HEAT TRANSFER AND HYDRAULIC RESISTANCE IN TUBES WITH BLADED SWIRLS

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Results from an experimental investigation of the heat transfer and hydraulic resistance in tubes with bladed swirls are examined; the swirls are unique in terms of the flow-twist angles which they produce and in terms of the radial variations in these angles.

Experimental study of twisted flows has demonstrated that bladed swirls are capable of intensifying heat transfer [1–3]. However, many aspects of this problem require additional investigation. We will deal with two of these here—the effect of tube length and the radial variation of circumferential speed on heat transfer and hydraulic resistance.

The use of a bladed swirl to twist an incompressible fluid may alter the coefficients of heat transfer and hydraulic resistance in two ways. First of all, because of the appearance of a circumferential component there is an increase in the speed of liquid motion relative to the wall. Secondly, centrifugal forces are generated, and these may affect the nature of fluid motion and the conditions of its interaction with the wall.

The circumferential component of velocity in the fluid layer at the wall increases with distance from the tube wall, thus setting up conditions favorable for the generation of Taylor-Goertler eddies. The distribution of the circumferential velocities in the remaining portion of the flow is a function of the variation in the radial blade-twist angle. Profiling of the swirl blades permits variation of the radial twist angle so as to enhance the appearance of secondary flows throughout.

Use of the Rayleigh method to analyze the motion of fluid particles over a curvilinear trajectory permits us to draw the conclusion that the mass forces will intensify the random radial displacements when

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

With the circumferential velocity an exponential function of the radial coordinate

$$ur^n=\mathrm{const} \qquad (2)$$

Inequality (1), for $n>1$, is satisfied under isothermal conditions. (The problem of a nonisothermally twisted flow is covered in [4].) It is to be expected that the mass forces will enhance intensification of heat transfer in the isothermal case.

With a uniform field of axial velocities in front of the swirling device, condition (2) is assured when the blades are profiled as per the law

$$\operatorname{tg}\varphi=\left(\frac{r_0}{r}\right)^n\operatorname{tg}\varphi_0. \qquad (3)$$

Heat transfer and hydraulic resistance were investigated for $n = 0$ and five flow-twist angles (φ = 15, 30, 45, 60, and 75°), as well as for φ = 45° and four blade-twisting laws (n = −1, 0, 1, and 3).

The swirling devices were made of bronze rings with a clearance diameter of 32.5 mm; eight blades made of brass sheets 0.5 mm in thickness were welded to the inside surfaces of each of the rings. The radial and axial dimensions of the blades are equal to half the tube radius. The blades were profiled to match the arc of a circle. When $n = 0$, the blades were identical in width throughout their entire height, and φ = const. When $n \neq 0$, the blades varied vertically in width, to ensure the required radial twisting law.

The gradient method is used to study heat transfer, since this makes it possible to determine the relationship between the heat-transfer coefficient and tube length. The flow of heat needed to determine the heat-transfer coefficient is found in this method from the temperature gradient in a wall whose temperature field is evaluated by measuring the temperatures over the contour of the longitudinal cross section of the tube. The analytical foundations of this method for a tube are presented in [5], while the method for the reduction of the experimental data by means of computers and the procedures for evaluating the reliability of the results are covered in [6].

The investigation was carried out in a test stand whose basic diagram is presented in Fig. 1. Distilled water is passed from a constant-level tank 13 through copper washer 10, stabilizer 15, inlet chamber 7, reaching the test section 1 at whose inlet the bladed swirl chamber 6 is positioned. Water enters funnel 21 from the test section to reach measuring tank 20 from which centrifugal pump 17 transfers the water to the constant-level tank 13, first cooling it in refrigerator 16. Valve 25 is used to regulate the flow rate.

The basic element of the test unit is a thick-walled tube with an outside diameter of 58.1 mm, an inside diameter of 32.5 mm, and a length of 2000 mm; it is made of 1Kh18N9T stainless steel. For technological reasons, the tube is made up of two segments of one-meter length, with each segment demountable along the axis.

