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ENTRANCE EFFECTS IN THE FLOW OF VISCOUS LIQUIDS IN CYLINDRICAL NOZZLES

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The pressure losses up to the entrance to nozzles in the flow of inelastic liquids in wide ranges of viscosities and flow rates are determined. An empirical equation is proposed for calculating entrance pressure losses.

The flow of a viscoelastic medium in the initial section of a cylindrical channel has been studied in numerous theoretical and experimental reports. The considerable interest in this problem is explained by its urgency for engineering application (such as the extrusion of polymer solutions and melts through short nozzles or spinners in obtaining chemical filaments), on the one hand, and by the great complexity of the observed phenomena, on the other. The unsteady flow of polymer systems is characterized by the simultaneous development of plastic and elastic deformations and the effect of thixotropic destruction of the structure; the conditions of the formation of the velocity profile at the entrance to a cylindrical channel also exert a certain effect. Whereas in a rotary viscosimeter one is able to separate the reversible and irreversible deformations experimentally, with flow in a capillary such a separation is impossible in principle, and is done by various indirect methods.

To allow for the additional energy expenditures before the nozzle entrance and in the section of unsteady flow one usually uses the so-called "inlet" correction l_{in}, which in a number of reports [1-4] is considered as a parameter of the viscoelastic behavior. However, in [5] it has been shown that the strength of the structure must create a large entrance effect, so that in the general case the possibility of using measurements of inlet corrections as a method of estimating the highly elastic properties of a system is connected with the relationship of the effects of the development of plastic reversible deformations and of destruction of the structure.

The published data relative to the quantity l_{geom}, the "geometrical" or "Couette" correction, which determines the additional energy losses due to the reorganization of the velocity profile at the nozzle entrance, are contradictory. Couette [6] found that the value of the inlet correction is equivalent to the fictitious lengthening of a capillary by five to six radii ($l_{geom} = nR$, $n = 5$–6). In [7, 9] l_{geom} was equal to 1.146R. Barr [8] took $n = 0.9$. Schurz [10] indicates that according to the literature data $n = 0.5$–1.0, but he notes in this connection that the direct determination of l_{geom} is possible only in measurements on inelastic liquids. The authors of [2, 11] assume that l_{geom} is negligibly small in comparison with l_{in} and does not depend on the discharge velocity; in [3, 12], conversely, it is noted that the energy dissipated in the entrance zone depends on the velocity gradient and can have rather large values.

And there is no single opinion on the question of where the additional pressure drop takes place: Some investigators [1, 2] assume that the entire end effect is concentrated in front of the entrance to the capillary; others [9] declare for two components characterizing the pressure drops up to the entrance to a nozzle and in its initial section.

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TABLE 1. Characteristics of Nozzles

Index	Length $L \cdot 10^3$, m	Hole radius $R \cdot 10^3$, m	Ratio L/R	Nozzle material
1001	149,3	0,466	320,5	Glass
1004	100,6	0,479	209,9	Same
1010	60,33	0,469	128,6	"
1011	28,62	0,467	61,29	"
1012	11,55	0,476	24,27	"
1013	5,188	0,469	11,07	"
1027	2,468	0,471	5,240	"
11	2,369	0,469	5,051	Stainless steel
1	0,988	0,460	2,172	Same
3	0,535	0,477	1,122	"
5	0,295	0,473	0,624	"
7	0,102	0,477	0,214	"

TABLE 2. Characteristics of Viscous Liquids

Index	Viscosity η, $N \cdot sec/m^2$	Density ρ, kg/m^2
I	0,1121	872,3
II	0,4396	873,5
III	0,9756	877,0
IV	4,184	884,5
V	11,99	873,7

The present report is devoted to an experimental study of the entrance effect in the flow of inelastic liquids in wide ranges of viscosities and flow rates.

The measurements were carried out on a GKVPD-150 constant-pressure gas capillary viscosimeter [13]. Calibrated nozzles with flat inlets were used, made in the form of glass capillaries or metal disks with holes of the same radius and different lengths (Table 1). In all the tests the temperature was held constant, 25 ±0.05°C. The relative error of the pressure measurements lay in the range of 0.5-1.5% depending on the absolute value of the applied pressure. The flow rate of liquid was determined by the weight method, with two successive batches being taken at each given pressure.

