INTENSIFICATION OF CONVECTIVE HEAT TRANSFER IN THE LOCAL SWIRLING OF AN ANOMALOUSLY VISCOUS FLUID FLOW BY AN AXIAL-FLOW BLADED SWIRLING UNIT

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The intensification of convective heat transfer in anomalously viscous media with local swirling of the flow by axial-flow bladed swirling units is investigated experimentally.

Axial-flow bladed swirling units (ABS units) are used rather extensively in present-day engineering as a device for the intensification of heat and mass transfer. Researchers have accumulated a wealth of experimental material to date on the intensification of convective heat transfer in ducts by means of such swirling units. However, all the data available in the literature [1, 2] refer to the case of turbulent flow of viscous fluids. In this connection it is evident from an analysis of the experimental data that ABS units (Fig. 1) in viscous media intensity heat transfer as much as twofold, the maximum effects being observed in the initial section of the pipe ($l/D < 40$). The reason for the enhancement of heat transfer in viscous fluids is the onset and development of secondary flows (or flows of the second kind) after the intensifier under the action of centrifugal forces [1].

Unfortunately, experimental data are totally lacking at the present time for anomalously viscous fluids, which find extremely widespread applications in modern technology.

The objective of the present study is to determine the thermohydrodynamic efficiency and optimal range of application of ABS units in long pipes from the Reynolds number and the effective viscosity of a non-Newtonian fluid.

The experiments were carried out on the experimental arrangement shown schematically in Fig. 1. The model fluid is pumped from the preliminary thermostatting tank 1 by the pump 2 through a closed loop comprising the final thermostatting heat exchanger 4 (with thermostat 3), the attenuating chamber 5, the swirling unit 6, the working section 7, and the mixing chamber 11. The fluid flow is regulated by varying the rpm of the pump 2 and by means of the valve 14. The experiments were carried out under the conditions of heating of the fluid. The working section 7 is heated by the cascaded sectional Nichrome electric heaters 9. Various thermal boundary conditions at the wall of the working section can be created by approrpriate switching of the sections and regulation of the current in each of them. The heat losses to the surrounding medium are compensated by means of an auxiliary electric heater on top of the main sections. The wall temperature is measured with a system of Chromel–Copel thermocouples 12 having a wire diameter of 0.12 mm and by means of the potentiometer 13. The temperature of the fluid at the entrance and exit of the working section is measured with the standard thermometers 10. The working section of the arrangement is a pipe made of 1Kh18N10T stainless steel $l/D = 52$. The inner surface of the pipe is finished to eighth-class purity. The ABS units installed at the entrance to the working section are made of sheet brass with a thickness of 0.3 mm. The diameter of the central stem of the units is $d = 4$ mm, and the outside diameter of the blades is $D = 19$ mm. The flow swirl angle α is varied from 15 to 75°. The pressure drop is measured between the flow cross section at the mixing chamber and a cross section beyond the limits of the ABS unit (Fig. 1). The given arrangement makes it possible in determining the coefficient of hydraulic friction to include the losses at the entrance and when the flow acquires rotational motion [1].

The swirl angle is understood to mean the angle of inclination of the blades relative to the tangent drawn to the circle formed any one plane of the cross section drawn perpendicular to the axis of the cylindrical duct between the fore and aft ends of the swirling unit.

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TABLE 1. Thermophysical Characteristics of Model Anomalously Viscous Fluids

Aqueous solution	Measurement temp., °C	ρ, kg/m³	$c_p \cdot 10^3$, J/kg °C	λ, W/m°C
0.65% Na-CMC	20	1014	3591	0,5618
	40	1011	3580	0,5615
	60	1009	3572	0,5614
	80	1006	3560	0,5611
1% Na-CMC	20	1022	3572	0,5612
	40	1017	3510	0,5610
	60	1012	3530	0,5608
	80	1008	3540	0,5606

