POWER INPUT OF SCREW ROTORS AND AGITATORS: THEORY OF CALCULATION OF POWER INPUT OF SCREW ROTORS

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A method has been proposed of the calculation of screw rotors. The calculation starts from the equation for the power input for the flow between two plates corrected by correction coefficients.

The first of the series of two papers relates to the earlier published series[1-3] devoted to the calculation of the pumping capacity of screw rotors and has been devoted to the problem of the calculation of the power input of screw rotors. In contrast to the theory of the calculation of the pumping capacity, which is well elaborated and in the literature described in detail, the data on the power input in the literature[4] are confined to the calculation of the screw rotors with a shallow channel when one can neglect the effect of curvature and the flight. The only exception is the work of Fenner[5] who attempted to describe the effect of screw flight on the power input. The aim of this paper therefore is to propose the method of calculation of the power input of real screw rotors with non-negligible effects of the flight and the curvature as well. The starting point shall be the well known relation for the power input of of screw rotors with a shallow channel the derivation of which[4] shall be reproduced in the following paragraph.

The Power Input of Screw Rotors with a Shallow Channel

If the geometry of the screw rotor, depicted in Fig. 1, is characterized by a relatively large diameter of the root – $d_1/d \to 1$, one can neglect the curvature of the barrel and to unroll the channel onto a plane – see Fig. 2. If, in addition, the screw channel is shallow or exhibits a relatively low depth, H, compared to W, $H/W \to 0$, one can neglect also the effect of its side walls and to interpret the flow in the channel of rectangular cross section as between two unconfined plates – see Fig. 3. In this case the velocity has two nonzero components – the longitudinal one, u_z, and the transverse, u_x, dependent on the coordinate y.

The longitudinal velocity profile is described by the following equation (see Eq. (*12*) in ref.[1])

$$u_z = \frac{U}{H} y - \frac{\Delta p}{2\mu L_z} (Hy - y^2) = u_{zd} - u_{zp} \,. \tag{1}$$

and consists of two parts: the velocity due to the drag flow, u_{zd}, and the velocity due

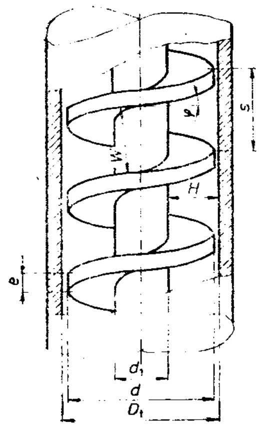

FIG. 1
Screw rotating in a guide roll

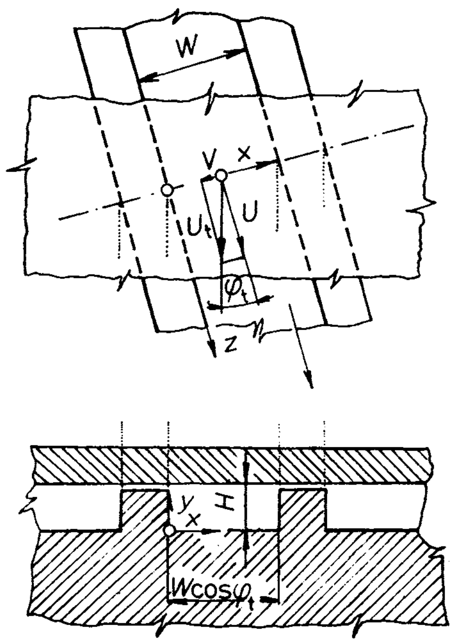

FIG. 2
Unrolled screw duct

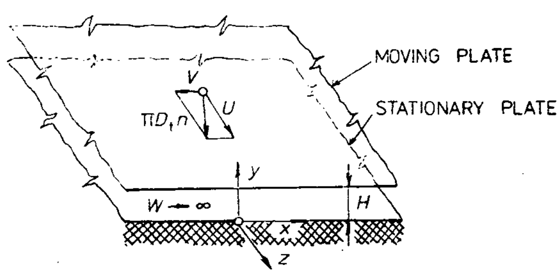

FIG. 3
Two infinite plates — mobile and stationary

to the pressure flow, u_{zp}. The transverse velocity profile describes (see Eq. (*15*) in ref.[1])

