THE PUMPING CHARACTERISTICS OF SCREW ROTORS.
I. THE THEORY OF CALCULATION OF THE PUMPING CAPACITY OF SCREW ROTORS

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This paper summarizes the present state of the theory of calculation of the pumping capacity of screw rotors. The calculation starts from the equation for the volumetric flow rate of the flow between two unconfined plates modified by correction coefficients obtained from the relationships for the flow rate in simpler geometrical configurations to which the screw rotor may be, under certain circumstances, reduced.

Screw rotors represent major components of many equipment where it is necessary to ensure the motion of highly viscous fluids. As examples may serve screw mixers, screw pumps and screw extruders. The pumping capacity of screw rotors, working in these apparatusses, appears one of the major parameters necessary for their design.

The first paper dealing with the theoretical calculation of the pumping capacity of screw rotors was published more than sixty years ago by Rowell and Finlayson[1]. However, trully intensive interest in the calculation of screw rotors was not seen untill the fifties in connection with the development of the plastics production. The results of the papers from this area have been summarized in monographs devoted to the technology of plastics processing[2,3]. Of these particularly significant appears the work of Squires[4] devoted to the effect of the screw blade on the pumping capacity and the work of Mohr and Mallouk[5] devoted to the effect of the clearance between the screw and the barrel on the pumping capacity. From the remaining papers one has to mention first of all the paper of Booy[6] dealing with the problem of the curvature and the end effect on the pumping capacity of screws, our[7] more detailed analysis of the effect of the screw blade and, finally, Říha's paper[8], presenting a numerical technique of calculation of the pumping capacity of screw rotors. The ultimate goal of all these works was to give more precision to the theoretical equation for the calculation of the pumping capacity of screw rotors.

THEORETICAL

The Geometry of the System: Screw Rotor – Barrel

The screw rotor rotating in a stable barrel is shown in Fig. 1. The screw rotor of diameter d and length L is formed by a cylindrical root of diameter d_1 and a screw blade of axial thickness e. The screw blade is formed by a straight helicoid surface with a pitch s. Depending on the number of screw blades, threaded on the root, the screws may be classified as single threaded $(i = 1)$ or multiple threaded $(i > 1)$. The diameter of the barrel is D_t.

The neighbouring surfaces of the screw blade, the root of the screw and the internal surface of the barrel confine a channel of depth H

$$H = (1/2)(D_t - d_1) \tag{1}$$

and width W

$$W = \left(\frac{s}{i} - e\right)\cos\varphi\,, \tag{2}$$

where φ is the helix angle

$$\varphi = \operatorname{arctg}\frac{s}{2\pi r}\,. \tag{3}$$

The helix angle φ is a function of the radius r. The least helix angle has a hypothetic

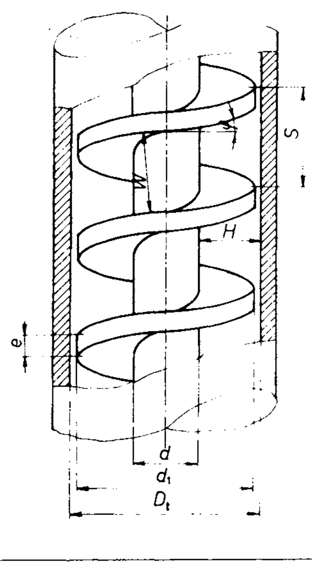

FIG. 1
Screw rotor in a stationary barrel

helicoid of the diameter D_t.

$$\varphi_t = \operatorname{arctg} \frac{s}{\pi D_t} . \tag{3a}$$

The helix angle on mean radius $d_s = 0{\cdot}5(d_1 + D_t)$ shall be designated as

$$\varphi_s = \operatorname{arctg} \frac{2s}{\pi(d_1 + D_t)} . \tag{3b}$$

From Eq. (*2*) it follows that also the width of the screw channel is a variable. For the width of the channel on the diameter D_t we may write

$$W_t = \left(\frac{s}{i} - e\right) \cos \varphi_t \tag{4a}$$

while for the mean width we have

$$W_s = \left(\frac{s}{i} - e\right) \cos \varphi_s . \tag{4b}$$

The length of the screw channel L_z may be determined from the following expression

$$L_z = L/\sin \varphi . \tag{5}$$

The geometrical variable to be introduced here is the clearance between the screw and the barrel, c

$$c = \tfrac{1}{2}(D_t - d) . \tag{6}$$

From the above considerations it follows that the geometry of the system screw rotor–barrel is relatively complex. Accordingly, solution of the flow pattern in this system in the general case is complicated and has not been so far obtained analytically. On the other hand it is possible, provided that certain geometrical conditions are met, to reduce the problem of the flow in the screw channel to one in simpler geometrical configurations amenable to analytical methods of solution. On the basis of such solutions we shall derive in this paper an equation for the calculation of the pumping capacity of screw rotors.

