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Investigating the Effect of Geometric Parameters of Screw Pump (Single-rotor hydraulic-machine) on its Operation using Different Kinematic Ratios

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Abstract- The appearance of new technologies in the second half of XX century in various industries, the progress of energy and oil and gas complexes caused rapid development of hydro-machine construction.

Development and improvement of hydraulic machines held by the expansion of the range of known types of pumps and hydraulic engines helped in the development of a fundamentally new dynamic and volumetric machines as identified by Baldenco (1979).

Among the volumetric hydraulic machines in general engineering and in the oil and gas industry, single-rotor hydraulic-machines play a noticeable role and are predominately used with cycloidal working bodies as pumps and motors. Simplicity of design and unique features of single-rotor hydraulic-machines (particularly, operation at high viscosity, gas content and pollution of working environment) make them effectively be used in a variety of industrial processes (Karahnayan, 2003).

The focus of this research work is to study and calculate the basic geometric parameters of screw pump (single-rotor hydraulic-machine) using different kinematic ratios (ratio of the threads of lead rotor to that of lead stator). The dependence of the parameters of the screw pump was studied and plotted. The most energy-effective screw pump with a given kinematic ratio was determined.

Index Terms— Volumetric hydraulic machines, Single- rotor hydraulic machines, Screw pump, Geometric parameters, Kinematic ratio, Rotor, Stator.

1 INTRODUCTION

1.1 Structure and principle of operation of screw pump

Screw pump is a device for pumping fluids, which was developed in the early 1920s for pumping viscous liquids and solutions. From the outset, screw pumps are widely used in different conditions in various industries (chemical, food, metal, paper, textile, tobacco, oil and waste recycling). Since the first serious attempts to use screw pump for artificial lift in the early 1980s, there was a gradual implementation of it in the petroleum industry (Kasyanov, 1981).

Baldenco, (1979) showed that screw pump systems have a number of distinctive features, which can make them preferable to artificial lift in comparison with other available technical means. Here are the most important of these features:

- The efficiency of the systems of screw pumps is 50 - 70%;
- Low capital expenditures and energy expenses;
- The ability to pump liquids with high viscosity, high-solids contents and free gas;
- The absence of valves or components with reciprocating prevents clogging, gas slugs or wearing of components;
- Easy installation and operation, minimum maintenance is required;
- Small size and low level of noise caused by the device at the top. Screw pump systems have a number of specific restrictions on the application condition. The most important of these limitations are performance, level of liquid rise and compatibility of rubber parts with pumped fluids. Below is a brief list of restrictive conditions and the operational problems associated with the use of screw pump systems;
- Performance 1-800 m^3 / day (5,000 bbl / day);
- Level 3000 meters (9800 feet) of liquid rise ;
- 150 ° C (300 ° F) of temperature;
- The tendency of occurrence of permanent damages of elastomeric parts in the pump even without liquidis very limited;.
- The impact of some liquids leads to swelling and damage of the elastomeric material.

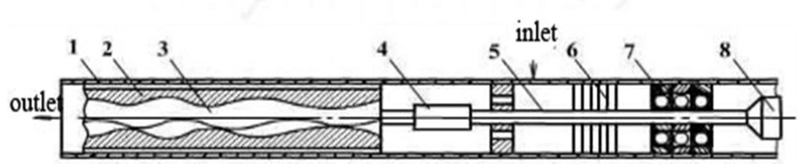

Figure 1.- Structure of single screw pump for oil fields (Baldenco, 1979)

1-pump casing; 2- liner; 3- screw; 4- eccentric coupling; 5- middle drive shaft; 6- seal device; 7- radial thrust bearing; 8- common coupling.

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Principle of operation of the screw pump:

• Design of the pump and the rotor comprises of a holder generally similar in construction to mud hydraulic motor;

• Figuratively speaking, screw pump can be linked to a conveyor assembly line; the higher the rate - the greater the productivity;

• In the cavity of the pump rotor is a metal and rubber ferrule. When the rotor chamber formed by the surfaces of the screw and the holder are moved spirally in the direction from the suction nozzle it provides fluid flow in a desired direction;

• The rotor is in the form of a screw auger and the wing has a single-filar or bifilar helical thread. Helical pitch of the cage is twice the pitch of the screw rotor;

• Compaction between the surface of the ferrule and rotor enable pumping of fluid with contained therein gases or solids;