$$u_x = 2(V/H)\,y - 3(V/H^2)\,y^2\,. \tag{2}$$

The power input P_1 necessary for the motion of the upper plate moving above the stable plate may be determined as a scalar product of the velocity of this plate, $\boldsymbol{U}_t$, and the force, $\boldsymbol{F}$, exerted by the liquid in the channel

$$P_1 = \boldsymbol{U}_t\,.\,\boldsymbol{F} = U_{tx}F_x + U_{tz}F_z = -VF_x + UF_y\,. \tag{3}$$

The components of the force may be expressed as a product of the surface of the channel, S, and the corresponding component of the shear stress

$$F_x = \tau_{xy}S = \mu\,\frac{\mathrm{d}u_x}{\mathrm{d}y}\bigg|_{y=H} S\,, \tag{4}$$

$$F_z = \tau_{zy}S = \mu\,\frac{\mathrm{d}u_z}{\mathrm{d}y}\bigg|_{y=H} S\,. \tag{5}$$

Having expressed the derivatives of the velocity from Eq. (*1*) or (*2*) one obtains for the components of the force the following expressions

$$F_x = -4\mu WL_z\,.\,V/H\,, \tag{6}$$

$$F_z = \mu WL_z\,.\,U/H + \tfrac{1}{2}\Delta pWH\,, \tag{7}$$

where we expressed, after some arrangement, the surface S as

$$S = WL_z\,. \tag{8}$$

Substituting Eqs (*6*) and (*7*) into Eq. (*3*) we obtain for the power input of the plate moving over the stable plate the following relation

$$P_1 = 4\mu WL_zV^2/H + \mu WL_zU^2/H + \Delta pHWU/2 = P_{dx1} + P_{dz1} + P_{p1}\,. \tag{9}$$

The first term on the right hand side of Eq. (*9*) represents the power input necessary for the transverse flow P_{dx1}, the second term the power input necessary for the drag longitudinal flow, P_{dz1}, and the third term the power input necessary for the pressure flow, P_{p1}. Since the expression $HWU/2$ represents the flow rate due to the drag flow

$\dot{V}_{d1}$ (see Eq. (*16*) from ref.[1]), the power input for the pressure difference Δp and the flow rate $\dot{V}_{d1}$

$$P_{p1} = \dot{V}_{d1}\,\Delta p\,. \tag{10}$$

The term P_{p1} thus represents the power input necessary to transform the flow rate of liquid from the drag flow to a higher pressure level.

If the screw rotor has several flights one obtains the overall power input as a product of the number of the channels (flights), i, and the power input for a single channel P_1

$$P = iP_1 = (\mu iWL_zU_t^2/H)(\cos^2\varphi_t + 4\sin^2\varphi_t) + \Delta p\,\dot{V}_d = \\ = (\mu iWL_zU^2/H)(1 + 4\,\mathrm{tg}^2\,\varphi_t) + \Delta p\,\dot{V}_d\,, \tag{11}$$

where the total flow rate due to the drag flow was expressed as a product $\dot{V}_d = i\dot{V}_{d1}$ and for the component of the velocity we substituted from Eqs (*7*) through (*9*) from ref.[1] the following relations

$$V = U_t \sin\varphi_t\,,\quad U = U_t\cos\varphi_t\,. \tag{12a, b}$$

From Eq. (*11*) it follows that the ratio of the power input for the transverse and the drag longitudinal flow $P_{dx}/P_{dz} = 4\,\mathrm{tg}^2\,\varphi_t$. This ratio thus increases with increasing helix angle, φ_t, or the pitch of the screw. For instance, for $s/d = 0{\cdot}5$ the ratio $P_{dx}/P_{dz} = 0{\cdot}1$ white for $s/d = 2$ the ratio $P_{dx}/P_{dz} = 1{\cdot}62$.

