THE PUMPING CHARACTERISTICS OF SCREW ROTORS. II. MEASUREMENT OF PUMPING CHARACTERISTICS OF SCREW ROTORS

František RIEGER

*Faculty of Mechanical Engineering,
Czech Technical University, 166 07 Prague 6*

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The contribution has been devoted to experimental determination of the pumping characteristics of screw rotors. The paper presents the geometrical parameters of investigated screw rotors and barrels, description of experimental equipment, obtained experimental results and their evaluation.

Although screw rotors are frequently used for transportation of highly viscous materials and a great deal of effort, summarized in the first part of this communication[1], has been devoted in the literature to the theoretical calculation of their pumping capacity, there are relatively few papers dealing with their experimental verification.

The first results of measurements of the pumping capacity of screw rotors were presented already by the pioneers of their theoretical calculation, Rowell and Finlayson[2]. Measurements of pressure exerted by a screw rotor with the closed outlet pipe were carried out by Pigot[3]. Additional measurements on rotors with a relatively shallow screw channel were presented by McKelvey[4]. The measurements with an open outlet for several screw rotors with considerably different geometry were carried out by Squires[5]. Apart from the just mentioned papers there are numerous measurements described in the literature, carried out with polymers exhibiting non-Newtonian behaviour; for this reason these papers were found unsuitable for the verification of the relationships for the calculation of the pumping capacity in the first part of this paper. The majority of the cited measurements were carried out with rotors characterized by a shallow channel and very small clearance between the screw and the barrel. The measurements were not carried out systematically and, as such, are not suitable for an overall verification of the recommended equations. This is the principal reason for undertaking an extensive series of measurements the results of which are presented in this part of the communication.

EXPERIMENTAL

The measurements were carried out on two set-ups. Namely a set-up with a rotating screw and a set-up with a stationary screw.

The set-up with the rotating screw designed to measure the pumping capacity is shown in Fig. 1. A screw **7** is supported in sliding bearings **5** and **6** and rotates in a fixed barrel **4**. The pressure at the inlet end is maintained at a fixed level by an overflow **1**. The overflowing liquid proceeds *via* an overflow tube **8**. The hydraulic resistance at the outlet end of the screw could be varied by a throttling valve **3**. This permitted adjustment of the outlet pressure, monitored by a manometric tube **2**.

The whole set-up is shown schematically in Fig. 2. A driving unit used to power the screw was equipped with a hydromotor JHMA 1 (position **2**). The pressurized oil was supplied by a pumping station **3**. A continuous regulation of r.p.m. in the range between 10 and 1 600 l/min was enabled by regulation valves **6**. The r.p.m.'s were measured photoelectrically using a photocell **9**. The signal of the cell **9** was fed into the transducer Tesla BP 3 620 — position **7**. The value of r.p.m.'s could be read off the display of the counter Tesla BM 362 (position **5**). Liquid entered the measuring equipment position **1** by the action of its own pressure head from the upper storage tank **10** and its flow rate was regulated by a clamp **8** so as to maintain an approximately constant flow rate of liquid through the overflow in all measurements. The pumped amount of liquid was measured by volume in a measuring cylinder **12**. The liquid pumped by the screw, as well as the excess liquid from the overflow coming through the pipe **13**, was collected in the lower storage tank **11** and from here returned to the upper storage tank by a pump **4**.

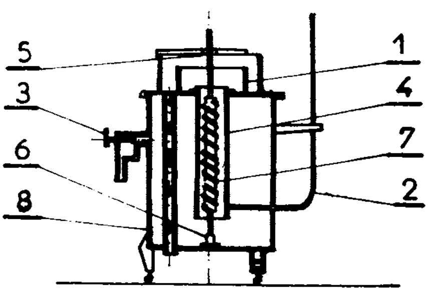

FIG. 1

The set-up with rotating screw. **1** overflow, **2** manometric tube, **3** throttle valve, **4** barrel, **5**, **6** sliding bearing, **7** screw