Viscous liquids prepared and certified at the Moscow Center of Measures and Standards were extruded through the nozzles. The calibration liquids consist of mixtures of oils: GOST 982-56 transformer oil and GOST 12869-67 octol-600 oil. In the certification the viscosity of the liquids was determined on standard capillary viscosimeters with an accuracy of 0.3% at a temperature of 25 ±0.01°C; the densities of the liquids were found by the pycnometric method with an accuracy of 1 kg/m^3. The characteristics of the viscous liquids are given in Table 2.

In the course of the experiment it was established that in a wide range of velocities (three orders of magnitude) the calibration liquids are characterized by a Newtonian mode of flow. A special test for the Weissenberg effect confirmed the absence of viscoelastic properties in the liquids. Consequently, the end effect in the flow of the given liquids in nozzles must be determined only by the energy dissipation connected with the jet flow and internal friction in the transfer of the liquid from the viscosimetric vessel into the orifice of the nozzle and with the formation of the velocity profile in the channel, i.e., $l_{in} = l_{geom}$.

Typical pressure—flow-rate characteristic curves for the flow of viscous liquids in nozzles of different lengths are presented in Fig. 1. Double logarithmic coordinates — the reduced volumetric flow rate $q = 4Q/\pi R^3$ and the reduced pressure drop $\Delta P/L$ — were chosen for the construction of these functions, since in the analysis of the data of experiments on thin disks containing holes (diaphragms) the question naturally arises of the correctness of calculations of the velocity gradients and shear stresses by the well-known equations for steady flow. It is seen from Fig. 1 that the results of the experiment with nozzles of L/R = 11.07-320.5 are shown satisfactorily by a common straight line having a slope of 45°, whereas the functions for the diaphragms are shifted successively toward larger pressure drops with a decrease in L/R from 5.24 to 0.214. The value of the viscosity calculated for the common straight line is 4.169 $N \cdot sec/m^2$. The good agreement between the value obtained for the viscosity and the certification data for liquid IV indicates the steady nature of the flow in nozzles of L/R = 320.5-11.07. Conversely, the location of these functions for the diaphragms (L/R = 5.24-0.214) confirms the assumption that flow with an undeveloped velocity profile occurs in short holes.

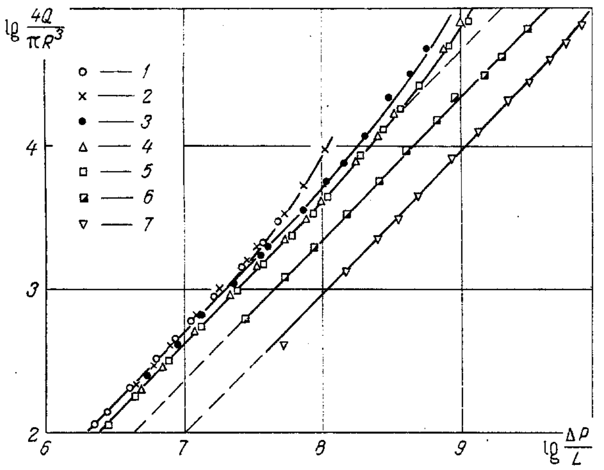

Fig. 1

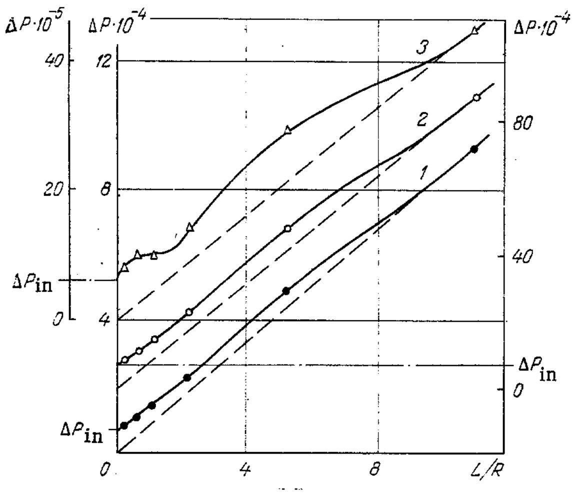

Fig. 2

Fig. 1. Pressure—flow-rate characteristic curves for the flow of liquid IV (η = 4.184 N·sec/m²) in nozzles of different lengths: 1) L/R = 320; 2) 128; 3) 11.1; 4) 5.24; 5) 5.05; 6) 0.624; 7) 0.214. $\Delta P/L$, (N/m²)/m; $4Q/\pi R^3$, sec⁻¹.