The temperature at the outside and inside surfaces was measured by means of nichrome-constantan thermocouples positioned lengthwise at seventeen points, and at 5 points on the end surfaces. The hot junctions of the thermocouples were welded to the tube surfaces by spot welding. The thermocouple leads were carried in annular grooves 0.4 mm in depth and 0.3 mm in width; these were then covered with an epoxy

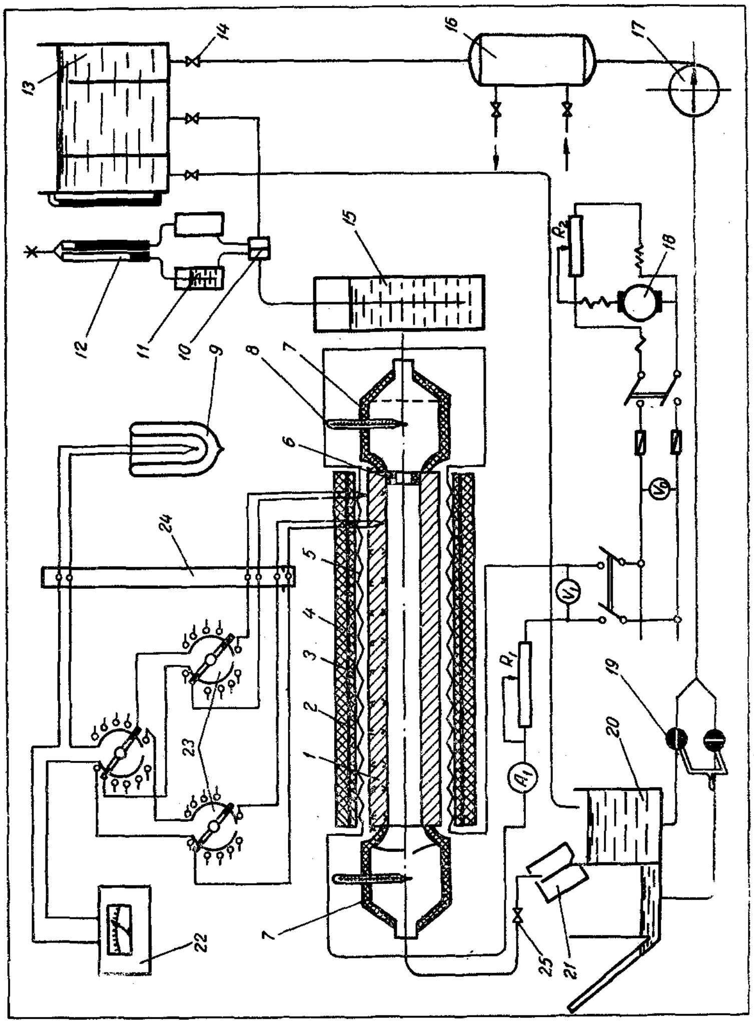

Fig. 1. Experimental installation: 1) testing tube; 2) casing; 3) ceramic inserts; 4) fibrglas; 5) heater; 6) swirl; 7) chamber; 8) thermometer; 9) vacuum flask; 10) measuring washer; 11) damper; 12) piezometer; 13) tank with stationary level; 14) tap; 15) stabilizer; 16) refrigerator; 17) centrifugal pump; 18) dc generator; 19) tap; 20) measuring tank; 21) funnel; 22) potentiometer P 2/1; 23) switchers; 24) copper unit; 25) valve.

resin. The points at which the thermocouple junctions were located were first thoroughly cleaned. The thermocouple leads were passed through two longitudinal grooves milled into the surface at which the tube separated.

After the thermocouple wiring was positioned, the tube was joined with an epoxy glue. The thermocouple leads were extended to a large copper block 24 housed in a thermostat, and they were connected to switches 23 of the semiautomatic P 2/1 potentiometer by means of copper wiring.

The testing section was housed in casing 2 to which ceramic inserts 3 were attached. For ease in assembly, the casing and the inserts were cut along their axes. The two sections of the tube were joined by means of a transitional coupling. The coaxiality of the testing section relative to the inside surfaces of the inserts was ensured by means of adjustable reference screws.

The heat flow is produced by a nichrome-strip radiator 10 mm in width and 1 mm in thickness; this strip is mounted on the inside surface of the insert. The heater leads are connected to the busbars of the dc generator 18.