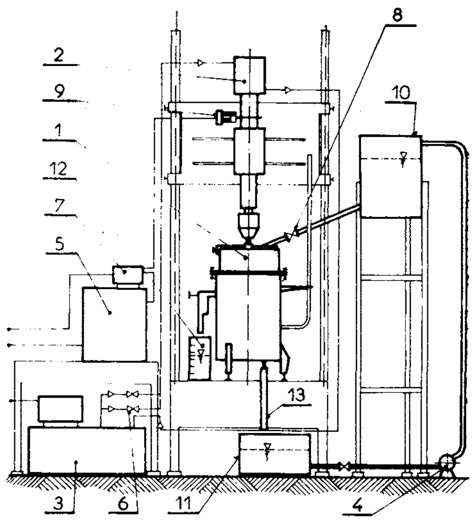

FIG. 2

The overall outlay of the experimental unit with rotating screw. **1** equipment with the rotating screw, **2** driving unit, **3** pumping station, **4** pump, **5** revolution counter, **6** regulating valves, **7** transducer, **8** regulating clamp, **9** photocell, **10** storage tank, **11** lower storage tank, **12** measuring cylinder, **13** tube

The set-up with a stationary screw, shown schematically in Fig. 3, served for additional measurement of the pressure flow and enabled elimination of the end effects. The principal part of the set-up was a barrel **1** housing investigated screw rotor **2**. Position of the rotor was fixed by two cone centers **4** and **5**. In the upper part of the barrel **1** there were openings serving as pressure ports for manometric tubes monitoring the profile of the pressure along the axis of the screw channel. The distance of the openings was 22 mm and was selected so as to make the pitch of all investigated screws an integer multiple of this distance. Thus we could measure the pressure profile along the screw channel. The manometric tube was fixed in a clamp **7**. The liquid was fed into the inlet chamber **2** and discharged from the exit chamber **3**. Partition **6** located in the exit chamber serves for good visibility of liquid in the last manometric tube.

The overall flow sheet of the set-up with the stationary screw is shown in Fig. 4. The measuring equipment proper described in Fig. 3 is in position **1**. The liquid was fed from an upper overflow storage tank **2** into the overflow **3**. The role of the overflow was to maintain constant pressure difference during a single measurement. The magnitude of this pressure difference on the screw was adjusted by a throttle clamp **6**. A measuring cylinder **7** served to measure the volumetric flow rate. The excess liquid from the overflow **3** was collected in the bottom storage tank **4** and from here was pumped by a gear-wheel pump **5** to the upper storage tank **2**. Individual components of the set-up were connected by rubber tubings.

The measurements were carried out with six single threaded screw rotors differing mutually by the diameter of the root, d_1 and the pitch, s. Their dimensions are shown in Table I. The clearance between the screw and the barrel was altered by replacing the barrels. Four barrels were used in the set-up with rotating screw, designated by capital letters A, B, C, and D. Three barrels were employed on the set-up with a stationary screw, designated by lower key letters a, b, c. The internal diameters of the used barrels are summarized in Table II. Oil and solutions of starch syrup with the viscosity ranging between 0·05 and 5 Pa.s were used for measurements.

RESULTS

Six screws and four barrels were used on the rotational set-up. This enabled measurement of altogether 24 screw–barrel configurations. The measurements for each con-

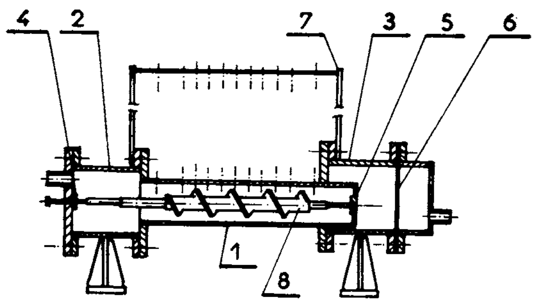

FIG. 3

The set-up with stationary screw. **1** barrel, **2** screw rotor, **3** outlet chamber, **4**, **5** cone centers, **6** partition, **7** clamp, **8** screw

TABLE I

Geometrical parameters of investigated screw rotors

Screw No	d, mm	d_1, mm	L, mm	s, mm	e, mm	d_1/d	s/d
1	39·4	20·2	216	22	2·4	0·508	0·558
2	39·6	12·2	220	22	2·2	0·308	0·556
3	39·6	20	220	44	2	0·505	1·11
4	39·6	10·2	220	44	2	0·258	1·11
5	39·6	20	220	66	2	0·505	1·66
6	39·6	10·2	220	66	2	0·258	1·66