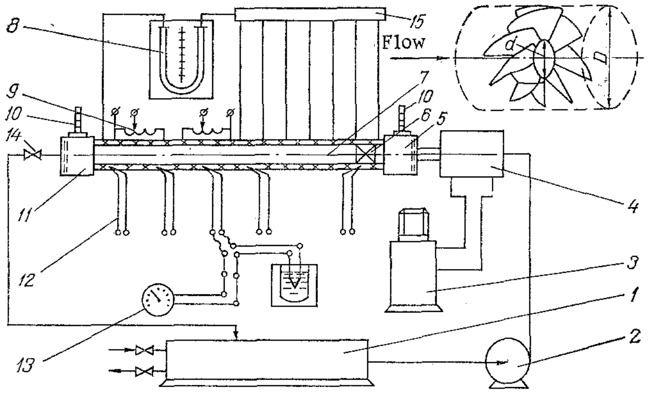

Fig. 1. Schematic view of the experimental arrangement and closeup of the axial-flow bladed swirling unit.

For the model anomalously viscous fluids we used 0.65% and 1% aqueous solutions of sodium carboxymethyl cellulose (Na-CMC). The rheological characteristics of the model fluids were determined on a Heppler viscometer and a Rheotest rotational (Couette-type) viscometer. The results of the viscometric measurements in the temperature range 20-80°C are shown in Fig. 2. The thermophysical characteristics of the model anomalously viscous fluids, determined according to [3], are summarized in Table 1.

Inasmuch as anomalously viscous fluids are characterized by a high viscosity, all of the experimental results presented here refer to the laminar flow regime.

To determine the relative increase of the heat-transfer coefficient in a pipe with an ABS unit in comparison with the values of the coefficients in the smooth pipe and to ascertain the region of Reynolds numbers where this increase is a maximum, we have processed the experimental results for the anomalously viscous fluids in the form of the function $\overline{\mathrm{Nu}}/\overline{\mathrm{Nu}}_0 = f(\mathrm{Re})$ (Fig. 3). In processing the experimental values we express the average heat-transfer coefficients in terms of the log-mean temperature difference:

$$\bar{\alpha} = \frac{Q}{F\Delta\bar{t}_{\log}} . \tag{1}$$

The temperature of the pipe wall is calculated as the weighted average over the length:

$$\bar{t}_w = \frac{\sum_{i=1}^{k} l_i (t_i + t_{i+1})}{2 \sum_{i=1}^{k} l_i} . \tag{3}$$

By analogy with a smooth pipe [1], the Reynolds numbers are calculated in terms of the flow-average velocity of the fluid. The effective viscosity of the fluid is determined for the average values of the temperature and shear stresses over the flow cross section. It is evident from Fig. 3 that the heat-transfer rate increases with the Reynolds number.

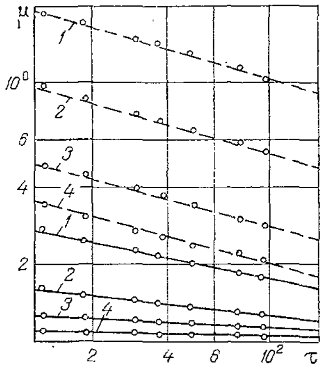

Fig. 2

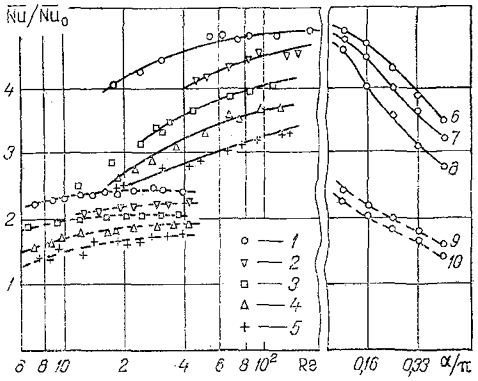

Fig. 3

Fig. 2. Results of viscometric measurements (μ, Pa · sec, versus τ, Pa) of model anomalously viscous fluids: 0.65% (solid lines) and 1% (dashed lines) Na-CMC. 1) 20°C; 2) 40; 3) 60; 4) 20.

Fig. 3. Relative increase of the heat transfer versus Reynolds number and the flow swirl angle referred to the number π: 1) α = 15°; 2) 30; 3) 45; 4) 60; 5) 75; 6) Re = 200; 7) 100; 8) 40; 9) 20; 10) 10 (solid curves for 0.65% and dashed curves for 1% Na-CMC).