In concluding this paragraph we must stress that while for the calculation of the pumping capacity of rotors with a shallow channel the effect of the screw flight is relatively small (e.g. for $H/W = 0{\cdot}1$ the error of the pumping capacity due to the neglected effect of the flight amounts to about 6% (refs[1–3])), the error of the determination of the power input is much larger due to the high dissipation of mechanical energy in the corners of the channel and between the flight and the barrel. For this reason the power input for the drag flow for real screws is, as a rule, much higher than the value computed from Eq. (*11*). For this reason the following paragraphs are devoted to expressing the effect of the flight on the power input.

The Power Input under the Drag Flow in Annular Channels of Rectangular Cross Section

Annular channels of rectangular cross section, depicted in Fig. 4, is an extreme case of a screw channel for a limiting pitch $s \to 0$. The power input necessary to drive the external cylinder of the annular channel may be expressed as a product of the power input for the drag flow between two unconfined plates and a correction factor,

G_d, for which the following expression was derived[6]

$$G_d = \frac{1}{8}\frac{1-\varkappa}{1-\bar{\varkappa}}(1+\bar{\varkappa})^3 \left[1 - \frac{8}{\bar{\varkappa}\pi^2}\sum\frac{1}{j^2}\frac{\dfrac{I_1(j\pi R/W)}{I_1(j\pi R_1/W)} - \dfrac{K_1(j\pi R/W)}{K_1(j\pi R_1/W)}}{\dfrac{I_1(j\pi R_2/W)}{I_1(j\pi R_1/W)} - \dfrac{K_1(j\pi R_2/W)}{K_1(j\pi R_1/W)}}\right]. \qquad (13)$$

$$j = 1, 3, 5 \ldots$$

From Eq. (*13*) it is apparent that the coefficient G_d depends on the curvature of the channel characterized by the ratio $\varkappa = R_1/R_2$, on the clearance between the walls confining the channel and the outer cylinder characterized by the ratio $\bar{\varkappa} = R/R_2$ (or alternatively by the ratio c/H, where $c = R_2 - R$ and $H = R_2 - R_1$) and on the relative width of the channel W/R_2 (or W/H). The dependence of the correction coefficient G_d on H/W calculated from Eq. (*13*) for several valuesof $\varkappa$ and $c/H = 0{\cdot}1$ is shown graphically in Fig. 5 taken over from ref.[6]. From the figure it is apparent that the value of the correction coefficient expressing the effect of the walls of the channel (discs) on the power input increases with increasing ratio of the depth of the channel H to its width W.

In a number of limiting cases the relatively complicated expression (*13*) may be replaced by simpler asymptotic formulas[6,7]. For instance, for $\varkappa \to 1$, when the curvature of the channel is negligible, Eq. (*13*) transforms into the relationship for a straight

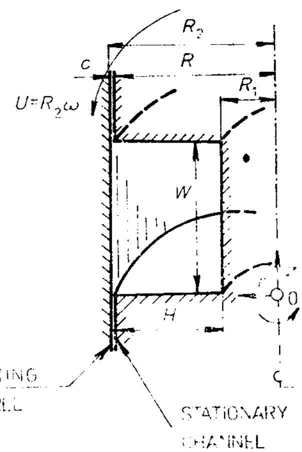

FIG. 4
Anular channel of rectangular cross section

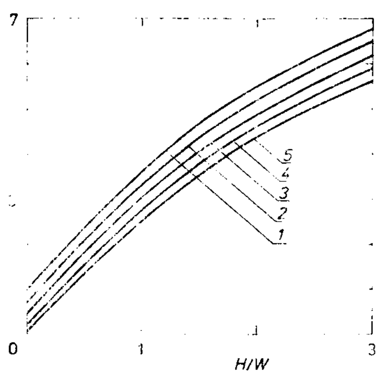

FIG. 5
Dependence of the coefficient G_d on the ratio H/W. $\varkappa$: *1* 1·0; *2* 0·8; *3* 0·6; *4* 0·4; *5* 0·2

