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Original Paper

Theoretical study on the pressurization characteristics of disc-seal single screw pump used in high viscosity oily sludge conveying field

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ABSTRACT

The disc-seal single screw pump (DSSP) used in the field of high viscosity oily sludge transport has a huge advantage. However, there is no research on the pressurization characteristics of the DSSP at present, which makes its application limited. In view of this, the pressurization process mathematical model of the DSSP was established based on the geometric model of the pump. By using this model, the pressurization characteristics of DSSP and the influence of working parameters on the pressurization process were studied combined with the principle of back-flow pressurization. Analysis results show that the instantaneous pressurization process could be realized mainly depending on the reflux pressurization from the outlet chamber to the pressurization chamber when the screw rotor rotating angle is located at $-5°$ to $+5°$. The pressure in the pressurization chamber will increase with the increase of working parameters which include inlet pressure, outlet pressure, screw rotation velocity and dynamic viscosity of fluid medium in the area of flow-back pressurization. The screw rotation velocity and the viscosity of the conveying medium have significant effects on the peak pressure in the pressurization chamber, and the peak pressure in the pressurization chamber is proportional to the screw rotation velocity and the dynamic viscosity coefficient of the conveying medium. The proportional coefficient between the peak pressure and the screw rotation velocity is 6.29×10^4. The proportional coefficient between the peak pressure and the dynamic viscosity of the conveying medium is 6.28×10^6.

1. Introduction

With the continuous development of economic and the progress of human society, the consumption of all kinds of energy resources by human beings is also increasing. So recovery of waste energy and high efficiency utilization of cleaned energy are of great importance (Bauer et al., 2016; Maria et al., 2015; Sadala et al., 2015; Zhang and Luo, 2013). For the considerable quantity of oily sludge generated from various petroleum industries (Hui et al., 2020; Da et al., 2012), it is considered as a potential recycling resource due to its high petroleum hydrocarbon content (15–50% by mass) (Gao et al., 2005; Li et al., 2009; Muzher, 2019). In addition, the oily sludge also contains a large number of toxic substances, including heavy metal pollutants, harmful microorganisms and benzene organic pollutants, if not promptly recovered and treated, will cause serious pollution to the environment (Liang et al., 2019). As a consequence, energy recovery and utilization technology from oily sludge has become a great research focus in recent years (Jublee and Suparna, 2015; Leng et al., 2017; Shen et al., 2016).

Due to the high viscosity of oily sludge, and many large particle impurities such as gravel, garbage, animal carcasses contained in the oily sludge (Evans et al., 2017; Lin et al., 2017; Wang et al., 2018a,b), the traditional fluid pump does not meet its pumping needs. Therefore, a pumping device with strong medium adaptability, large displacement and strong suction capacity is urgently needed. In order to meet the high viscosity medium pressurized transport requirements, the first screw pump was created by Rene Moineau in 1930. However, the traditional screw pump is always in the friction state between the screw rotor and the seal bushing

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Nomenclature

P_d	Outlet section pressure, Pa
P_k	Working chamber pressure, Pa
u	Velocity of fluid, $m \cdot s^{-1}$
h_f	Friction loss
h_w	Local energy loss
l_1	Length of the leakage channel, m
d	Diameter of the leakage channel, m
b	The width of the parallel plate, m
e	Eccentric distance of seal disc, m
R	Seal disc radius, m
s_1	The cross-sectional area near the leakage channel, m^2
s_2	The cross-sectional area of the outlet part, m^2
Y	The expansion coefficient of the internal medium
C	The flow coefficient of the leakage orifice model
A	The cross-sectional area of the leakage channel
Δp	The pressure difference between the two sides of the leakage channel, Pa
d/D	The ratio of orifice diameter to the equivalent inner diameter of the leakage flow chamber
a	Central distance, m
R_{sr}	Screw rotor radius, m

Subscripts

U	The relative velocity of motion between two plates at the parallel plate model
h	The distance between the two plates, m
Δm	The variation of the fluid medium quality in the pressurization chamber per unit time, kg/s
V_p	The volume of the screw chamber
dV_p	The volume increment of the screw chamber in unit time
V	The theoretical drainage volume of DSSP per turn when leakage is considered
V_s	The backflow leakage volume from the pressurized chamber to the inlet position in screw unit rotation angle
n	Screw rotation speed, $r \cdot s^{-1}$
in	The medium that flow into volume
out	The medium that comes out
1	The fluid flows through the initial position
2	The fluid flows through the terminal position
sr	Screw rotor