Simplifying Conditions of the Flow in Screw Rotors

As has been already mentioned screw rotors have been used predominantly for transport of high viscosity materials. Hence, the inertial forces under the flow in rotors are, as a rule, negligible compared to the viscous forces and creeping motion is assumed to prevail in the system.

In industrially used equipment usually rotates, for design purposes, the screw while the barrel remains stationary. Under the creeping flow regime, however, the pumping capacity does not change if it is the barrel that rotates at the same angular velocity in the opposite direction while the screw remains stationary. The case of the rotating barell and stat'onary screw, however, is more advantageous from the stand point of the solution of the equations of motion. For this reason we shall utilize exclusively this situation in the subsequent considerations.

Even though the transported material displays in practice often the non-Newtonian behaviour, in this text we shall consider only the fundamental case of pumping Newtonian liquids. Non-Newtonian flow behaviour renders the governing differential equations nonlinear and their solution is amenable only to numerical methods of solution. We shall also confine ourselves to the isothermal case of the flow. The effects of the non-Newtonian behaviour and nonisothermal conditions on the pumping characteristics of screw rotors have been throughly examined in the monograph by Fenner[9].

Before going over to the solution of the flow proper we shall introduce gradually the following simplifying assumptions: We shall consider only sufficiently long screws, where the effect of flow stabilization in end regions will be negligible. In practice the screw must be manufactured with a certain clearance between the screw and the barrel; here, however, we shall consider, as a first approximation, this clearance to be negligible. Thus we take $D_t = d$.

In case that the diameter of the root, d_1, is close to the diameter of the screw d, *i.e.* $d_1/d \rightarrow 1$, one may neglect the curvature of the screw channel and reduce the solution to one for the flow in a straight channel of rectangular cross section with a moving wall shown in Fig. 2. The wall moves at the peripheral velocity of the barrel, U_t

$$U_t = \pi D_t n \tag{7}$$

and makes with the axis of the channel an angle φ_t. This velocity may be decomposed to a component in the direction down of the channel, U

$$U = \pi D_t n \cos \varphi_t \tag{8}$$

and a component V perpendicular to the direction of the channel

$$V = \pi D_t n \sin \varphi_t \,. \tag{9}$$

Also the flow in the channel may be conveniently split to the longitudinal one (in the direction of the channel z) and the transverse one (in the direction of the coordinate x)

The Flow in a Shallow Channel with a Mobile Wall

If the width of the channel, W, is much greater than its depth, H, or, if the ratio H/W tends to zero, one may neglect the effect of the side walls of the channel (this effect would show only in their immediate vicinity) and the problem may be solved as one of the flow between two unconfined parallel plates the upper of which is mobile – see Fig. 3. Solution of this case, obtained for the first time by Rowell and Finlayson[1] has become the basis for the calculation of screw rotors.

Let us derive first the velocity profile in the direction of the channel by solving the z-direction component of the Navier–Stokes equation. For the above case the latter takes the form (see *e.g.* ref.[10])

$$\mu \frac{d^2 u_z}{dy^2} = \frac{\Delta p}{L_z}, \tag{10}$$

where Δp designates the overall pressure difference along the screw. Eq. (*10*) must be still supplemented by the following boundary conditions

$$y = 0, \quad u_z = 0 \tag{11a}$$

$$y = H, \quad u_z = U. \tag{11b}$$

Solution of Eq. (*10*) with the boundary conditions (*11*) yielded for the longitudinal