channel of rectangular cross section derived by Fenner[5]. For the calculation of the screw rotors, exhibiting as a rule relatively wide channels, one can make use of the following asymptotic formula

$$F_d = \frac{2\varkappa^2}{1 + \varkappa} + \bar{K}_1 \frac{H}{W} \qquad (14)$$

expressing the straight sections of the dependence on the left hand side of Fig. 5. The graphical dependences of the coefficient $\bar{K}_1$ on $\bar{\varkappa}$ computed for several values of $\varkappa$ from the expressions derived in ref.[6] are depicted in Fig. 6. As also the relations for $\bar{K}_1$ are relatively complicated (Eq. (*27*) from ref.[7] and Eq. (*29*) or (*34*) from ref.[6]), the following correlations were proposed for the dependence of $\bar{K}_1$ on $\varkappa$ and $\bar{\varkappa}$

$$\bar{K}_1 = a_1 - 2{\cdot}9 \log \frac{R_2 - R}{R_2 - R_1} = a_1 - 2{\cdot}9 \log \frac{1 - \bar{\varkappa}}{1 - \varkappa}, \qquad (15)$$

where the dependence of the coefficient a_1 on the ratio $\varkappa$ is correlated by

$$a_1 = -0{\cdot}544 + 1{\cdot}21\varkappa - 1{\cdot}1\varkappa^2 \quad \text{for} \quad 0 \leqq \varkappa \leqq 0{\cdot}4 \qquad (15a)$$

or

$$a_1 = -0{\cdot}398 + 0{\cdot}518\varkappa - 0{\cdot}267\varkappa^2 \quad \text{for} \quad 0{\cdot}4 < \varkappa \leqq 1{\cdot}0 \qquad (15b)$$

or

$$\bar{K}_1 = 1{\cdot}06 \left[\left(\frac{\bar{\varkappa}^4}{1 - \bar{\varkappa}}\right)^{0.39} - \left(\frac{\varkappa^4}{1 - \varkappa}\right)^{0.39}\right]. \qquad (16)$$

The relation (*16*) respects also the empirical finding[8] that the real power input is for lower values of $\bar{\varkappa}$ lower than the theoretical one and appears therefore more suitable for the calculation of the power input.

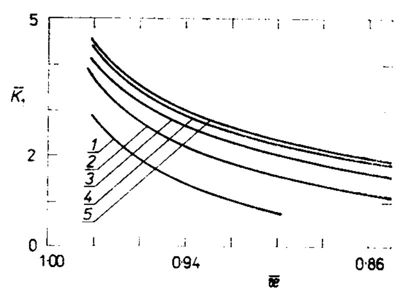

FIG. 6

Dependence of the coefficient $\bar{K}_1$ on the ratio $\bar{\varkappa}$ for several selected values of the ratio $\varkappa$: *1* 0·8; *2* 0·6; *3* 0·4; *4* 0·2; *5* 0

The Power Input under the Helicoidal Flow between Two Unconfined Cylinders

Another extreme case to which the flow in a screw rotor may reduce under negligible effect of the flight (e.g. at $H/W \to 0$) but non-negligible curvature is the helicoidal flow between two unconfined cylinders. The velocity profile for this case has been derived by Booy[9]. The velocity has in the cylindrical coordinates two nonzero components u_φ and u_z, which depend on the coordinate r only. The power input necessary for the rotation of the outer cylinder is obtained as a product of its angular velocity ω and the corresponding torque M_k

$$P = \omega M_k = \omega R_2 F_{\varphi 2}\,, \tag{17}$$

where $F_{\varphi 2}$ is the tangential component of the force acting on the outer cylinder of radius R_2. Upon expressing this component of the force as a product of the area of the outer cylinder $2\pi R_2 L_c$ and the shear stress $\tau_{r\varphi 2}$ acting on it we obtain

$$P = 2\pi R_2^2 \omega L_c \tau_{r\varphi 2} = 2\pi R_2^2 L_c \omega \mu \left[r \frac{\mathrm{d}}{\mathrm{d}r} \left(\frac{u_\varphi}{r} \right) \right]_{r=R_2}, \tag{18}$$

where the component of the shear stress $\tau_{r\varphi 2}$ was expressed from the Newton law as a product of viscosity and the shear rate. For the calculation of the power input we thus have to know the velocity component u_φ which may be further decomposed into the velocity of the drag flow and the pressure flow. It may be proven that the power input for the pressure flow can be again expressed as a product of the pressure difference and the flow rate due to the drag flow (see Eq. (*10*)).