To set the device for a specific water-flow regime, provision is made for a metering washer 10. The water flow rate is measured volumetrically by means of a calibrated metering compartment in tank 20.

The average liquid temperature at the inlet to the testing section was measured by means of a mercury thermometer graduated in 0.1° C.

The thermal conductivity of the tube material was determined experimentally on a special device. The results from the experimental determination of λ over the range of operating temperatures coincide to within 1% with the Neymark formula [7] which, for the given chemical composition of the tube material, has the form

$$\lambda = 12.49 + 0.0119t. \tag{4}$$

The hydraulic resistance to twisted flow was studied on an installation whose test section consisted of a tube with an inside diameter of 32.5 mm, a length of 2750 mm, fabricated of plastic.

The total pressure was measured at 6 points along the length by means of total-head tubes. The direction of the velocity vector for the flow was determined by means of attaching a silk thread to the rear portion of the metering tube and this was tested against the mag-

Table 1

Tube-length Correction Factor for the Coefficient of Hydraulic Resistance

	l/d						
	5	10	20	30	40	50	60
$\varphi°$	$n=0$						
15	1.95	1.55	1.15	1.10	1.05	1.03	1.00
30	2.25	1.80	1.30	1.17	1.10	1.05	1.00
45	2.45	1.95	1.45	1.25	1.17	1.09	1.00
60	3.50	2.50	1.65	1.40	1.25	1.12	1.00
75	5.20	3.30	1.95	1.55	1.35	1.20	1.00
n	$\varphi = 45°$						
−1	1.70	1.45	1.20	1.12	1.05	1.00	1.00
1	2.80	1.95	1.45	1.30	1.20	1.07	1.00
3	4.50	2.95	1.80	1.50	1.30	1.15	1.00

nitude of the pressure drop (the magnitude of the pressure drop reaches its maximum when the axis of the orifice in the metering tube coincides with the direction of the velocity vector). The total pressure of the liquid was measured in each section at five points positioned at the centers of equal areas. The average value of the total pressure in a section was calculated with the formula

$$\bar{p}^* = \frac{1}{5}\sum_{i=1}^{5} p_i^*. \tag{5}$$

The hydraulic and thermal tests of a testing section without a swirl chamber when Re = 10^4–$9\cdot10^4$ demonstrated that the coefficients of hydraulic resistance and heat transfer derived in these tests for the entire working section coincide, to within ± 3%, with the familiar Blasius and Mikheev formulas.

All of the measurements were carried out only after the installation was brought to the steady-state regime.

The experimental heat-transfer data were processed on a Ural-2 computer. The heat-transfer coefficients α were calculated for 16 values of l/d. It took 3 min to process one experiment.

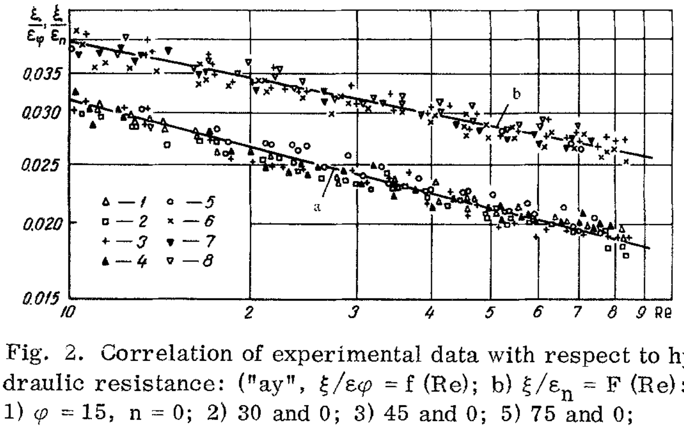

Fig. 2. Correlation of experimental data with respect to hydraulic resistance: ("ay", $\xi/\varepsilon\varphi$ = f (Re); b) ξ/ε_n = F (Re): 1) φ = 15, n = 0; 2) 30 and 0; 3) 45 and 0; 5) 75 and 0; 6) 45 and −1; 7) 45 and 1; 8) 45 and 3.