Fig. 2. Dependence of pressure drop on relative length of nozzle for flow of liquid IV in the initial section of the channel. ΔP, N/m²; L/R, units. Reduced flow rate q: 1) $1\cdot10^3$; 2) $1\cdot10^4$; 3) $1\cdot10^5$ sec⁻¹.

The "branching" of the pressure—flow-rate characteristic curves for long nozzles in the region of large flow rates is due to the thermal effect of the flow of a viscous liquid: a direct measurement of the temperature of the jet discharging from the nozzle of L/R = 320.5 at a flow rate q = $3\cdot10^3$ sec^{-1} showed a heating of 1.8°C. Allowance for the thermal effect through the known temperature coefficient of the viscosity made it possible to obtain the true viscosity in all cases.

It must also be noted that the pressure—flow-rate characteristic curves for glass and metal nozzles of the same length (L/R = 5.24 and 5.05, respectively) coincide.

The data of Fig. 1 were used to construct dependences of the pressure drop on the nozzle length by Bagley's method [1]. An analysis of the "Bagley diagrams" for nozzles of L/R = 320.5-11.07 showed that within the limits of accuracy of the graphic construction the ΔP—L/R dependences for all the liquids studied are extrapolated to the origin of coordinates and do not give an inlet correction; at the same time, the "Bagley diagrams" for short nozzles (Fig. 2) are nonlinear, and consequently their extrapolation to a zero pressure drop in order to determine the traditional inlet correction [1] is impossible.

The nonlinear trend of the ΔP—L/R dependence in the initial section of a channel for a Newtonian liquid was noted earlier [14]. An important feature of our data is the fact that they were obtained with the use of very short nozzles (L/R = 11.07-0.214); this makes it possible to reliably extrapolate curves 1-3 (Fig. 2) to a zero channel length and thereby find the final values of the pressure drop ΔP_{in} taking place up to the nozzle entrance.

From the experimental facts presented it follows that in the flow of viscous liquids in cylindrical nozzles the end effect in general is composed of the pressure losses up to the entrance of the channel and in its initial section:

$$\Delta P_{end} = \Delta P_{in} + \Delta P_{init}. \tag{1}$$

Here the quantity ΔP_{in} determines the additional expenditures of energy on jet flow and the internal friction in the flow of the liquid into the channel while ΔP_{init} determines the expenditures in the transformation of the rectangular velocity profile into the parabolic profile corresponding to steady flow. The authors of [9], who studied the flow of a viscoelastic medium (4 and 8% solutions of polyisobutylene in transformer oil), came to a similar conclusion earlier. The quantity ΔP_{init} depends on the Reynolds number, so that the problem of the allowance for ΔP_{init} in the end effect must be solved after an estimate of Re. In our experiments with q = $1\cdot10^4$ sec^{-1} the values of Re were 1.0 and 0.08 for liquids III and V, respectively, while the lengths of the sections of stabilization of the velocity profile, calculated by the equation L_{init} = 0.16R Re of S. M. Targ [15], proved to equal 0.159R and 0.013R. The examples presented show that in a number of cases in practice there is every reason to ignore ΔP_{init} and take $\Delta P_{end} = \Delta P_{in}$.

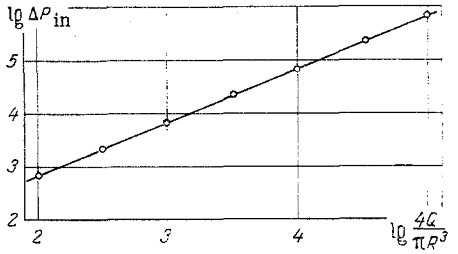

Fig. 3

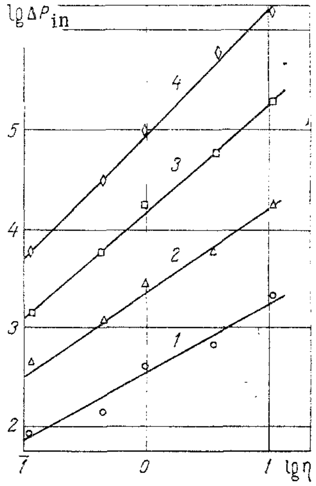

Fig. 4

Fig. 3. Dependence of inlet pressure losses on reduced flow rate in the flow of liquid IV. ΔP_{in}, N/m^2; $4Q/\pi R^3$, sec^{-1}.