TABLE II

Designation and internal diameters of barrels

Designation of barrel	A	B	C	D	a	b	c
D_t, mm	40·6	41·7	44·5	51·2	39·8	41·6	44·0

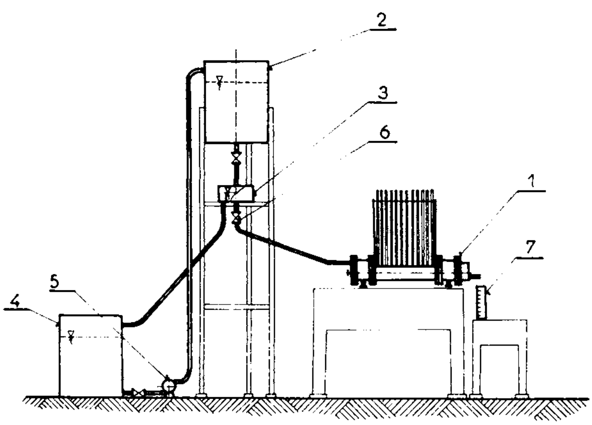

FIG. 4

The overall outlay of the experimental unit with stationary screw. 1 equipment with stable screw, 2 storage tank, 3 overflow, 4 storage tank, 5 gear wheel pump, 6 throttle valve, 7 measuring cylinder figuration were carried out with several liquids. The measurement with each liquid was carried out for several preselected values of r.p.m.'s. For each value of r.p.m. the pumping capacity, $\dot{V}$, was measured for various values of the pressure, Δp, adjusted by the throttle valve. Thus we obtained a total of 1 554 pairs of Δp and $\dot{V}$ data. All obtained pairs Δp and $\dot{V}$ were transformed into a dimensionless form of a pair of the dimensionless pressure difference, $\Delta p/\mu n$ and the dimensionless pumping capacity, $\dot{V}/nd^3$. All measurements were carried out in the creeping flow region at the values of the modified Reynolds number $Re_1 = nd^2\varrho/\mu$ less than 10.

The pairs of data $\Delta p/\mu n$ and $\dot{V}/nd^3$, obtained for a single configuration, served to construct the pumping characteristic of the given configuration. The dimensionless pumping characteristics for selected configurations are shown in the three following figures.

Fig. 5 shows the characteristics of two screw rotors differing mutually in the diameter of the root. Fig. 5*a* presents the characteristics obtained for the configuration 1A; Fig. 5*b* the characteristics for the configuration 2A. From the figure it is apparent that both characteristics are linear in agreement with the following theoretical