• Pumped fluid acts as a lubricant to the contacting surfaces of the rotor and ferrule;

• Screw pumps can operate at a depth up to 2440 m;

• Maximum discharge is 3600 m, which corresponds to the pressure drop in the pump of 1560 m / sq. inches;

• Flow rate of the liquid is 5 - 5,300 barrels per day;

• Rotational speed of the rotor of the pump is 100 - 1000 rev/ min; and for the actual operating conditions more typical range is 100 - 500 rev/min;

• Pump manufacturers indicate their performance as a pumped volume of fluid per day based on the predetermined number of revolutions;

• It's allowed to be operated at temperatures up to about 350 ° F (restrictions are determined by the properties of the pump elastomer used in the construction)(Baldenco, 1995).

1.2 Kinematic ratio of the operative parts

Operative parts of a single- screw hydraulic machine is a gyrator mechanism - helical gear pair of internal space of cycloidal gearing consisting of z_2 - lead-rotor metal (screws) and z_1 - lead - stator (sleeve with elastic lining) screw spaces which are formed from the working chambers (Baldenco and Gnoevikh 2005.).

The overall aim of this study is to : a) calculate the basic geometric parameters of screw pump using different kinematic ratios; b) study the dependence of the parameters of the screw pump on the kinematic ratios; c) determine the most energy-effective screw pump with a given kinematic ratio.

Foam EOR partially overcomes the effects of poor sweep efficiency by reducing gas mobility (Schram, 1994; Rossen 1996). Foam can be classified as a mixture of gases and liquids, specifically, gas bubbles dispersed in liquid separated by liquid films called lamellae. Foam greatly increases the gas flow resistance, as it has higher viscosity and thus diverts the gas from unwanted layers. Unfortunately, the contact of foam with most crude oils destabilizes foam, which greatly limits the widespread use of foam for EOR (Mannhardt et al., 1998). Thus, it is important to understand the complexities involved in the effect of oil on foam.

2 METHODOLOGY

2.1 Calculation of the geometric parameters of the screw pump

Based on the aims of this research work, the basic geometric parameters of the screw pump was calculated using the following given data for pumping viscous liquids at normal temperature.

Fluid flow rate - Q = 24 m³ / day (0,28 *l*/s); Pressure - P = 15 MPa, and Average differential pressure P_a = 0.25MPa.

In this case, it is advisable to use a pump with multiple-thread operative parts at a rated speed of the drive shaft n = 150 rev /min. (Karahanyan, 2003; Baldenco and Gnoevikh, 2005).

At kinematic ratio: i = 4: 5 ($i = Z_2 : Z_1$)

Where;

Z_2 = 4, lead rotor thread number (screw)

Z_1 = 5, lead stator thread number

Pump volumetric capacity

$$V = \frac{Q}{n\eta_0} = \frac{60 \cdot 0{,}28}{150 \cdot 0{,}8} = 0{,}14l \tag{2.1}$$

Eccentricity

$$e = \sqrt[3]{\frac{V}{z_1 s_T \pi\left(\overline{D_c} - 3\right)\left(\overline{D_c} - 4\right)}}. \tag{2.2}$$

Taking the normalized values of the coefficients of non-centrode and thread forms (c_0= 1.175, s_e = 2.175), and the optimal range of the coefficient of surface configuration S_T= 1.5 - 3.5, Baldenco defined the eccentricity of the required number of thread of rotor (Z_2 = 4) for gearing without displacement (ζ = 0).(Baldenco and Gnoevikh, 2005).

Where : $\overline{D_c}$ - average contour diameter

$$\overline{D_c} = 2\left(z_2 c_0 + s_e + \xi + 1\right) = 2\left(4 \cdot 1{,}175 + 2{,}175 + 0 + 1\right) = 15.75 \tag{2.3}$$

$$e = \sqrt[3]{\frac{0{,}14}{5 \cdot 2{,}5\pi\left(15{,}75 - 3\right)\left(15{,}75 - 4\right)}}. \approx 3\text{mm}$$

Based on the analysis of criteria performance and available nomenclature of grooving tool,we select the option of operative parts: Z_2 = 4; e = 3.0 mm).(Baldenco and Gnoevikh, 2005).

Contour diameter

$$D_c = 2e\,(C_0\, Z_2 + s_e + 1) = 47.3 \text{ mm.} \tag{2.4}$$

The average diameter of rotor (2.5)

$d_{av} = D_c - 4e = 35.3$ mm.

