frac{6}{5}(1-\bar{r}) + \frac{3}{5}(1-\bar{r})^2 - \frac{4}{5}(1-\bar{r})^3\right]. \tag{6}$$

Then (5) and (6) allow one to define the second term in (1):

$$\frac{1}{\rho}\frac{d\rho}{dr} = \frac{\frac{H}{R}A}{1+\frac{H}{6}B}. \tag{7}$$

Here $A = 2 - \bar{r} - 2(1-\bar{r})^2$ and $B = 1 - 6\bar{r} + 3(1-\bar{r})^2 - 4(1-\bar{r})^3$; then the activity condition for the mass forces is put as follows on the basis of (4) and (7):

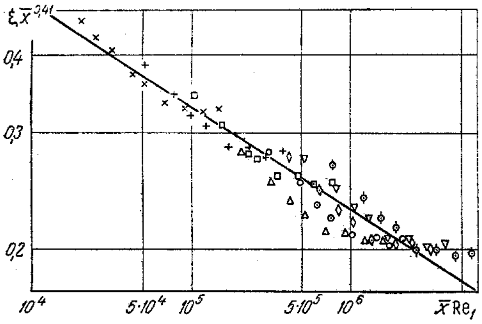
$$\frac{2}{\bar{r}-m} + \frac{AH}{1+\frac{H}{6}B} < 0. \tag{8}$$

Then (8) allows us to find the critical value of H, which defines the boundary between the active and conservative regions:

$$H_{cr} = \frac{2}{A(m-\bar{r}) - \frac{B}{3}}. \tag{9}$$

The only such systems that are physically possible are ones in which the signs of the heat flux and the density gradient are the same, which is so if [1 + HB/6 > 0], in which case (8) and (9) imply that for $H < H_{cr}$ the mass forces have the first type of action, whereas for $H > H_{cr}$ they have a conservative effect.

Figure 1 shows $H_{cr} = f(\bar{r}, m)$ for devices having blades following the law of (2); the line corresponding to the division between the realistic and unrealistic cases has been derived from the condition [1 + HB/6 = 0]. The hatching denotes the region of unrealistic conditions, and it is clear that on heating the liquid (H < 0) the mass forces will have a tendency to accentuate perturbations throughout the flow for m ≥ 1. If m is negative, the numerical values determine whether the amplification occurs near the wall or some distance away from it. Under cooling conditions, namely, H > 0, the mass forces accentuate effects if m ≥ 1.



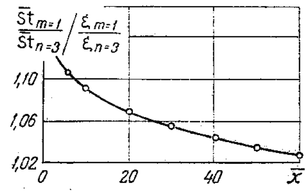


Fig. 3 Fig. 4

Fig. 3. Comparison of measurements on hydraulic resistance with calculations from (13) (symbols as in Fig. 2).

Fig. 4. $(\bar{St}_{m=1}/\bar{St}_{n=3})/(\xi_{m=1}/\xi_{n=3}) = f(\bar{x})$.

If the device is installed at the start of the tube, with no thermal stabilization section preceding it, then the second term in (1) becomes zero, and the mass forces always accentuate the effect for $m \geq 1$.

The economic desirability of such a device with this style of blade in accelerating heat transfer has been evaluated by means of measurements on heat transfer and hydraulic resistance in the initial part of a tube with a device having blades in accordance with (2) and $m = 1$; the device was a bronze ring of clear diameter 32.5 mm whose inner surface bore blades. The radial size b of a blade was equal to half the radius of the tube. The blades were placed around the arc of a circle. In the section of radius $r_b = R - b$, the liquid flowed through axially without induced spiral motion.

The tests were performed with the equipment described in [3].

The experiments on the local heat-transfer coefficient were processed with a Ural-2 computer to give $St = f(Re_c^{**})$ curves, which represent the heat transfer in the initial section under these conditions. The method of processing the data has previously been described [8]. The following form applies to ±6% for the correlation in the heat transfer for $Re_t^{**} = 10^2\text{-}4 \cdot 10^3$:

$$St = \frac{0.0269}{Re_t^{**0.25}\, Pr^{0.75}}. \tag{10}$$

Local simulation [4-7] was used to draw the general curve from the measurements; the known heat-transfer law of (10) was used with the solution to the integral equation for the thermal boundary layer [4] with q_w = constant to derive the general formula for the local heat-transfer coefficient at the start in the presence of this device:

$$St = \frac{0.0554}{Re_f^{0.2}\, Pr_f^{0.6}\, \psi^{0.4}\, \bar{x}^{0.2}}. \tag{11}$$