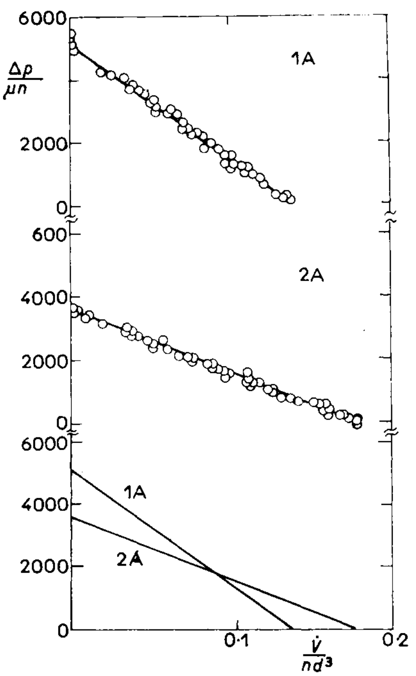

FIG. 5

Characteristics of screw rotors with different screw root diameter

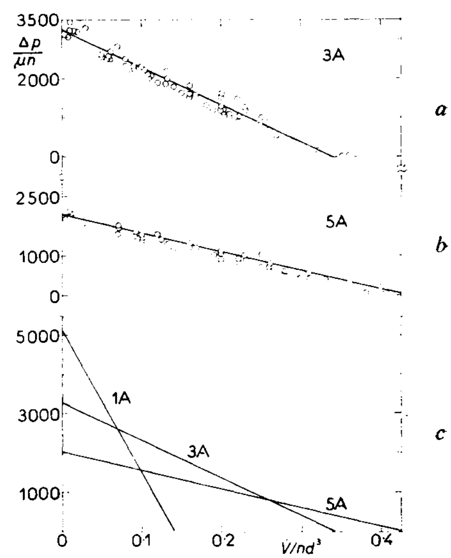

FIG. 6

Characteristics of screw with different pitch equation

$$\frac{\Delta p}{\mu n} = A - B \frac{\dot{V}}{n d^3} . \tag{1}$$

The equation was derived in the first part of this communication[1], presented there as equation (*35*). The straight lines in Figs 5*a* and 5*b* were obtained by linear regression from the experimental values. The intersection of the characteristics with the vertical axis indicated the maximum value of the dimensionless pressure difference between the outlet and the suction end exerted by the screw with the closed outlet. The intersection of the characteristics with the horizontal axis indicates, on the contrary, the maximum value of the pumping capacity achieved by the screw discharging into an open space. From Fig. 5*c*, showing both characteristics together, it is apparent that the screw **1**, exhibiting a greater diameter of the root, provides greater pressure with the closed outlet than the screw **2**. The screw **2**, with a smaller diameter of the root, provides, on the contrary, a greater pumping capacity with open discharge, than the screw **1**.

Fig. 6 shows the experimental characteristics of screws with different pitch. Fig. 6*a* presents the characteristics of the screw rotor **3** in the barrel A, while Fig. 6*b* the characteristic of the rotor **5** in the same barrel. The effect of the pitch is well apparent from Fig. 6*c*, showing summarily the resulting characteristics for the screws **1**, **3**, and **5** in the barrel A, taken over from Figs 5*a*, 6*a*, and 6*b*. From the figure it is apparent that maximum pressure with the closed outlet is achieved by the screw rotor **1**, exhibiting minimum pitch. Maximum pumping capacity with open discharge is exhibited, on the contrary, by the screw **5** with the highest pitch of all three screws.

The influence of the clearance between the screw and the barrel on the pumping characteristic is illustrated in Fig. 7. Fig. 7*a* shows the characteristic of the screw **3** in the barrel B. The characteristics of the same screw rotor in the barrels C and D are then shown in Figs 7*b* and 7*c*. The evaluated characteristics, measured with the screw rotor **3** in all four barrels, taken over from Figs 6*a*, 7*a*, 7*b*, and 7*c*, are summarily shown in Fig. 7*d*. The configuration with the barrel A exhibits minimum clearance while for the barrel D the clearance between the screw and the barrel is maximal. From Fig. 7*d* it is apparent that with increasing clearance the maximum pressure reached by the screw significantly decreases, while the maximum flow rate at zero pressure difference is affected by the clearance far less significantly.

The values of the constants from Eq. (*1*) for all configurations investigated on the set-up with rotating screw are summarily presented in Table III. The given values of the constants, together with their 95% confidence limits, were obtained by linear regression from experimental data. The values of the constant A indicate maximum dimensionless pressure difference that may exist in the set-up with its discharge end closed. The value of the constant B characterizes the hardness of the pumping characteristic. The high values of both of these constants are desirable especially in those situations where the screw must overcome high hydraulic resistances. From this standpoint as most advantageous appears the screw number 1, exhibiting relatively large diameter of the root, low pitch and small clearance between the screw and the barrel. The value of the maximum dimensionless pumping capacity, reached by the screw at zero gauge-pressure at the outlet end, characterizes the ratio of both constants A/B equalling the constant a from Eq. (*34*) of the first part of this communication[1]. The values of this ratio are presented in the last column of Table III. From these values it is apparent that, judged by this criterion, the most advantageous screw is the screw rotor number 6, exhibiting a relatively large pitch and small diameter of the root. Such rotors are advantageous in those situations where there is a need for a large pumping capacity against a small pressure.