Flow area

$S = \pi e\,(D_c - 3e) = 0{,}04\ dm^2$. (2.6)

Axial stroke of the threads

$$t_0 = \frac{V}{z_1 z_2 s} = \frac{0.14}{5 \cdot 4 \cdot 0{,}04} = 0.18\text{dm} \approx 18\text{mm} \quad (2.7)$$

Steps of helical surfaces of the stator and rotor

$T = Z_1 t_0 = 5 \cdot 18 = 90$mm; (2.8)

$t = Z_2 t_0 = 4 \cdot 18 = 72$ mm. (2.9)

Surface configuration coefficient of rotor

$S_t = t / d_{av} = 72 / 35.3 = 2.04$. (2.10)

The required number of steps of operative parts

$$k = \frac{1}{z_1}\left[\frac{p}{[P_a]} + z_2\right] = \frac{1}{5}\left[\frac{15}{0{,}25} + 4\right] = 13 \quad (2.11)$$

The length of operative part

$L = kT = 13 * 90 = 1170$ mm. (2.12)

Flow moment determination :

$$M_f = \frac{4PRt^2}{\pi^2} = \frac{4 \cdot 15 \cdot 10^6 \cdot 0.0177 \cdot 0.072^2}{3.14^2} \approx 560Nm \quad (2.13)$$

Torque moment of stator determination:

$$M = M_0 P e^2 T, \quad (2.14)$$

$$M = 23 \cdot 15 \cdot 10^6 \cdot 0.003^2 \cdot 0.09 = 280\,Nm$$

Where - M_0 is the specific torque moment, which is the torque moment for screw hydraulic machines with single dimensions (e, *T*) and single pressure drop (Karahanyan, 2003).

From general theory of positive-displacement hydraulic machines M_0 is defined from the formula given below (Karahanyan, 2003).

$$M_0 = z_2\left(z_2 - 1 + \frac{4S_e}{\pi}\right). \quad (2.15)$$

$$M_0 = 4\left(4 - 1 + \frac{4 \cdot 2{,}175}{\pi}\right) = 23Hм$$

Determination of the stator power

$$N = M\omega, \quad (2.16)$$

Where; $\omega = \frac{\pi n}{30} = 15.7\,grad / \min$

$$N = 280 \cdot 15.7 = 4.4\,kWatt$$

2.2 Further study of geometric parameters of screw pump with kinematic ratio 2:3

The minimum and maximum diameter of the screw surface of rotor

$d_{min} = D_C - 6e = 23$ mm. (2.17)

$d_{max} = D_C - 2e = 41$ mm. (2.18)

The average diameter of the operative part of the pump

$D_{OP} = D_C - 3e = 36.5$ mm. (2.19)

Angle of inclination of thread

$\Theta = \text{arctag}\,(S_T / \pi) = \text{arc tag}\,(2.8 / \pi) = 41.70^0$ (2.20)

Pump capacity per revolution (actual)

$V = Z_2 ST = 2 \cdot 0.516 \cdot 135 = 0.138\ l$ (2.21)

The projected area of the contact lines, responsible for axial forces in the identities of the shaft and the pump housing

$$S_a = \frac{\pi}{4} D_{op}^2 = \frac{\pi}{4} 0{,}0365^2 = 11\text{cm}^2 \quad (2.22)$$

Number of contact lines separating the entrance and exit

$\Lambda = (k - 1)\, Z_1 + 1 = (21 - 1)\, 3 + 1 = 61$ (2.23)

Average inter-track differential pressure (actual)

$P_{av} = P / \Lambda = 15/61 = 0.25$MPa (2.24)

The length of the contact line of the operative parts

$$L_c \approx k\left(z_1\sqrt{t^2 + \pi^2 D_{op}^2} + T\right) = 21\left(3\sqrt{90^2 + \pi^2 36{,}5^2} + 135\right) = \quad (2.25$$
$$= 12016\,mm$$

Rotor mass

$$m = \rho_m \frac{\pi d_{av}^2}{4} L = 7850 \frac{\pi \cdot 0.032^2}{4} 2.835 = 17.9\,kg \quad (2.26)$$

Where; ρ_m - density of the material of the rotor (for steel $\rho_m = 7850 kg / m^3$)