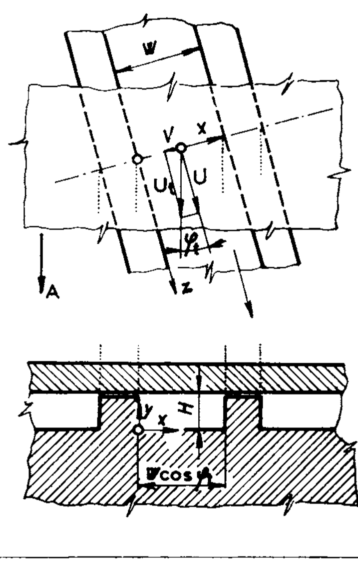

FIG. 2

Flow in a straight channel with mobile wall. A direction of peripheral velocity

velocity component, u_z, the following expression

$$u_z = \frac{U}{H} y - \frac{\Delta p}{2\mu L_z} (Hy - y^2) = u_{zd} - u_{zp} . \qquad (12)$$

The first term on the right hand side Eq. (*12*) represents the velocity of the so-called drag flow, u_{zd}, due to the motion of the upper plate (rotation of the screw). The second term represents then the velocity of the so-called pressure flow, u_{zp}, induced by the pressure difference, Δp, that must be overcome by the screw. The net longitudinal velocity, u_z, is given by the difference of the drag and the pressure flow, as it is illustrated in Fig. 4.

Apart from the just discussed longitudinal flow there exists in the screw also a transverse flow under the action of the component of peripheral velocity, V, perpendicular to the direction of the channel. The profile of the x component of the velocity may be obtained by solving the appropriate component of the Navier–Stokes equation

$$\mu \frac{d^2 u_x}{dy^2} = \frac{\partial p}{\partial x} \qquad (13)$$

with the boundary conditions

$$y = 0 , \quad u_x = 0 \qquad (14a)$$

$$y = H , \quad u_x = -V . \qquad (14b)$$

The unknown component of the pressure gradient $\partial p/\partial x$ may be obtained, for the case with no clearance between the side walls of the duct and the mobile plate, from the condition of zero flow rate in the transverse direction. This consideration

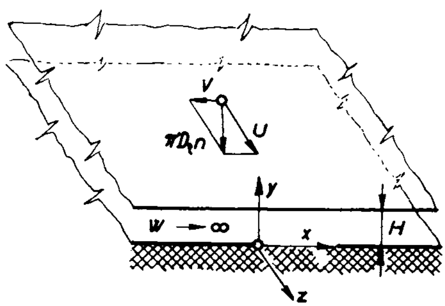

FIG. 3

Flow between two unconfined parallel plates

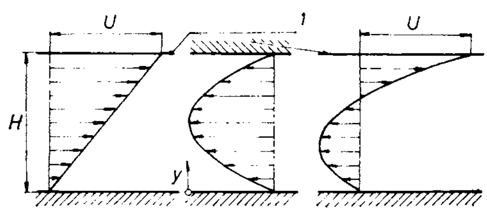

FIG. 4

Longitudinal velocity profile in a channel. 1 mobile plate leads to the following equation for the distribution of the velocity in the transverse flow in the form.

$$u_x = \frac{2V}{H} y - \frac{3V}{H^2} y^2 . \qquad (15)$$

The shape of the velocity profile computed from this expression is shown in Fig. 5. The transverse flow in case of zero clearance does not affect the pumping capacity of the screw, but, the transverse circulation contributes to the mixing effect of screw rotors.

The volume flow rate through the channel, $\dot{V}_1$, is thus determined solely by the longitudinal flow and may be obtained by integration of the longitudinal velocity component over the cross sectional area of the channel

$$\dot{V}_1 = W_t \int_0^H u_z \, dy = \frac{UHW_t}{2} - \frac{\Delta p W_t H^3}{12 \mu L_z} = \dot{V}_{d1} - \dot{V}_{p1} . \qquad (16)$$

The first term on the right hand side of Eq. (*16*) represents the contribution of the drag flow, $\dot{V}_{d1}$, to the flow rate and the second term the contribution of the pressure flow, $\dot{V}_{p1}$. The net flow rate is given by the difference of the two terms.