Greeks

ρ	Liquid medium density, $kg \cdot m^{-3}$
μ	Dynamic viscosity coefficient, $N \cdot s \cdot m^{-2}$
λ	Frictional drag coefficient
ϕ	Screw angle
φ	The expansion coefficient of the material on both sides of the leakage channel

during the working process, the bushing is easy to wear, and its working chamber is small, which limits its application in the field of high viscosity sludge transportation, especially in the field of high viscosity sludge transportation when the oily sludge containing large particles of impurities (Du, 2010; Lin, 2014; Shi et al., 2012; Zhou et al., 2014). In order to solve the problems above, the single screw pump with meshing pair composed by two multiple-tooth metal and screw rotor was proposed (Bjornberg et al., 1982). However, the single working chamber formed by the meshing pair of the multiple-tooth metal sealing disc and the screw rotor is still small, which reduce the utilization rate of the spiral groove and weaken the adaptability to solid impurity. In order to further improve the adaptability of single screw pump to solid particle impurities and the utilization ratio of screw groove and reduce the processing difficulty of screw rotor, the disc-seal single screw pump (DSSP) was proposed by Dr. Andes Johansson of Sweden in 1991. DSSP has a series of advantages of large displacement, compact structure and strong adaptability to solid impurities (Johansson, 1991, 2000). At present, with the development of this new single screw pump and the improvement of its structure, DSSP has shown a strong competitiveness in the fields of oil spill emergency treatment and high viscosity medium transportation.

However, to the best of author's knowledge only a few studies on the pressurization characteristics and process of the DSSP were carried out. The meshing characteristic analysis model and the screw groove volume theoretical calculation model of the disc-seal single screw pump was established according to the space meshing theory and the coordinate transformation principle by Wang et al. (2018) and Wang et al. (2020a), and the mathematical model of the suction capacity was established by Wang et al. (2020b) based on the three-dimensional meshing theory. In these studies, the influence of the structural parameters of the screw pump on the meshing characteristics, the theoretical drainage volume and the suction capacity of the disc-seal single screw pump were analyzed. However, there are few analyses on the pressurization characteristics of DSSP. Shen (2018) carried out a numerical simulation on the pressurized flow process of DSSP, but the theoretical study on the pressurization mechanism of DSSP is still in urgent need. Although the pressure distribution law inside the traditional screw pump was analyzed according to the principle of reflux pressurization (Cao et al., 2001; Ye et al., 2009; Xue et al., 2012; Zhao, 2011), but the traditional screw pump and the disc seal single screw pump are quite different in the original structure principle. Therefore, it is necessary to establish the numerical model of the pressurization process for DSSP to study the pressurization characteristics of the pump.

In view of this, the in-depth study on the pressurization characteristics of DSSP has been carried out in this paper. The numerical model of the pressurization process in the pump cavity was established firstly based on the geometric model and the meshing characteristics of the pump. By using this model, the pressurization law inside the working chamber was obtained, the pressurization characteristics of DSSP and the influence of working parameters on the pressurization process were studied combined with the principle of back-flow pressurization, so as to lay the theoretical foundation for further optimization of the pump.

2. The basic structure and working principle

The overall structure of DSSP is shown in Fig. 1. The DSSP is mainly composed by screw rotor, two seal discs, pump shell and driving gears (Shen. 2018). The core working part of this pump is the meshing pair composed by the screw rotor and the seal discs. Two seal discs are symmetrically arranged on both sides of the screw. Screw grooves of the DSSP are divided into three parts: entrance section, mesh section and exit section according to the different functions of screw grooves. Along with the screw rotating, the whole meshing area is divided into two parts: the

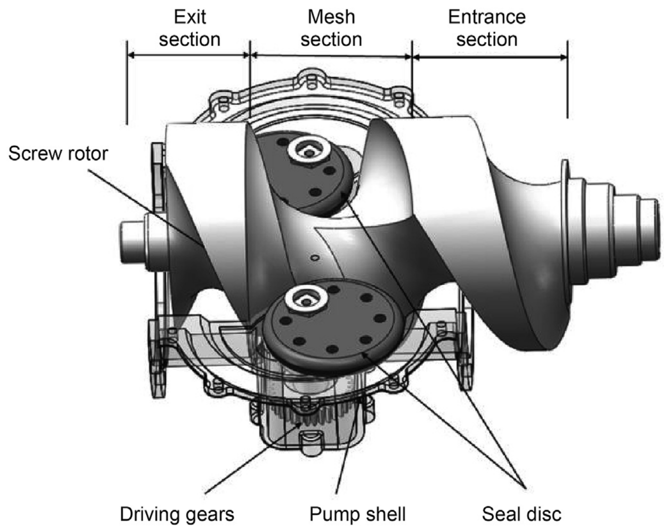

Fig. 1. Overall structure of DSSP.

pressurization section and the return section. When the seal disc is located in the pressurization section, the meshing angle between the seal disc and the screw groove is large enough so that the meshing driving force provided by screw groove can drives the seal disc to rotate. When the seal disc is located in the return section, the meshing angle between the seal disc and the screw groove is so small that the meshing driving force is not enough to drive the seal disc rotate. In order to make the seal disc complete a rotation cycle successfully in the return section, the driving gears is arranged between two seal discs so that the seal disc which in the return section can smoothly complete a rotation cycle to realize the continuous work of the DSSP.