Maximum slip velocity

$$v_{sl\max} = 2\pi ne\left[z_2\left(c_o + 1\right) + s_e\right] =$$
$$2\pi 150 \cdot 4{,}5 \cdot 10^{-3}\left[2\left(1{,}175 + 1\right) + 2{,}175\right] = 0{,}46м/c \quad (2.27)$$

Inertial force

$$F_{in} = me(2\pi n z_2)^2 =$$
$$= 17{,}9 \cdot 4{,}5 \cdot 10^{-3}(2\pi \cdot 2{,}5 \cdot 2)^2 = 79{,}4N \quad (2.28)$$

Specific inertial force

$$\overline{F}_{in} = \frac{\pi}{4} z_2^2 e d_{av}^2 = \frac{\pi}{4} 2^2 \cdot 4{,}5 \cdot 10^{-3} \cdot 0{,}032^2 = 1{,}45 \cdot 10^{-3} N \quad (2.29)$$

Velocity of the fluid in the channels of the operative parts

$$V = z_2 T n \sqrt{1+\left(\frac{\pi}{S_T}\right)^2} = 2\cdot 0{,}135\cdot 2{,}5\sqrt{1+\left(\frac{\pi}{2{,}8}\right)^2} = 1{,}02\,\mathrm{m/s}$$

(2.30)

Calculated the fluid velocity in the channels of the operative parts for different values of T (ν = f (T)) and S_T (ν = f (S_T)). The calculated results are summarized in table 2.

3 RESULTS & DISCUSSIONS

Using the same pattern and data given, same geometric parameters were calculated for kinematic ratio of *i* = 3:4*, i* = 2:3 and *i* = 1:2, as that of *i* = 4:5. Table 1 shows the summarized results.

Table 1: Basic geometric Parameters

Kinematic ratios (*i*) / Parameters	*4:5 (4)*	*3:4 (3)*	*2:3 (2)*	*1:2 (1)*
Pump volumetric capacity , V	0.14*l*	0.14*l*	0.14*l*	0.14*l*
Eccentricity , e	3.0mm	3.6mm	4.5mm	4.3mm
Contour diameter, D_c	47.3mm	48mm	50mm	60mm
Average diameter of rotor, d_{av}	35.3mm	33.6mm	32.0mm	43.0mm
Flow area, S	0.04dm^2	0.042dm^2	0.052dm^2	0.064dm^2
Pitch of helical surfaces of the stator and rotor (T, t) resp.	90mm, 72mm	112mm, 84mm	135mm, 90mm	218mm, 109mm
Coefficient of surface configuration of rotor, S_c	2.04	2.5	2.8	2.5
Length of operative parts of stator, L	1170.0mm	1792mm	2835mm	6758mm
Flow moment, M_f	560Nm	730Nm	789Nm	1555Nm
Torque moment of stator, M	280Nm	312Nm	308Nm	387Nm
Stator Power, N	4.4 kWatt	4.62 kWatt	4.84 kWatt	6.1 kWatt

Table 2.– Fluid velocity in the channels of the operative parts.

T, m	V, m/s	S_T	V, m/s
0,1	0,76	1	2,22
0,2	1,51	2	1,26
0,3	2,27	3	0,98
0,4	3,02	4	0,86
0,5	3,78	5	0,80

The following graphs were plotted for some of the calculated basic geometric parameters dependent on kinematic ratios as shown below.

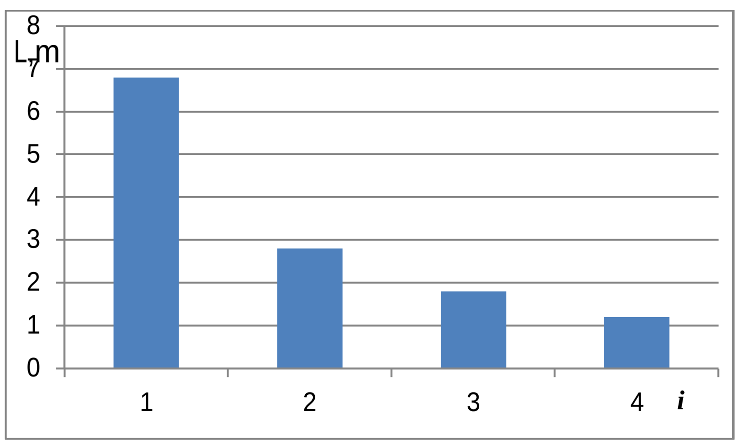

Figure 2. - Length of operative parts of Stator versus kinematic ratios

Figure 2 shows that, the higher the kinematic ratio, the lower the length of operative parts of stator and vise versa.