For the calculation of power input for drag flow one must know the drag part of the component u_φ for which one can write according to ref.[9] or ref.[10] that

$$u_\varphi = U_t \left\{ \frac{\varkappa}{1 - \varkappa^2} \left(\frac{r}{R_1} - \frac{R_1}{r} \right) - \frac{2s^2}{\pi^2 R_2 R_1} \frac{f''(\varkappa)}{(2s^2/\pi^2 R_2 R_1) f'(\varkappa) + f(\varkappa)} \cdot \right.$$
$$\left. \cdot \left[\frac{r}{R_1} \ln \frac{r}{R_1} + \frac{\ln \varkappa}{1 - \varkappa^2} \left(\frac{r}{R_1} - \frac{R_1}{r} \right) \right] \right\}, \tag{19}$$

where the ratio of the radius of the inner and outer cylinder R_1/R_2 is again designated as $\varkappa$ and we further designate

$$f(\varkappa) = \frac{1}{\varkappa^4} \left[1 - \varkappa^4 + \frac{(1 - \varkappa^2)^2}{\ln \varkappa} \right] \tag{20a}$$

$$f'(\varkappa) = \frac{1 - \varkappa^2}{4\varkappa^3} - \frac{1}{\varkappa}\frac{(\ln \varkappa)^2}{1 - \varkappa^2} \qquad (20b)$$

$$f''(\varkappa) = \frac{\ln \varkappa}{1 - \varkappa^2} + \frac{1}{2\varkappa^2}. \qquad (20c)$$

Having substituted Eq. (*19*) into Eq. (*18*) and after differentiation one obtains for the drag part of the power input P_d the following relation

$$P_d = 2\pi\mu\omega^2 R_2^2 L_c \left[\frac{2\varkappa^2}{1 - \varkappa^2} + \frac{2}{\pi^2}\left(\frac{s}{R_2}\right)^2 \frac{1}{\varkappa} \frac{f''(\varkappa)}{(2/\pi^2)(s/R_2)^2 \varkappa^{-1} f'(\varkappa) + f(\varkappa)} \cdot \left(\frac{1}{\varkappa} + \frac{2\varkappa \ln \varkappa}{1 - \varkappa^2}\right)\right]. \qquad (21)$$

From Eq. (*21*) it is apparent that the power input under the helicoidal flow between two cylinders may be taken as a sum of the power input under the tangential flow between two unconfined cylinders (for $s = 0$) P_{d1}, expressed by the first term on the right hand side of Eq. (*21*) and the power input due to the helicoidal flow P_{d2}, expressed by the second term.

The effect of the curvature, the clearance and the flight on the power input under the tangential flow (flow in the direction φ) may be expressed on the basis of results of the previous case of the flow in a annulus of rectangular cross section. We shall therefore turn now our attention to the second term expressing the effect of the curvature under the helicoidal flow. In practical calculations it is usually advantageous to incorporate the effect of the curvature on the power input into a correction coefficient defined as a ratio of the second term in Eq. (*21*) to the power input under