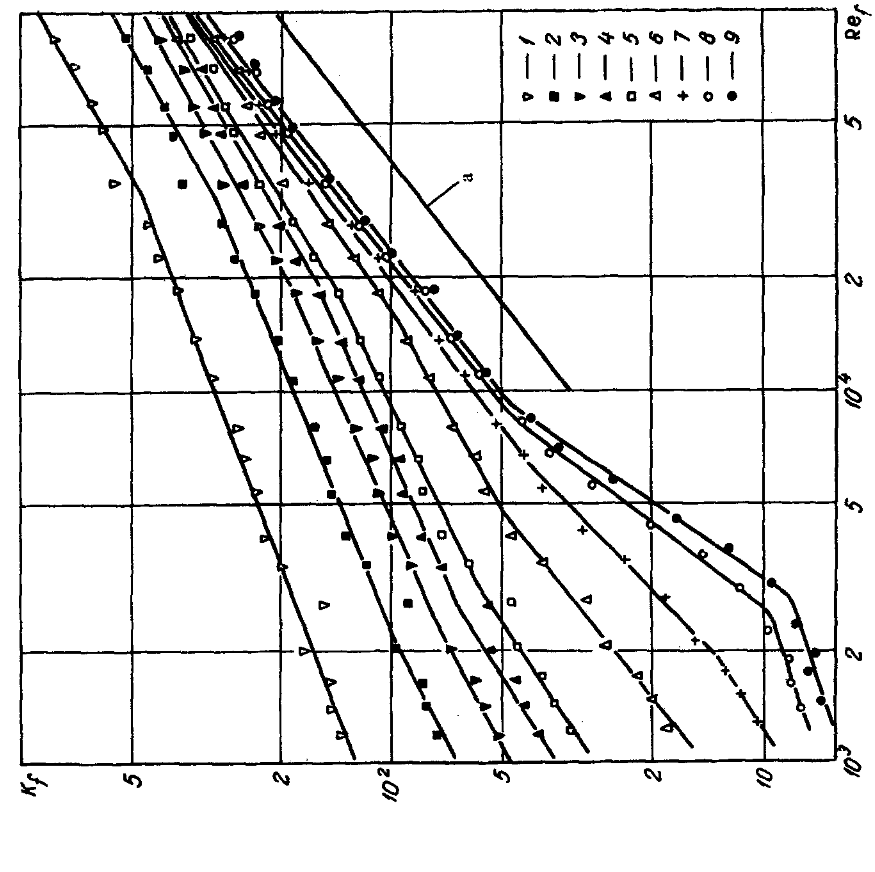

Fig. 3. Effect of Reynolds number Re on the mean heat-transfer coefficient in tube with blade-type swirl at φ = 75°, n = 0: 1-9 at l/d = 1, 2, 3, 4, 5, 10, 20, 40, and 60, respectively; a tube without swirl.