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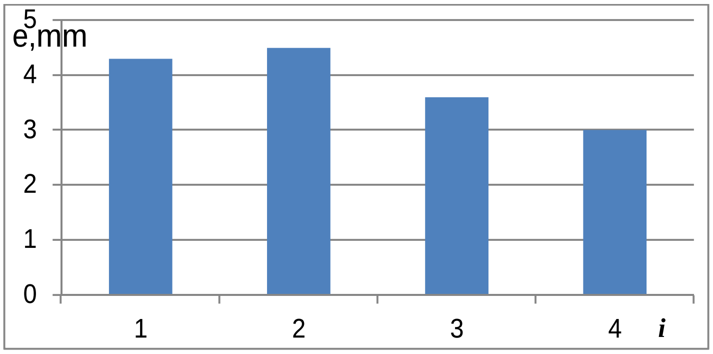

Figure 3.- Eccentricity versus kinematic ratios

Figure 3 shows that, the higher the kinematic ratio, the lower the eccentricity, except for the screw pump with kinematic ratio 1:2, where Z_2 =1 (lead rotor thread number) which gave a lower value of average contour diameter and subsequently a lower value of eccentricity.

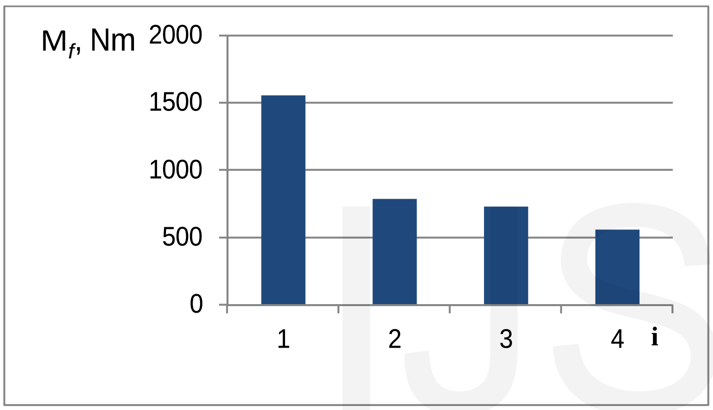

Figure 4.- Flow moment versus kinematic ratios

In figure 4, the flow moment of the pump decreases with an increase in the kinematic ratio and vise versa.

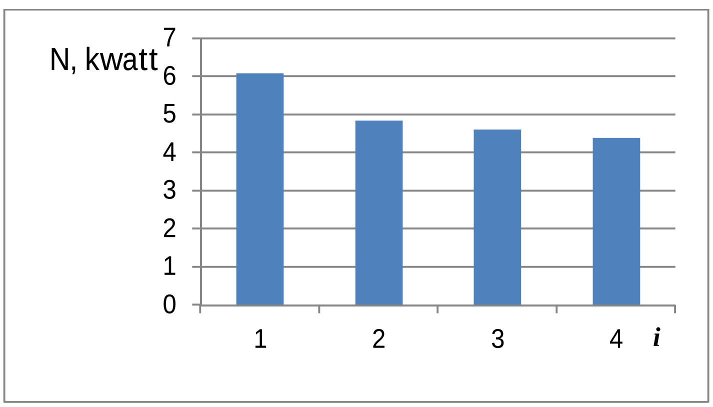

Figure 5.- Stator power versus kinematic ratios

Figure 5 above shows that, the lower the kinematic ratio, the higher the stator power, which is dependent on the diameter of the pitch of helical surface of the stator (T). That is, the higher the diameter of the pitch, the higher the stator power and vise versa.

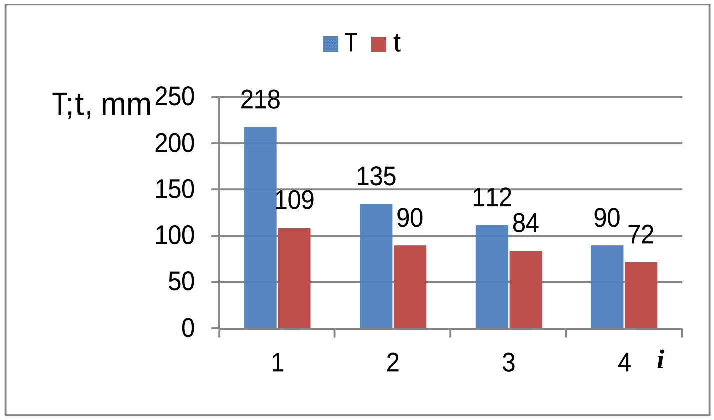

Figure 6. - Pitch of helical surfaces of the stator and rotor (T, t) versus kinematic ratios

Figure 6 shows that, the pitch of helical surfaces of the stator and rotor (T, t,) respectively, decreases, with an increase in kinematic ratio and vise versa.