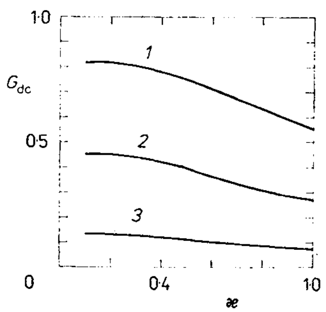

FIG. 7

Dependence of the correction coefficient G_{dc} on the ratio $\varkappa$ for several selected values s/D_t: *1* 1·5; *2* 1·0; *3* 0·5 the drag flow between two plates of surface area equal $2\pi R_2 L_c$ and mutual distance $H = R_2 - R_1$ while one of these plates moves at a velocity equal the peripheral velocity of the outer cylinder ωR_2

$$G_{dc} = \frac{P_{d2}(R_2 - R_1)}{2\pi R_2 L_c \mu \omega^2 R_2^2} = \frac{8}{\pi^2}\left(\frac{s}{D_t}\right)^2 \frac{1-\varkappa}{\varkappa} \frac{f''(\varkappa)}{(8\pi^2)(s/D_t)^2 \varkappa^{-1} f'(\varkappa) + f(\varkappa)} \cdot \left(\frac{1}{\varkappa} + \frac{2\varkappa \ln \varkappa}{1-\varkappa^2}\right). \quad (22)$$

The dependence of G_{dc} on the ratio $\varkappa$ is shown graphically for several values of s/D_t in Fig. 7. From this figure it is apparent that the magnitude of the coefficient G_{dc} depends primarily on the value of the ratio of the pitch to the diameter of the outer cylinder s/D_t while its dependence on the ratio $\varkappa$ is much less conspicuous.

The Power Input under the Flow between Two Cylinders with Axial Blades

As has been mentioned the effect of the clearance between the barrel and the flight on the power input at zero pitch may be expressed from the solution for the case of the flow in a annular channel with rotating outer cylindrical wall. We shall attempt to express the effect of the clearance on the power input under the helicoidal flow at nonzero pitch from the solution of the second extreme case, i.e. the case of the flow between two cylinders with axial blades to which the screw transforms at the pitch $s \to \infty$. The solution of the flow in this configuration has been presented in ref.[11]. For the calculation of the power input for the flow between two cylinders with axial blades one can again use Eq. (*18*). Having performed the differentiation of the earlier derived velocity profile in this configuration (Eq. (*8*) from ref.[11]) and substitution into Eq. (*18*) we obtain after some modification

$$P = 2\pi R_2^2 L_c \mu \omega^2 \left[\frac{2\varkappa^2}{1-\varkappa} + \frac{1}{\mu\omega}\frac{\partial p}{\partial \varphi}\left(\frac{1}{2} + \frac{\varkappa^2 \ln \varkappa}{1-\varkappa^2}\right)\right]. \quad (23)$$

As has been pointed out in the previous paragraph the first term on the right hand side of Eq. (*23*) represents the power input under the tangential flow between two cylinders without axial blades. This term is not affected by the magnitude of the clearance between the blades and the outer cylinder because the clearance affects only the value of the pressure drop $\partial p/\partial \varphi$. The ratio of the pressure drops with and without the effect of the clearance, expressed by Eqs (*17*) and (*10*) from ref.[11], expresses thus the effect of the clearance on the component of the power input due to the flow induced by the presence of the blades. We shall designate this ratio as G_{dm}

$$G_{\mathrm{dm}} = \frac{\dfrac{\partial p}{\partial \varphi}}{\left(\dfrac{\partial p}{\partial \varphi}\right)_0} = \frac{8\left[\dfrac{\varkappa^2}{1-\varkappa^2}\ln\varkappa - \dfrac{\bar{\varkappa}^2}{1-\bar{\varkappa}^2}\ln\bar{\varkappa}\right]}{1+\alpha-\varkappa^2-\alpha\bar{\varkappa}^2-4\left[\dfrac{\varkappa^2}{1-\varkappa^2}(\ln\varkappa)^2+\dfrac{\alpha\bar{\varkappa}^2}{1-\bar{\varkappa}^2}(\ln\bar{\varkappa})^2\right]} \cdot$$

$$\cdot \frac{\dfrac{1}{4}(1-\varkappa^2) - \dfrac{\varkappa^2}{1-\varkappa^2}(\ln\varkappa)^2}{1+\dfrac{2\varkappa^2}{1-\varkappa^2}\ln\varkappa} \,. \qquad (24)$$

where the coefficient α was defined by the following relation (see Eq. (*18*) from ref.[11])

$$\alpha = \frac{(\pi/i)-(b/d)}{(b/d)} = \frac{\pi d}{ib} - 1\,, \qquad (25)$$

The dependence of the coefficient G_{dm} on the ratio of the radius of the blade to the radius of the outer cylinder, $\bar{\varkappa}$, is depicted in Fig. 8 for several values of the ratios b/d and $\varkappa$.