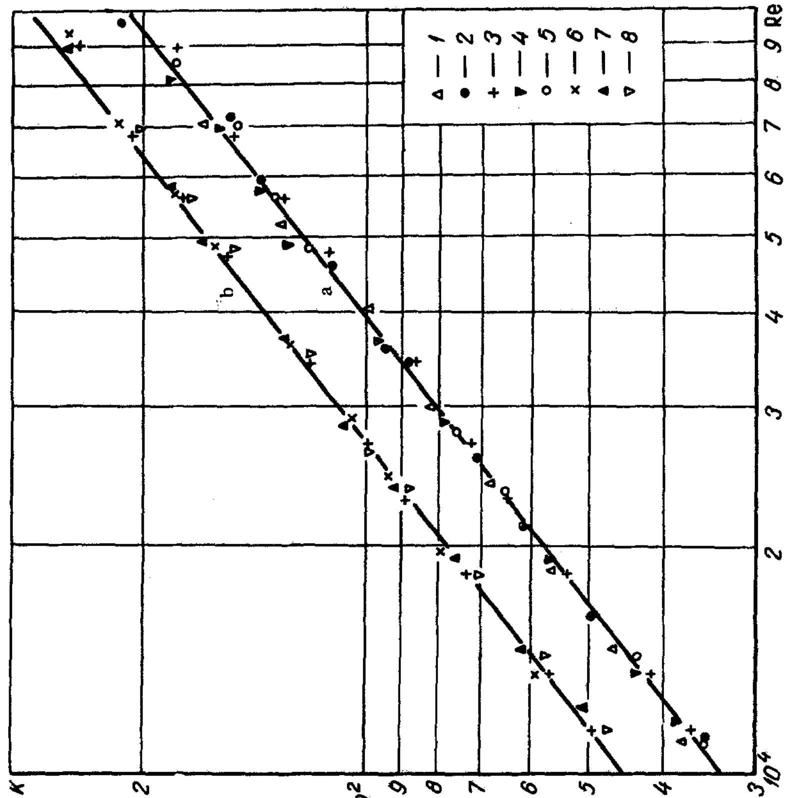

Fig. 4. Generalization of experimental data on heat transfer: a) Kφ = f (Re); b) Kn = F (Re); 1-8 see Fig. 2.

With $n = 0$, $\varphi = 15–75°$, and $Re = 1–9 \cdot 10^4$, the experimental data on the hydraulic resistance for tubes with $l/d = 68.3–75.8$ are generalized by the formula

$$\xi = \frac{0.3164}{Re^{0.25}} (1 - 0.13\,\bar{\varphi}^{0.65}) Re^{0.02\bar{\varphi}}. \tag{6}$$

The line $\xi/\varepsilon_\varphi = f(Re)$ in Fig. 2 corresponds to this formula. Here we have denoted

$$\varepsilon_\varphi = (1 - 0.13\bar{\varphi}^{0.65}) Re^{0.02\bar{\varphi}}. \tag{7}$$

For a flow-twist angle of 45° near the tube surface, with n ranging from -1 to 3, and with $Re = 1–9 \cdot 10^4$, the experimental hydraulic-resistance data for long tubes are described by the equation

$$\xi = \frac{0.2265}{Re^{0.19}} (1 - 0.02\,n) Re^{0.01n}. \tag{8}$$

The line $\xi/\varepsilon_n = F(Re)$ in Fig. 2 is plotted according to this equation. Here we have denoted

$$\varepsilon_n = (1 - 0.02\,n) Re^{0.01n}. \tag{9}$$

The greatest deviation in the experimental points from functions (6) and (8) does not exceed 8%.

With local twisting of the flow, the circumferential component of the velocity diminishes with increasing distance of the liquid from the swirl chamber, so that a reduction in tube length involves a substantial increase in the coefficient of hydraulic resistance. The coefficient of hydraulic resistance for a short tube can be estimated by means of the correction factor ψ_l

$$\xi_l = \xi\,\psi_l. \tag{10}$$

This correction factor is virtually independent of the Re number. Its numerical values are presented in Table 1.

The transfer of heat for all of the swirl chambers was investigated for $Re = 10^4–9 \cdot 10^4$. Figure 3 shows the results of heat-transfer tests in a tube with a swirl chamber in which $\varphi = 75°$ and $n = 0$, for nine values of l/d. The line a shows the heat transfer in a tube without a swirl chamber in the case of a turbulent flow regime.

Here

$$K_f = \frac{Nu_f}{Pr_f^{0.43} (Pr_f/Pr_w)^{0.25}}. \tag{11}$$

Substantial intensification of heat transfer is noted at the initial segment of the tube. When $l/d > 15$, the relationship between the heat-transfer coefficients for twisted and untwisted flows is a weak function of tube length and amounts to 1.35–1.50.

The results of the study of heat transfer when $n = 0$, $\varphi = 15–75°$, and $Re = 1–9 \cdot 10^4$ for long tubes ($l/d = 60$) have been generalized by the criterial equation

$$Nu_f = 0.021\,Re_f^{0.8} Pr_f^{0.43} \left(\frac{Pr_f}{Pr_w}\right)^{0.25} (1 + 0.147\bar{\varphi}^{0.82}). \tag{12}$$

In Fig. 4 the line $K_\varphi = f(Re)$ has been plotted from formula (12). Here we have denoted

$$K_\varphi = \frac{K_f}{(1 + 0.147\bar{\varphi}^{0.82})}. \tag{13}$$

The results of the investigation of heat transfer when $\varphi = 45°$ and when n is equal to -1, 0, 1, and 3