Fig. 4. Dependence of inlet pressure losses on viscosity of liquid. ΔP_{in}, N/m^2; η, N·sec/m^2. Reduced flow rate q: $1 \cdot 10^2$; 2) $1 \cdot 10^3$; 3) $1 \cdot 10^4$; 4) $1 \cdot 10^5$ sec^{-1}.

For a comparison of the literature data presented above on the size of the inlet correction with our results the values of ΔP_{in} obtained were converted to l_{in} on the assumption of a linear pressure drop (Table 3). An analysis of the values of l_{in} presented in the table does not allow one to draw definite conclusions concerning the dependence of the inlet correction on the flow rate and viscosity of the liquid. Thus, the inlet effect in the flow of inelastic liquids in cylindrical nozzles can be estimated quantitatively only with the help of the pressure losses ΔP_{in} up to the entrance to the capillary.

The dependence of ΔP_{in} on the reduced flow rate was studied for five viscous liquids in the range of $q = 10^2$–10^5 sec^{-1} and in all cases it is linear in double logarithmic coordinates. A typical graph of the function $\log \Delta P_{in} - \log q$ is presented in Fig. 3 for liquid IV ($\eta = 4.184$ N·sec/m^2). The conducting of tests with a set of liquids calibrated by viscosity (two orders of magnitude) made it possible to construct dependences of the inlet losses on the viscosity with the condition q = const (Fig. 4). Here also we observe a linear dependence of the function $\log \Delta P_{in} - \log \eta$. The latter fact makes it possible to quantitatively describe this dependence by a power-law function of the type

$$\Delta P_{in} = a\eta^b, \tag{2}$$

where a is the value of ΔP_{in} intercepted by the straight line on the ordinate axis and b is the slope of the straight line. An analysis of the sizes of the coefficients a and b with variation in q from $1 \cdot 10^2$ to $1 \cdot 10^5$ sec^{-1} showed that they, in turn, are functions of the reduced flow rate:

$$\lg a = 0.78 + 0.84 \lg q \tag{3}$$

or

$$a = 6.03\, q^{0.84}; \tag{4}$$

$$b = 0.24 + 0.20 \lg q. \tag{5}$$

Consequently, the empirical dependence of the inlet pressure losses on the viscosity of the liquid and the reduced flow rate takes the form

$$\Delta P_{in} = 6.03\, q^{0.84}\, \eta^{0.24+0.20 \lg q}. \tag{6}$$

A check of Eq. (6) in the investigated ranges of viscosity (0.1–12.0 N·sec/m^2) and reduced flow rates ($1 \cdot 10^2$–$1 \cdot 10^5$ sec^{-1}) showed that the calculated values of ΔP_{in} differ from the experimental values by no more than 7%.

In conclusion, let us compare our experimental data on inlet pressure losses in the flow of viscous liquids with the results obtained by other investigators. The flow of a viscous Newtonian liquid through a fine

TABLE 3. Comparison of Inlet Pressure Losses ΔP_{in} and Inlet Corrections l_{in}

Reduced flow rate q, sec^{-1}	Liquid									
	I		II		III		IV		V	
	ΔP_{in}, N/m^2	$l_{in}=nR$	ΔP_{in}	l_{in}	ΔP_{in}	l_{in}	ΔP_{in}	l_{in}	ΔP_{in}	l_{in}
$1 \cdot 10^2$	$7,50 \cdot 10^1$	3,36	$1,20 \cdot 10^2$	1,34	$3,50 \cdot 10^2$	1,75	$6,90 \cdot 10^2$	0,83	$1,60 \cdot 10^3$	0,66
$1 \cdot 10^3$	$4,75 \cdot 10^2$	2,13	$1,08 \cdot 10^3$	1,21	$2,80 \cdot 10^3$	1,40	$6,90 \cdot 10^3$	0,83	$1,75 \cdot 10^4$	0,72
$1 \cdot 10^4$	$1,42 \cdot 10^3$	0,64	$6,00 \cdot 10^3$	0,67	$1,80 \cdot 10^4$	0,90	$6,90 \cdot 10^4$	0,83	$1,80 \cdot 10^5$	0,74
$1 \cdot 10^5$	$6,20 \cdot 10^3$	0,28	$3,00 \cdot 10^4$	0,34	$1,05 \cdot 10^5$	0,52	$6,90 \cdot 10^5$	0,83	$1,60 \cdot 10^6$	0,66