The Flow Rate in a Channel of Rectangular Cross Section with a Mobile Wall

Unless the condition $H/W \rightarrow 0$ is met, the effect of the side walls of the channel cannot be neglected. Longitudinal flow through a channel of rectangular cross section with a mobile wall, depicted in Fig. 2, has been investigated in the past by numerous authors. A review of these studies is given in ref.[2]. Details of the method of solution may be found, for instance, in the thesis[11]. Here we shall present only final results for the volumetric flow rate. For the contribution of the drag flow we have obtained the following expression

$$\dot{V}_{d1} = \frac{8UW_s^2}{\pi^3} \sum_{m=1,3,5...}^{\infty} \frac{1}{m^3} \operatorname{tgh} \frac{m \pi H}{2W_s} \qquad (17)$$

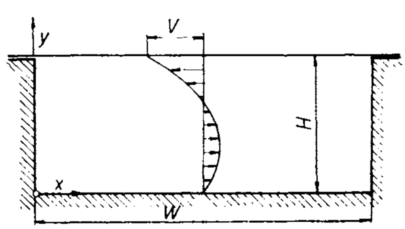

FIG. 5
Shape of the transverse velocity profile

while the contribution of the pressure flow is given by

$$\dot{V}_{p1} = \frac{\Delta p}{2\mu L_z}\left[\frac{W_s H^3}{6} - \frac{32H^4}{\pi^5} \sum_{m=1,3,5\ldots}^{\infty} \frac{1}{m^5} \operatorname{tgh} \frac{m\pi W_s}{2H}\right]. \quad (18)$$

The above relationships contain infinite series and as such appear for current technical calculation too complicated. For this reason Squires[4] proposed a method of calculation starting from the equation valid for infinite plates, Eq. (*16*), while respecting the effect of the side walls through correction coefficients F_d and F_p for the drag and the pressure flow respectively

$$\dot{V}_1 = \frac{UHW_s}{2} F_d - \frac{\Delta p W_s H^3}{12\mu L_z} F_p . \quad (19)$$

The expression for the calculation of the correction coefficients may be obtained from comparison of the corresponding terms in Eqs (*19*) and (*17*) or (*18*). Thus we have obtained the following relationships

$$F_d = \frac{16}{\pi^3} \frac{W_s}{H} \sum_{m=1,3,5\ldots}^{\infty} \frac{1}{m^3} \operatorname{tgh} \frac{m\pi H}{2W_s} \quad (20)$$

$$F_p = 1 - \frac{192}{\pi^5} \frac{H}{W_s} \sum_{m=1,3,5\ldots}^{\infty} \frac{1}{m^5} \operatorname{tgh} \frac{m\pi W_s}{2H}, \quad (21)$$

The values of both correction coefficients depend only on the shape of the cross section of the channel, *i.e.* on the ratio H/W_s only. In the engineering calculations the correction coefficients may be determined from graphical dependences shown in Fig. 6, constructed on the basis of Eqs (*20*) and (*21*). From this figure it is apparent that the values of both coefficients are less than unity, as the presence of the side walls obstructs the flow within the channel. From the same figure it is further apparent that the pressure flow is limited by the presence of the side walls to a greater extent than the drag flow.

For the calculation of the correction coefficients in slender channel one can make use of the following asymptotic equations derived in ref.[12]

$$F_d = 1 - 0{\cdot}543 \frac{H}{W_s} \quad \text{for} \quad (H/W_s) < 0{\cdot}6 \quad (22a)$$

$$F_d = 0{\cdot}543 \frac{W_s}{H} \quad \text{for} \quad (H/W_s) > 1{\cdot}5 \quad (22b)$$

$$F_p = 1 - 0{\cdot}630 \frac{H}{W_s} \quad \text{for} \quad (H/W_s) < 0{\cdot}6 \tag{23a}$$

$$F_p = \left(\frac{W_s}{H}\right)^2 \left(1 - 0{\cdot}630 \frac{W_s}{H}\right) \quad \text{for} \quad (H/W_s) > 1{\cdot}5\,. \tag{23b}$$

Of maximum importance for technical calculations are Eqs (*22a*) and (*23a*) as the majority of screws used in technical practice is characterized by relatively wide channels.