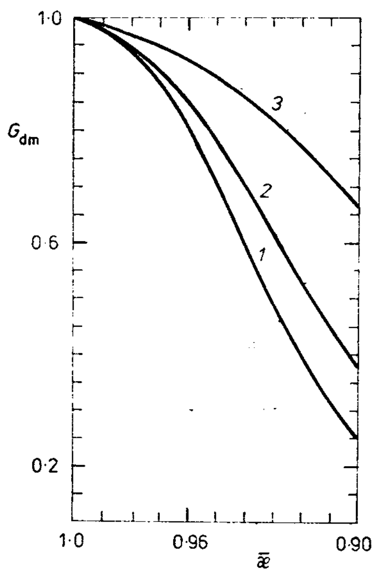

FIG. 8

Dependence of the correction coefficient G_{dm} on the ratio $\bar{\varkappa}$ for several values of the ratios b/d and $\varkappa$: *1* 0·02, 0·6; *2* 0·04, 0·6; *3* 0·04, 0·4

The Power Input Characteristics

The power input required to drive the screw rotor may be expressed as a sum of the power input for the drag flow in the screw channel P_d, the power input for the pressure flow in the screw channel P_p and the power input for the energy dissipation in the clearance between the screw flight and the barrel P_c:

$$P = P_d + P_p + P_c\,. \tag{26}$$

The determination of the power input for the pressure flow poses, as has been pointed out, no difficulties and can be done on the basis of the knowledge of the correction coefficients for the flow rate under the drag flow from Eq. (*10*). The component of the power input P_c may be determined from the relation for the power input under the Couette flow between two cylinders (see the first term in Eq. (*21*) or (*23*)) of the total width equal the axial thickness of the flight multiplied by the number of the threads L_t/s within the barrel (provided that it is shorter than the screw) and the number of the flights i

$$P_c = 4\pi^3 \mu d^2 n^2 e \frac{1}{1 - \bar{\varkappa}^2}\, i \frac{L_t}{s}\,. \tag{27}$$

The true magnitude of this component shall be somewhat lower as a consequence of the existence of end effects as has been shown in ref.[8].

The most difficult part is the expression for the component of the power input due to the drag flow in the screw channel P_d as there is no exact solution at hand as was the case of the previous two components. For this reason we shall attempt to express the effect of individual factors on the power input from the results of the above presented solutions for simpler geometrical configurations. We shall start from the expression for the power input under the drag flow between two plates. The effect of the side walls, curvature and the clearance on the power input under the tangential flow shall be expressed by means of the coefficient G_d, the effect of the helicoidal flow by means of the coefficient G_{dc} and the effect of the flight by means of the coefficient G_{dm}. Thus we obtain for the power input of the drag flow the following equation

$$P_d = \frac{2\pi R_2^3 L_k \mu \omega^2}{R_2 - R_1} \left(G_d + G_{dc} G_{dm}\right), \tag{28}$$

where L_k is the axial length of the screw channel, i.e. the length of the screw less the axial thickness of its flights, which is, as already mentioned, equal to the thickness of a single flight multiplied by the number of the threads L/s and the number of the flights i

$$L_k = L - eiL/s\,. \tag{29}$$

In the derivation of Eq. (*28*) we actually started from Eq. (*21*) in which the effect of the flight and the clearance on the first term was expressed by means of the correction coefficient G_d and on the second term by means of the coefficient G_{dm}, where the length L_c was replaced by the length of the screw channel L_k. The equation for the coefficient α necessary for the calculation of the coefficient G_{dm} shall be obtained if one substitutes into the definition equation (*25*) for b the tangential thickness of the screw flight $e/\text{tg}\,\varphi_d$.