Another cause of the growth of the heat transfer, mirroring the specific attributes of non-Newtonian media, is the actual anomalous dependence of the effective viscosity on the intensity of the shear stresses. The majority of anomalously viscous media (like those discussed in the present article) are characterized by pseudoplasticity, i.e., a decrease in the effective viscosity with increasing shear stress intensity. The centrifugal forces produced in swirling cause the tangential stress intensity to increase, and this in turn induces a reduction of the effective viscosity and an increase in the axial and circumferential components of the flow velocity.

Assessing the influence of the effective viscosity and degree of deviation from Newtonian behavior on the heat-transfer rate in the swirled flow of an anomalously viscous fluid, we conclude that an increase in the effective viscosity lowers the heat-transfer rate in connection with the abrupt decrease in the length of the swirl part of the flow and the decrease in the Reynolds number. On the other hand, anomalously viscous media are characterized by an increase, simultaneous with growth of the viscosity, in the degree of deviation from Newtonian behavior, including a certain (depending on the degree of anomaly) increase in the flow velocities in comparison with a Newtonian fluid having the same viscosity.

Experiments have shown that when the effective viscosity of an anomalously viscous fluid attains the order of 0.25-0.35 Pa · sec, it begins to exert a conservative action on the length of the swirl zone of the flow. Beginning with this range of the effective viscosity, the application of an ABS unit no longer affords any appreciable practical gain in the heat transfer.

It is evident from Fig. 3 that the greatest increase in the heat transfer is observed for flow swirl angles α = 15-45°, and the maximum thermal efficiency has been demonstrated by an ABS unit with α = 15°. Thus, a decrease in the angle α causes the heat-transfer rate to increase as a result of the increase in the flow velocity in the entrant section of the duct and the flow disturbances in it.

On the other hand, the onset and growth of a circumferential component of the velocity lead to a growth of the hydraulic friction and a loss of power in pumping the fluid through the duct. It is necessary in this connection to estimate the efficiency of the given intensification method. One index of the efficiency of the method is an estimate of the thermohydrodynamic efficiency [4] by means of the relation

$$(\overline{\mathrm{Nu}}/\overline{\mathrm{Nu}}_0)/(\xi/\xi_0) = f\,(\mathrm{Re}) \tag{3}$$

with the condition that ξ is determined with allowance for the swirling losses in the flow and normalization of the values of $\overline{\mathrm{Nu}}$ and Re to the same Reynolds numbers as in a smooth pipe.

Figure 4 shows the thermohydrodynamic efficiency of using an ABS unit in anomalously viscous media as a function of the Reynolds number.

TABLE 2. Correction Factors Adjusting ξ for the Length of the Pipe

$\alpha°$	l/D = 5	10	20	40	52
			0.65% Na-CMC		
15	8,35	4,85	1,90	1,10	1
30	5,10	3,30	1,75	1,05	1
45	3,95	2,85	1,40	1	1
60	3,20	2,15	1,30	1	1
75	2,90	1,90	1,20	1	1
$\alpha°$			1% Na-CMC		
15	10,20	6,45	2,65	1	1
30	7,00	5,10	1,80	1	1
45	5,25	3,90	1,20	1	1
60	3,80	2,90	1	1	1
75	3,60	2,60	1	1	1

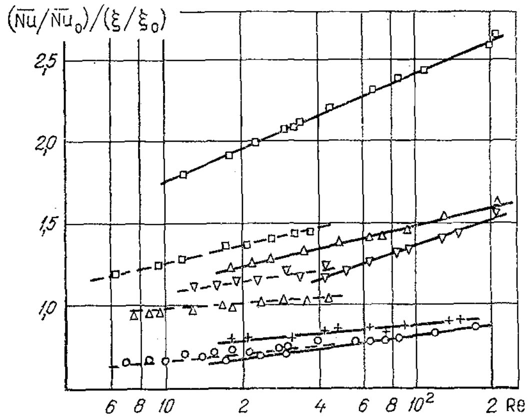

Fig. 4. Thermohydrodynamic efficiency of using axial-flow bladed swirling units to intensify heat transfer in anomalously viscous media (the nomenclature is the same as in Figs. 2 and 3).