TABLE 4. Values of Coefficient C_0 in Eq. (7) Calculated from Experimental Values of ΔP_{in}

log q, sec^{-1}	Values of C_0 for liquids				
	I	II	III	IV	V
2,0	8,518*	3,475*	4,564*	2,099*	1,699
2,5	4,670*	3,113	3,630	1,925*	1,914
3,0	5,394	3,128	3,652	2,099	1,858
3,5	3,161	2,106	2,846	1,925	1,981
4,0	1,612	1,738	2,348	2,099	1,912
4,5	1,077	1,328	1,650	1,925	1,847
5,0	0,704*	0,869	1,369	2,099	1,699*

*The values of the pressure drops in nozzles 5 and 7, obtained through extrapolation of the pressure-flow rate characteristic curves (see Fig. 1, for example), were used in determining the corresponding values of ΔP_{in}.

round hole was studied theoretically and experimentally by the authors of [16]. On the basis of the Helmholtz theorem of the minimum dissipation of energy they derived an equation for the average pressure drop in a diaphragm:

$$\Delta P = C_0 \eta \frac{Q}{R^3}, \tag{7}$$

where C_0 is the "hole constant," equal to 3.310 for a plane velocity profile at the diaphragm, 1.500 for a velocity distribution proportional to the 1/2 power of the fully developed parabolic profile, and 3.825 for a parabolic velocity profile. Equation (7) was checked experimentally on viscous oil (η = 21.94 $N \cdot sec/m^2$ at 25°C) using diaphragms of R = $(1-2.5) \cdot 10^{-4}$ m, L = $3 \cdot 10^{-4}$ m, and an inlet angle of 21-35°. Values of C_0 = 1.91-1.93 were obtained in the range of shear velocities of 10-70 sec^{-1}.

The problem of the flow of a viscous Newtonian incompressible medium in the cavity of a hyperboloid of rotation is solved analytically in [17]. For the particular case of a right angle the equation derived by the authors for the pressure drop in the upper part of the hyperboloid gives the solution of the problem of flow in a round hole:

$$\Delta P = 1.5\, \eta \frac{Q}{R^3}. \tag{8}$$

Thus, if the streamlines up to the entrance to the hole are hyperbolas then the "hole constant" equals 1.5.

Using Eq. (7) we calculated the values of the "hole constant" from the experimental values of the inlet pressure losses. The results of the calculations, presented in Table 4, show that the shape of the streamlines in front of the entrance to a round hole can be arbitrary and be determined by the viscosity of the liquid and the mode of flow.

NOTATION

L, length of nozzle; R, radius of hole; Q, volumetric flow rate of liquid; $q = 4Q/\pi R^3$, reduced flow rate of liquid; ΔP, pressure drop in flow of liquid through nozzle; ΔP_{in}, pressure losses up to entrance of liquid into nozzle; ΔP_{init}, additional pressure losses in initial section of channel caused by unsteady mode of flow; ΔP_{end}, end effect in flow of liquids in cylindrical nozzles; L_{init}, length of section of stabilization of velocity profile in channel; $l_{in} = nR$, inlet correction; l_{geom}, "geometrical" or "Couette" correction; η, viscosity of liquid; ρ, density; Re, Reynolds number; C_0, "hole constant."

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ENERGY-SEPARATION EFFECT IN A GAS EJECTOR

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A new effect of inverse energy separation in a gas ejector is discovered.

In those cases when the output cross section of a constricting mixing chamber of a gas ejector is blocked to discharge or when an ejector with a cylindrical mixing chamber works against a grid with a high enough hydraulic resistance the reverse discharge of the ejecting gas from the mixing chamber through the inlet for the ejected gas can take place. Despite the fact that such modes have an important effect on the operation of a whole series of systems, including the gas ejectors of supersonic wind tunnels and the test stands of jet engines, for example, especially during start-up, insufficient attention has been paid to their investigation.

In the course of experimental studies of the indicated modes of operation of a gas ejector operating on air at a high absolute pressure we discovered a new effect of inverse energy separation in a homogeneous gas stream which is initially steady with respect to the stagnation pressures and temperatures, consisting in the fact that the different air zones moving along the ejector acquire different temperatures, with some of them

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