Plotted the graphs of pitch helical surfaces of the stator and the coefficient of surface configuration dependent on fluid velocity in the channels of the operative parts as shown below.

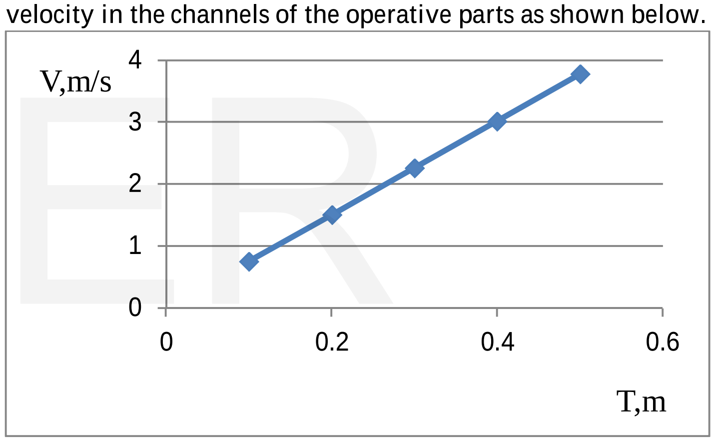

Figure 7.- Fluid velocity in the channels of the operative parts versus pitch of helical surfaces of the stator.

In figure 7 above, the plot shows that, the larger the pitch of helical surface of the stator (T), the higher, the speed of the pump and the dependence is linear.

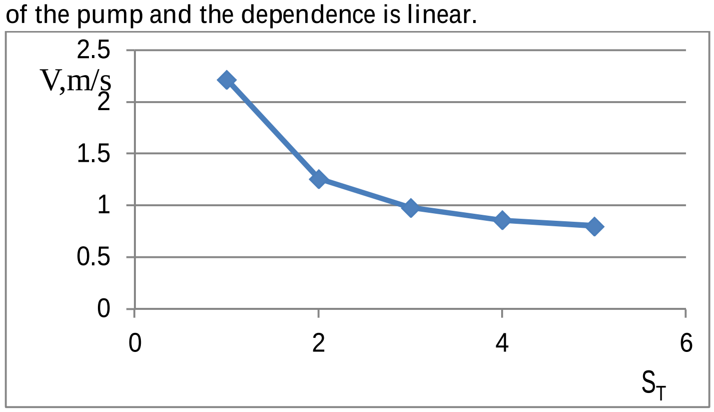

Figure 8. - Fluid velocity in the channels of the operative parts versus coefficient of surface configuration

Figure 8 shows that, the more the coefficient of surface configuration, the more the pump speed and vise versa

4 CONCLUSIONS

● The analysis of literature and patents dealing with designs of screw pumps, showed that most currently used pumps have a kinematic ratio of 1: 2 and 4: 5, which is often not effective for use in specific well circumstances.(Baldenko, 1993; Baldenko and Gnoevikh 2005; karayanhan, 2003; Pat. RU2334125-Denisov, Ratmanov ,Syzrantsev and Plotnikov).

● Laboratory tests showed that the axial load on the rotor of the pump increases with increasing pressure, with an increase in pump pressure, volumetric efficiency falls, and the stronger the pump pressure, the smaller the discharge, while with the increasing thrust of pump its performance decreases.(Karahanyan, 2003 ,Buhalenko, 1990).

● Study of the dependence of the basic geometric parameters of screw pump on the kinematic ratios showed that the most energy-efficient is screw pump with kinematic ratio of 2: 3. (Baldenko and Gnoevikh 2005).

● Study of fluid flow in the pump shows that, the most influencing factors on it are the pitch of the holder and the coefficient of surface configuration. The larger the pitch of holder, the more the speed of the pump, and the dependence is linear. With the increase in coefficient of the surface configuration, pump speed drops, and the drop is exponential.

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