Measurement of the Hydraulic Resistance of Screw Rotors

The hydraulic resistance was measured on the set-up with stationary screw. A typical profile of pressure along the screw, measured by the manometric tubes, is shown in Fig. 8. From this figure it is seen that the height of the liquid in the manometric

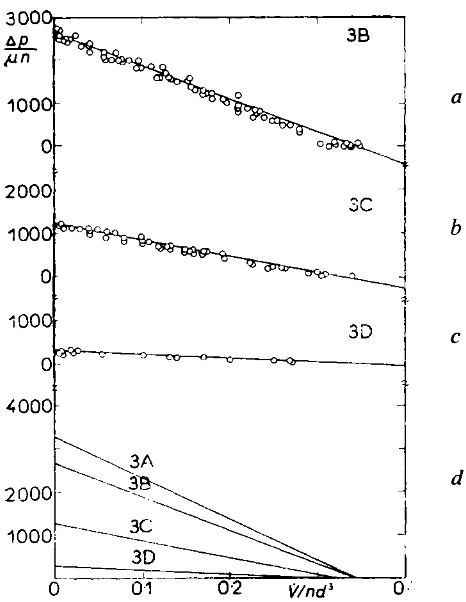

FIG. 7
The effect of clearance between the screw and the barrel on the pumping characteristic

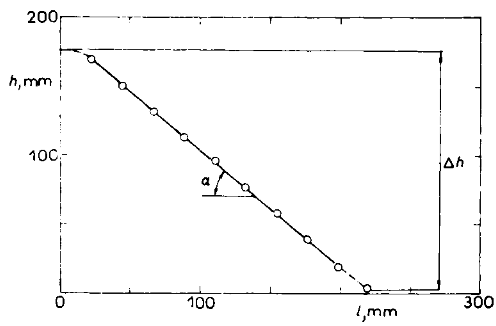

FIG. 8
The pressure profile along the screw

tubes, with the exception of the end regions, decreases linearly along the length of the screw channel. From the slope of this straight line we have computed the value of the pressure gradient in region of stabilized flow $(\partial p/\partial l)_s$. From the difference of the levels of the outermost tubes, located just in front of the inlet and just past the exit end of the screw, we have evaluated the pressure difference across the screw Δp.

The experimentally obtained values of the pressure difference, Δp, were processed into the form of a dimensionless criterion $\Delta p d^3/\mu \dot{V}$ and the values of the pressure gradient in region of the stabilized flow, $(\partial p/\partial l)_s$, were processed into the dimensionless form $(\partial p/\partial l)_s\ d^4/\mu \dot{V}$. Typical courses of the dimensionless pressure difference $\Delta p d^3/\mu \dot{V}$, and the dimensionless pressure gradient, $(\partial p/\partial l)_s\ d^4/\mu \dot{V}$, as functions of the modified Reynolds number, $\mathrm{Re}_2 = \dot{V}\varrho/\mu d$, computed from all experimental values obtained with several liquids for the configuration 2a, are shown in Fig. 9. From

TABLE III

Results of measurements with the rotating screw

Screw	Barrel	No of measurements	*A*	*B*	*a*
1	A	100	5 152 ± 57	36 556 ± 640	0·141
1	B	126	3 718 ± 69	26 090 ± 854	0·143
1	C	116	1 216 ± 21	9 293 ± 280	0·131
1	D	55	229 ± 8	2 029 ± 130	0·113
2	A	79	3 613 ± 40	19 691 ± 354	0·183
2	B	66	3 286 ± 63	18 559 ± 612	0·177
2	C	94	1 402 ± 24	8 392 ± 262	0·167
2	D	66	284 ± 8	1 758 ± 91	0·162
3	A	83	3 284 ± 98	9 578 ± 494	0·343
3	B	85	2 662 ± 30	7 789 ± 153	0·342
3	C	71	1 263 ± 25	3 886 ± 147	0·325
3	D	26	299 ± 10	993 ± 62	0·301
4	A	58	1 459 ± 21	3 362 ± 93	0·434
4	B	47	1 427 ± 29	3 147 ± 120	0·453
4	C	63	914 ± 12	2 035 ± 56	0·449
4	D	24	342 ± 17	767 ± 94	0·445
5	A	62	2 030 ± 38	4 778 ± 159	0·425
5	B	64	1 668 ± 34	3 846 ± 157	0·434
5	C	67	931 ± 15	2 272 ± 73	0·410
5	D	23	279 ± 11	727 ± 70	0·384
6	A	47	967 ± 23	1 752 ± 76	0·552
6	B	54	955 ± 19	1 727 ± 74	0·553
6	C	54	661 ± 14	1 188 ± 70	0·556
6	D	24	287 ± 8	523 ± 41	0·548

this figure it is apparent that neither the dimensionless pressure difference, nor the pressure gradient depend in the creeping flow region on the value of the modified Reynolds number and remain constant.