An analysis of the curves in Fig. 4 shows that with an increase in the effective viscosity (1% Na-CMC) of the fluid the total thermohydrodynamic efficiency of the ABS unit decreases in connection with the drastic attenuation of swirling after the intensifier. The maximum increment of the heat-transfer rate over the hydrodynamic losses is threefold under the given experimental conditions and occurs in the range of effective viscosities below 0.35 [Pa · sec]. For higher values of the viscosity the ABS unit, as mentioned, does not yield a significant gain in the heat transfer with a sufficient increase in the hydraulic losses.

Examining the influence of the swirl angle on the total thermohydrodynamic efficiency of using ABS units in anomalously viscous media, we see that for $\alpha < 45°$ the thermohydrodynamic efficiency decreases in connection with the predominant growth of the hydraulic friction. It can be assumed on the basis of the experimental results that the absolute optimum from the point of view of thermohydrodynamic efficiency is a swirl angle $\alpha \approx 45°$, the efficiency increasing with the Reynolds number.

With increasing distance of the fluid from the ABS unit the circumferential component of the velocity diminishes, and so shortening the pipe causes the hydraulic friction coefficients to increase. The hydraulic friction coefficient of a short pipe (and, accordingly, the length of the swirl zone) can be estimated as in [1] by means of correction factors:

$$\xi_l = \xi\psi_l, \tag{4}$$

which are given in Table 2.

NOTATION

$\overline{\alpha}$, Average heat-transfer coefficient; F, area of heat-transfer surface of duct; $\Delta\overline{t}_{log}$, log-mean temperature difference; Q, quantity of heat; $\overline{t}_W$, average wall temperature; t_1, t_{i+1}, thermocouple readings; l_i, thermocouple spacing; k, number of thermocouples; $\overline{Nu}_0$, ξ_0, average Nusselt number and coefficient of hydraulic friction for smooth duct; $\overline{Nu}$, ξ, the same for the duct with intensifier; Re, Reynolds number; α, flow swirl angle; μ, effective viscosity of fluid; C_p, specific heat at constant pressure; λ, thermal conductivity; D, inside diameter of duct; d, diameter of central stem of intensifier; ρ, density of fluid; l, length of duct.

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COUPLED PROBLEM OF STEADY-STATE HEAT TRANSFER DURING TURBULENT FLOW OF LIQUID THROUGH A PLANE SLOT WITH DISSIPATION OF MECHANICAL ENERGY

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An analytical solution is obtained for the coupled problem of steady-state heat transfer during turbulent flow of a liquid through a plane slot with dissipation of mechanical energy.

The problem of steady-state heat transfer during turbulent flow of the coolant and the need to account for dissipation of mechanical energy in the analysis of this problem arise in practical applications [1] such as transport of crude oil under conditions of Northern climate [2].

We make the following assumptions: 1) The flow of the fluid and the heat transfer are both quasisteady; 2) the fluid is incompressible and its physical properties remain constant; 3) the change in the thermal flux density along the axis caused by heat conduction and turbulent heat transfer is small in comparison with its change in the transverse direction [1]; 4) the flow through the heat-transfer region has been hydrodynamically stabilized; 5) the temperature of the liquid and the slot wall in the entrance section is a known function of the transverse coordinate y; 6) the temperature of the outside surface of the slot is given as an arbitrary integrable function of the axial coordinate x.

The problem will be formulated as a system of equations in dimensionless variables: equation of energy for the liquid

$$W(Y)\,\frac{\partial\Theta_1}{\partial X}=\frac{\partial}{\partial Y}\left[\frac{\mathrm{Re}}{2\xi_0}\left(\frac{1}{\mathrm{Pr}}+\frac{\varepsilon_\sigma}{\nu}\right)\frac{\partial\Theta_1}{\partial Y}\right]+\frac{\mathrm{Br}}{\mathrm{Pr}}\sqrt{\frac{\xi}{8}}\left(1+\frac{\varepsilon_\sigma}{\nu}\right)\left(\frac{dW}{dY}\right)^2, \tag{1}$$
$$0<X<\infty,\ 0<Y<1,$$

equation of heat conduction for the wall

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