$$\alpha = \text{tg}\,\varphi_d\,(\pi d/ie) - 1 = (s/ie) - 1 \tag{30}$$

If we now substitute gradually Eqs (*10*) and (*27*)–(*29*) into Eq. (*26*) we obtain after division by the expression $\mu n^2 d^3$ and after some modification the resulting dimensionless equation of the power input characteristics of the screw rotor in the form

$$P^* = \frac{P}{\mu n^2 d^3} = 2\pi^3 \left(\frac{D_t}{d}\right)^2 \frac{L - ie(L/s)}{d} \frac{1}{1 - \varkappa} (G_d + G_{dc} G_{dm}) +$$
$$+ 4\pi^3 i \frac{L_t}{s} \frac{e}{d} \frac{1}{1 - \bar{\varkappa}^2} + a \frac{\Delta p}{\mu n}\,, \tag{31}$$

where we have designated in accord with the introduced symbolics, the dimensionless flow rate under the drag flow $\dot{V}_d/nd^3$ by the symbol a. The value of the coefficient a depends on the geometry of the screw and may be determined from Eq. (*33*) in ref.[1].

Eq. (*31*) may be written in a more compact form as

$$P^* = a_p + a \frac{\Delta p}{\mu n}\,, \tag{32}$$

where the coefficient a_p incorporating the first two terms on the right hand side of Eq. (*31*) expresses the net power input for the drag flow in a screw channel and in the clearance between the screw flight and the barrel.

LIST OF SYMBOLS

a	dimensionless pumping capacity under the drag flow
a_p	dimensionless power input under the drag flow
a_1	coefficient in Eq. (*15*)
b	width of the blade in tangential direction, m

c	clearance between screw flight and barrel, m
d_1	diameter of screw root, m
d	diameter of rotor, m
D_t	diameter of barrel, m
e	axial thickness of screw flight, m
F	force, N
f, f', f''	functions defined by Eqs (*20*)
G_d	correction coefficient incorporating the effect of flight on power input under tangential flow
G_{dc}	correction coefficient incorporating the effect of helicoidal flow on power input
G_{dm}	correction coefficients respecting the effect of flight on helicoidal flow
H	depth of channel, m
i	number of flights
I_1, K_1	modified Bessel functions of the first order and the first and second kind
K_1	coefficient in Eq. (*14*)
L	length of screw, m
L_c	length of cylinders, m
L_k	axial length of screw channel, m
L_t	length of barrel, m
L_z	length of screw channel, m
M_k	torque, Nm
n	frequency of revolution, s^{-1}
p	pressure, Pa
Δp	pressure drop per unit length L
P	power input, W
P^*	$= P/\mu n^2 d^3$ dimensionless power input
P_1	power input per single screw channel, W
R	radius of rotor, m
R_1	radius of rotor root, m
R_2	radius of barrel, m
s	pitch of screw, m
S	area, m^2
u	velocity, m s^{-1}
U	longitudinal component of velocity of mobile plate, m s^{-1}
U_t	peripheral velocity of outer cylinder, m s^{-1}
V	transverse component of velocity, m s^{-1}
$\dot{V}$	volume flow rate, m^3 s^{-1}
W	width of channel, m
x, y, z	Cartesian coordinates
r, φ, z	cylindrical coordinates
α	coefficient defined by Eq. (*25*)
φ_t	$= \text{arctg}(s/\pi D_t)$ helix angle of screw surface on diameter D_t
φ_d	helix angle of screw surface on diameter d
$\varkappa$	$= d_1/D_t = R_1/R_2$ relative diameter of root
$\bar{\varkappa}$	$= d/D_t = R/R_2$ relative diameter of screw
ω	angular velocity, s^{-1}
μ	dynamic viscosity, Pa s
τ	dynamic stress, Pa

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