The Flow Rate under the Helicoidal Flow between Two Unconfined Cylinders

Another shape of the screw channel for which an analytical solution can be obtained, displays opposite properties than the just discussed case. Namely we shall be concerned, on the one hand, with non-negligible curvature ($d_1/d \ll 1$) but, on the other hand, with negligible effect of the screw blade on the flow rate, that means $H/W \to 0$. This requirement fulfil well, for instance, screws with a relatively large pitch. The flow within the screw, under such circumstances, may be simulated by the flow between two unconfined cylinders one of which is rotating under the specified pressure drop and under the condition of zero flow rate in the direction perpendicular to the blade of the screw. Solution of this case is relatively laborious and the interested reader may find it either in the original work[6] or in ref.[11]. The resulting expression is again rather cumbersome for current technical calculations and de Booy[6] therefore proposed a method starting from Eq. (*16*) while the effect of the curvature, similarly as the effect of the side walls, is respected by the correction coefficients F_{dc} and F_{pc},

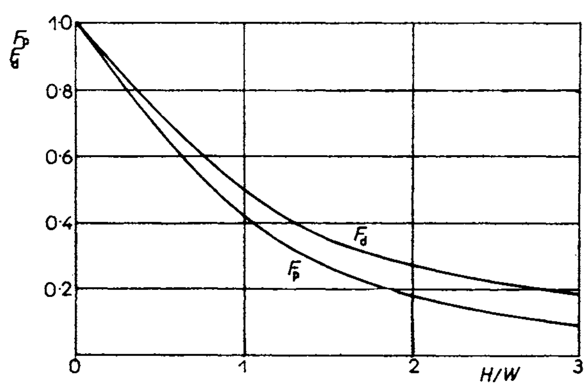

FIG. 6

Chart of the correction coefficients according to Eqs (*20*) and (*21*) for which one can write the following expressions

$$F_{dc} = \frac{2\varkappa^2}{1-\varkappa}\left[1 + \frac{1}{\pi^2}\left(\frac{s}{D_t}\right)^2\right]\frac{f''(\varkappa)}{1 + \frac{8}{\pi^2\varkappa}\left(\frac{s}{D_t}\right)^2\frac{f'(\varkappa)}{f(\varkappa)}} \tag{24}$$

$$F_{pc} = \frac{6\varkappa^3}{(1-\varkappa)^3}\left[1 + \frac{1}{\pi^2}\left(\frac{s}{D_t}\right)^2\right]\frac{f'(\varkappa)}{1 + \frac{8}{\pi^2\varkappa}\left(\frac{s}{D_t}\right)^2\frac{f'(\varkappa)}{f(\varkappa)}}, \tag{25}$$

where

$$f(\varkappa) = \frac{1}{\varkappa^4}\left[1 - \varkappa^4 + \frac{(1-\varkappa^2)^2}{\ln\varkappa}\right] \tag{25a}$$

$$f'(\varkappa) = \frac{1-\varkappa^2}{4\varkappa^3} - \frac{1}{\varkappa}\frac{(\ln\varkappa)^2}{1-\varkappa^2} \tag{25b}$$

$$f''(\varkappa) = \frac{\ln\varkappa}{1-\varkappa^2} + \frac{1}{2\varkappa^2}\,. \tag{25c}$$

As may be apparent from Eqs (*24*) and (*25*) both correction coefficients F_{dc} and F_{pc} depend on the geometrical simplexes characterizing the curvature of the channel, *i.e.* on the values of d_1/D_t and s/D_t. Apart from the above equations one can utilize for their evaluation the charts presented in Fig. 7 taken over from the original work[6]. The two coefficients are here presented in dependence on the ratio H/D_t and on the external helix angle φ_t. From Fig. 7 it is also seen that with a small helix angle we have $F_{dc} > 1$ and the contribution of the drag flow is thus a little bit higher than follows from the planar theory, while for higher values of the pitch the situation is reverse. The value of F_{pc} for current values of the pitch is greater than 1 and the calculations, respecting the curvature, therefore yield usually greater contribution from the pressure flow than calculations with neglected curvature.