TABLE IV
Results of measurements with the stationary screw

Screw	Barrel	No of measurements	$\frac{\Delta p d^3}{\mu \dot{V}}$	No of measurements	$\left(\frac{\partial p}{\partial l}\right)_s \frac{d^4}{\mu \dot{V}}$
1	a	13	49 530 ± 1 421	20	9 096 ± 163
1	b	21	30 067 ± 1 820	21	5 396 ± 362
1	c	13	13 031 ± 1 404	25	2 261 ± 133
2	a	22	21 490 ± 713	29	4 040 ± 77
2	b	13	17 454 ± 617	13	3 098 ± 267
2	c	6	8 190 ± 215	17	1 689 ± 117
3	a	17	11 533 ± 322	34	2 341 ± 56
3	b	17	8 754 ± 329	34	1 669 ± 31
3	c	5	5 373 ± 236	32	1 067 ± 30
4	a	13	4 168 ± 130	26	750 ± 14
4	b	6	3 219 ± 222	12	635 ± 25
4	c	5	3 085 ± 289	30	540 ± 19
5	a	13	6 451 ± 156	39	1 244 ± 30
5	b	15	4 816 ± 187	45	900 ± 19
5	c	5	3 374 ± 130	42	557 ± 18
6	a	12	2 402 ± 78	36	438 ± 10
6	b	12	2 116 ± 76	36	377 ± 12
6	c	—	—	24	265 ± 8

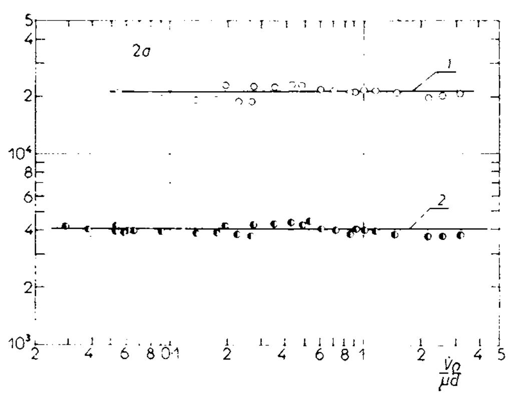

FIG. 9
Dependence of the dimensionless pressure difference on Re_2 for screw 2a. *1* $\Delta p d^3/\mu \dot{V}$; *2* $(\partial p/\partial l)_s d^4/\mu \dot{V}$

The values of these constants, calculated as an arithmetic average from all experimental data, are summarily shown, for all experimental configurations, in Table IV. From the values presented in this table it follows that the maximum hydraulic resistance exhibit screw with a small pitch, relatively large root and small clearance between the screws and the barrel. With increasing lead, decreasing diameter of the root and increasing clearance between the screw and the barrel the hydarulic resistance decreases.

The main purpose of the measurements, carried out in this work, was to verify the theoretical equations[1] for the calculation of the pumping capacity of screw rotors. The last of the series of papers dealing with the pumping characteristics of screw rotors shall be devoted to this problem.

LIST OF SYMBOLS

A		coefficient in Eq. (*1*)
B		coefficient in Eq. (*1*)
d		diameter of screw rotor, m
n		frequency of revolution, s^{-1}
l		coordinate of length, m
p		pressure, Pa
Re_1	$nd^2\varrho/\mu$	modified Reynolds number
Re_2	$\dot{V}\varrho/\mu d$	modified Reynolds number
$\dot{V}$		volumetric flow rate $m^3 s^{-1}$
μ		dynamic viscosity Pa.s
ϱ		density, $kg\ m^{-3}$

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