In (11), we have incorporated the increase in the speed at the core of the subsonic flow in accordance with the following equation [4]:

$$Re_f = Re_{1f} + 5.2\,\psi\, Re_t^{**}. \tag{12}$$

Figure 2 shows calculations from (11) together with measurement data; the maximum deviation of the measurements from the line does not exceed 8% throughout the range $\bar{x} = 1\text{-}60$; the dashed line here corresponds to the formula from [8] for local heat transfer with a blade having the form ur^n = const for n = 3 and $\varphi_{r=rb}$ = idem; it is clear that the circumferential velocity at the surface after the device falls ultimately to zero, which in the presence of identical circumferential velocities at radius r_b results in a slight fall in the local heat-transfer coefficient (not more than 3-4%) throughout the ranges in Reynolds number and in the relative tube length.

The spiral flow increases the hydraulic resistance, and the general formula for this for tubes of relative length $\bar{x} = 2\text{-}60$ with $Re_f = 10^4\text{-}9 \cdot 10^4$ for the case m=1 takes the form

$$\xi = \frac{2.08}{Re_{1f}^{0.16}\, \bar{x}^{0.57}}. \tag{13}$$

The solid line in Fig. 3 corresponds to (13); most of the measurements fall close to this line, the deviations not exceeding 10%.

Figure 4 shows the ratio of the mean heat-transfer coefficients for devices whose blades are in accordance with (2) with $m = 1$ and $ur^n = \text{const}$ with $n = 3$, the ratios being to the hydraulic resistance for a plain tube under the same conditions. It is clear that for $G = \text{idem}$, the increase in the hydraulic resistance with length for $m = 1$ is more marked than the change in the heat-transfer coefficient for the device with $n = 3$.

We examined the performance of this device and found that the mean heat-transfer coefficients for $\bar{x} = 60$ were higher than the corresponding ones for plain tubes by 35% for a given pumping power. This improvement by comparison with an axial flow is obtained with much lower speeds for the fluid.

NOTATION

c_p, specific heat at constant pressure, J/kg·deg; d, tube diameter, m; $H = \beta Rq/\lambda$; q, heat load on heat-transfer surface, W/m^2; Pr, Prandtl number; r, radius, m; R, internal radius of tube, m; $\bar{r} = r/R$; $Re_t^{**} = (\rho_0 w_0 \delta_t^{**}/\mu_w)$, Reynolds number based on energy loss thickness; $Re_f = w_0 d\rho_0/\mu_0$, Reynolds number; $St = [q_w/\rho_0 w_0 c_{p_0}(T_0 - T_w)]$, Stanton number; T, temperature, °K; u, peripheral speed, m/sec; w, axial speed, m/sec; $\bar{x} = x/d$, relative distance from inlet; β, volumetric expansion coefficient, 1/deg; δ_t^{**}, energy loss thickness, m; $\theta = T_w - T$; $\theta_0 = T_w - T_0$; λ, thermal conductivity, W/m·deg; ξ, hydraulic resistance coefficient; ρ, density, kg/m^3; φ, angle between velocity vector and axis, deg; ψ, temperature factor. Indices: 0, axis; 1, outlet; f, heating agent; w, wall; t, thermal.

LITERATURE CITED

1. V. K. Shchukin, Heat Transfer and Hydrodynamics of Internal Flows in Response to Mass Forces [in Russian], Mashinostroenie, Moscow (1970).
2. E. R. G. Eckert and R. M. Drake, Jr., Heat and Mass Transfer, 2nd ed., McGraw-Hill (1959).
3. A. F. Koval'nogov and V. K. Shchukin, Inzh.-Fiz. Zh., 14, No. 2 (1968).
4. S. S. Kutateladze and A. I. Leont'ev, Heat and Mass Transfer and Friction in a Turbulent Boundary Layer [in Russian], Énergiya, Moscow (1972).
5. A. I. Leont'ev and V. K. Fedorov, Inzh.-Fiz. Zh., 4, No. 8 (1961).
6. P. N. Romanenko and N. V. Krylova, Inzh.-Fiz. Zh., 7, No. 4 (1964).
7. V. M. Buznik, Z. N. Geller, and A. K. Pimenov, Inzh.-Fiz. Zh., 16, No. 4 (1969).
8. A. F. Koval'nogov, V. K. Shchukin, and Yu. F. Gortyshev, Izv. Akad. Nauk SSSR, Énerget. Transport, No. 3 (1973).