The Flow Rate under the Flow in a Shallow Channel with Clearance

Solution of the case of the flow in a shallow channel with clearance between the side walls and the moving upper plate has been attempted by Mohr and Mallouk[5]. In contrast to the case with no clearance one cannot neglect the effect of the transverse flow on the volumetric flow rate. The resulting expression for the flow rate is again relatively complicated and, in analogy with the previous cases, it is advantageous to start with the equation for the shallow channel without clearance while implementing its effect through the use of correction factors k_d and k_p defined by the following equation

$$\dot{V}_l = \frac{UHW}{2}(1 - k_d) - \frac{\Delta p W H^3}{12\mu L_z}(1 + k_p). \tag{26}$$

By comparison of this equation with the resulting expression for the shallow channel with clearance, derived in the original work[5], the following expressions for the correction coefficients were obtained

$$k_d = \frac{c}{H} + \frac{H - c}{H} \frac{(W + e\cos\varphi_t)(c/H)^3}{W(c/H)^3 + e\cos\varphi_t} \tag{27}$$

$$k_p = \frac{e\cos\varphi_t}{W}\left(\frac{c}{H}\right)^3 + \frac{(\pi D_t \cos\varphi_t)^2 (c/H)^3}{i^2 W(W(c/H)^3 + e\cos\varphi_t)}. \tag{28}$$

Both coefficients always take positive values and hence the clearance always decreases the resulting flow rate.

The Effect of Flow Development on the Flow Rate in Shallow Channel

Up to now the investigated flow has been always one with fully developed velocity and pressure profile. For short screws, however, the development of the flow in the end regions may have considerable effect on the flow rate. For this reason de Booy[6]

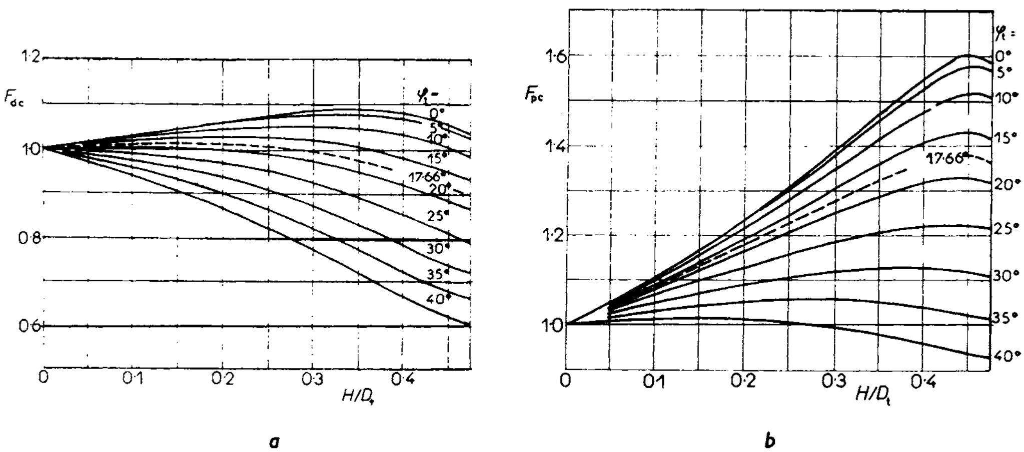

FIG. 7
Chart of the correction coefficients *a* F_{dc} and *b* F_{pc} according to Booy[6]

studied the effect of the development of the pressure profile on the flow rate in shallow channel. The principles of his approach may be apparent from Fig. 8 showing the developed shape of the screw channel. While at the end of the channel the direction of the isobars must be identical with the outlet surface of the channel, the direction of the isobars in the middle section of the channel is determined by the operating conditions of the screw rotor (*i.e.* depends on the pressure to be overcome by the screw). To account for the effect of the pressure profile development de Booy proposed[6], similarly as in all previous cases, correction coefficients F_{de} and F_{pe}. For screw exceeding in length the length of the region necessary for fully developed flow he recommended the following relationships

$$F_{pe} = \left(1 + \frac{W}{L_z}(0{\cdot}96 \operatorname{cotg} \varphi - 0{\cdot}64)\right)^{-1} \tag{29}$$

$$F_{de} = 1 + \operatorname{tg}^2 \varphi(1 - F_{pe}) \tag{30}$$

obtained on the basis of analysis of his numerical solutions for helix angle below 30 degrees.