Table 2

Tube-length Correction Factor for the Heat-transfer Coefficient

		l/d: 5	10	20	30	40	50	60
$\varphi°$	Re	$n = 0$						
15	10^4	1.64	1.23	1.11	1.06	1.03	1.01	1.00
30	10^4	1.90	1.35	1.16	1.09	1.05	1.02	1.00
45	10^4	1.93	1.38	1.17	1.10	1.06	1.03	1.00
60	10^4	1.97	1.43	1.18	1.10	1.06	1.03	1.00
75	10^4	2.03	1.50	1.20	1.12	1.06	1.03	1.00
n		$\varphi = 45°$						
−1	10^4	1.65	1.32	1.16	1.09	1.05	1.02	1.00
1	10^4	1.80	1.42	1.18	1.10	1.05	1.02	1.00
3	10^4	2.00	1.48	1.20	1.12	1.06	1.03	1.00
		$n = -1–3$; $\varphi = 15–75°$						
	$9 \cdot 10^4$	1.32	1.16	1.10	1.06	1.04	1.02	1.00

have been generalized for long tubes in the same range of variations in the Re number by the criterial equation

$$Nu_f = 0.0286\,Re_f^{0.8}\,Pr_f^{0.43} \left(\frac{Pr_f}{Pr_w}\right)^{0.25} (1 + 0.04\,n). \tag{14}$$

On the basis of this formula we have plotted the function $K_n = F(Re)$ in Fig. 4, and here

$$K_n = \frac{K_f}{(1 + 0.04n)}. \tag{15}$$

The greatest deviation in the experimental points from the relationships corresponding to (12) and (14) does not exceed 6%.

The values of the Nu numbers for short tubes—calculated from (12) and (14)—must be corrected by means of the correction factor ε_l which is given in Table 2 for two values of the Re number. When $Re = 9 \cdot 10^4$, the correction factor ε_l was independent of both φ and n.

For tubes with bladed swirls, the value of ε_l is greater than for tubes without flow twisting. This difference is particularly noticeable when $Re = 10^4$, for which the ε_l increases as φ and n increase. As the value of the Re number increases, there is a drop in the magnitude of the correction factor ε_l for both twisted and untwisted flows.

An analysis of the derived results shows that with a constant flow-twist angle near the heat-transfer surface an increase in n leads to an increase in the coefficients of heat transfer and hydraulic resistance. At the same time, for identical coefficients α, swirl chambers with $n > 0$ are characterized by a somewhat lower hydraulic resistance than swirl chambers with $n = 0$. Thus, long tubes with swirl chambers exhibiting $\varphi = 75°$, $n = 0$ and $\varphi = 45°$, $n = 3$, when Re = idem, are characterized by identical intensity of heat transfer. the coefficient of hydraulic resistance for $Re = 1-9 \cdot 10^4$ in the case of a tube with the former swirl chamber is greater by 2.5–5% when $l/d = 60$ than in the case of the latter swirl chamber, and when $l/d = 50$, it is greater by 8–12%; when $l/d = 30$, it is greater by 10–15%.

Comparison of the thermal efficiency on the basis of the energy factor [8] shows that swirl chambers with $n = 0$, $\varphi = 15$, 30, 45, 60, and 75°, and $n = 3$, and $\varphi = 45°$ yield a gain in heat transfer with identical expenditure of power for the pumping of the coolant by 12, 15, 18, 20, 24, and 25%, respectively.

NOTATION

d is the tube diameter; l is the tube length; Nu is the Nusselt number; n is the exponent in expression (2); Pr is the Prandtl number; Re is the Reynolds number; r is the instantaneous radius; r_0 is the inside radius of the tube; u is the circumferential component of flow velocity; w is the mean axial velocity of the flow; α is the heat-transfer coefficient; ε_l tube-length correction for the heat-transfer coefficient; λ is the thermal conductivity of the fluid; ν is the kinematic viscosity coefficient; ξ is the hydraulic resistance factor; φ is the angle between the velocity vector and the tube axis; φ_0 is the angle of flow twisting near the tube surface; $\bar{\varphi} = \varphi_0/15$; ψ_l is the tube-length correction for the hydraulic-resistance factor.

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