Theoretical Equation for the Pumping Characteristics of Screw Rotors

In the previous paragraphs we have presented relationships for the correction coefficients respecting the effects of individual simplifying assumptions which help us transform the complicated case of the flow in a screw channel to a simple case of the flow between two unconfined plates. These correction coefficients were obtained separately from the solutions of the flow in geometries in which always one of the simplifying assumptions is invalid. Booy, however, assumed superposition of individual effects and proposed in turn a general equation for the calculation of the flow rate in a screw channel with a general geometry in the form

$$\dot{V} = i\,\frac{UHW_t}{2}\,F_d F_{dc} F_{de}\,(1 - k_d) - i\,\frac{\Delta p W_t H^3}{12\mu L_z}\,F_p F_{pc} F_{pe}\,(1 + k_p)\,. \tag{31}$$

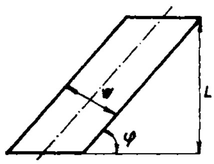

FIG. 8
The screw channel developed in a plane

The pumping capacity of the screw rotor, $\dot{V}$, was obtained by multiplying the flow rate through a single screw channel by the number of the channels in the rotor, *i.e.* the number of the threads of the screw, i

$$\dot{V} = \dot{V}_1 i \,. \tag{32}$$

The resulting dimensionless equation of the pumping capacity may be obtained by modifying Eq. (*31*) in which all quantities, characterizing the screw channel are replaced by geometrical characteristics of the system screw rotor–barrel using Eqs (*1*), (*4a*), (*5*) and (*9*).

$$\frac{\dot{V}}{nd^3} = \frac{\pi}{4}\left(\frac{s}{d} - i\,\frac{e}{d}\right)\left(\frac{D_t}{d} - \frac{d_1}{d}\right)\frac{D_t}{d}\cos^2\varphi_t\, F_d F_{dc} F_{de}(1 - k_d) - $$
$$- \frac{1}{96}\left(\frac{s}{d} - i\,\frac{e}{d}\right)\left(\frac{D_t}{d} - \frac{d_1}{d}\right)^3 \frac{d}{L}\sin\varphi_t\cos\varphi_t F_p F_{pc} F_{pe}(1 + k_p)\,\frac{\Delta p}{\mu n} \tag{33}$$

From this equation it follows that the dimensionless pumping characteristic, *i.e.* the dependence of the dimensionless pumping capacity, $\dot{V}/nd^3$, on the dimensionless pressure drop, $\Delta p/\mu n$, is linear and may be written in the form

$$\frac{\dot{V}}{nd^3} = a - b\,\frac{\Delta p}{\mu n}\,, \tag{34}$$

where the coefficients a and b are functions of the geometry of the system screw rotor–barrel. The dimensionless equation of the pumping characteristic is often written in an alternative form as

$$\frac{\Delta p}{\mu n} = A - B\,\frac{\dot{V}}{nd^3}\,. \tag{35}$$

From comparison of Eqs (*34*) and (*35*) one can obtain the following relations between the coefficients in both equations

$$A = a/b\,,\quad B = 1/b\,,\quad a = A/B\,,\quad b = 1/B\,. \qquad (36a,b,c,d)$$

Since the theoretical computational equation has been based on the principle of superposition of individual effects, its applicability for the calculation of the pumping characteristics of screw rotors must be verified experimentally. Accordingly, the measurement of the pumping characteristics of screw rotors shall be subject of the following communication.

LIST OF SYMBOLS

a	coefficient in Eq. (*34*)
A	coefficient in Eq. (*35*)
b	coefficient in Eq. (*34*)
B	coefficient in Eq. (*35*)
c	clearance between screw and barrel, m
d	diameter of screw, m
d_1	diameter of screw root, m
D_t	internal diameter of barrel, m
e	axial thickness of screw blade, m
F	correction coefficient
H	depth of screw channel, m
i	number of threads of screw
k	correction coefficient on the effect of clearance
L	length of screw rotor, m
L_z	length of screw channel, m
n	frequency of revolution, s^{-1}
p	pressure, Pa
s	pitch of screw, m
u	velocity, m s^{-1}
U	component of velocity U_t in the direction of the channel, m s^{-1}
U_t	fictious peripheral velocity of barrel, m s^{-1}
V	component of velocity U_t in the direction perpendicular to the channel, m s^{-1}
$\dot{V}$	volumetric flow rate, m^3 s^{-1}
W	width of screw channel, m
x, y, z	Cartesian coordinates, m
φ	helix angle
μ	dynamic viscosity, Pa. s
ϱ	density, kg m^{-3}
$\varkappa$	ratio of diameters d_1/D_t

Subscripts

d	drag flow
p	pressure flow
c	curvature
e	end effects
s	mean
t	related to diameter D_t
1	related to single channel

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