The DSSP is a positive displacement pump, which relies on the screw groove, the seal disc and the pump shell to form a periodically changing volume to realize the pressurized transportation of liquid medium with the principle of back-flow pressurization. When the DSSP starts to work, the volume of the chamber at the inlet section increases gradually, then the negative pressure was produced by the increase of chamber. Under the action of pressure difference between inside and outside the suction inlet, the medium flows into the working chamber from the suction inlet. The suction liquid enters the pressurization chamber under the promotion of the spiral groove at the inlet section and then the seal disc intervenes to form an independent pressurization chamber. The pressurization chamber is opened to connect with the exit at the moment of closing. Pressurization is achieved through the backflow of the outlet side fluid, and the medium is deduced from the outlet section by the push action of the seal disc, forming a continuous liquid medium transportation.

3. Mathematical model for pressurization characteristic

3.1. Principle of reflux pressurization

In order to reduce the friction and wear between the relative moving parts in the actual working process of the DSSP, a fit clearance should be reserved between the moving parts in the pump. Because of the meshing relationship between the seal disc and the pressurization section of the screw rotor, the meshing clearance will also be generated. DSSP also has a unique type of leakage channel, that is, the suction and backflow pressurized leakage channel formed between the seal disc, screw rotor and the casing when the pressurized cavity is not fully closed. This unique leakage channel only exists in the very small rotation angle range before and after the pressurized cavity is completely closed. The existence of these clearance channels leads to the backflow leakage of the liquid medium in the pump cavity, including the backflow leakage from the high-pressure chamber connected with the outlet section to the pressurization chamber (Fig. 2a) and the backflow leakage from the pressurization chamber to the low-pressure chamber connected with the inlet section (Fig. 2b).

The backflow pressurization process of DSSP can be simplified as the control volume as shown in Fig. 3. The backflow leakage of the liquid medium makes the high-pressure medium and the liquid medium in the pressurization chamber to mix with each other, thus the pressurization of the liquid medium inside the working cavity is realized. This is the principle of the backflow pressurization of the DSSP.

3.2. Basic assumptions

The fluid flow in the three-dimensional space channel is complex and changeable, so it is difficult to directly reveal its complex flow law. Therefore, some basic assumptions need to be made before establishing the mathematical model of the pressurization process for the DSSP, as follows:

(1) The liquid medium transported by the DSSP is incompressible liquid, ignoring the resistance loss along the flow process of the liquid medium.
(2) Neglecting the kinetic energy of liquid medium flowing in the pump cavity.
(3) It is considered that the positive pressure and mass force of the liquid medium transported by the DSSP are potential and only subject to gravity, the liquid medium is steady flow.
(4) The change of potential energy in the process of liquid flow is ignored, and it is considered that the pressure state at different positions in the chamber is the same at any time.
(5) It is considered that the pressure in the working chamber which is connected with the inlet and outlet is constant.
(6) The influence of temperature in the working process of liquid medium transported by the DSSP is ignored.

3.3. Establishment of mathematical model

In the mathematical modeling of the pressurized flow process, leakage is the key factor affecting the quality of the medium in the working chamber. The backflow leakage channel of the DSSP includes the leakage channel between the inner wall of the pump shell and the screw rotor, the leakage channel of the meshing clearance between the screw rotor and the seal disc, the leakage channel between the seal disc and the pump shell inner wall and the backflow pressurization leakage channel (It only exists in the case that the seal disc and the pressurized section screw groove are not engaged when the screw rotor rotating angle is located at -5° to $+5^\circ$).

Since the liquid medium is incompressible fluid, the energy equation describing the change of state parameters in the supercharging process is not considered in the mathematical modeling process, only the solution of mass conservation equation, Bernoulli equation and Reynolds equation are considered. Therefore, Bernoulli equation of the fluid flow between adjacent working chambers is established firstly. As shown in Fig. 4, take the liquid medium between the cross sections 1 and 2 of two adjacent working chambers as the research object, the following expression can be obtained:

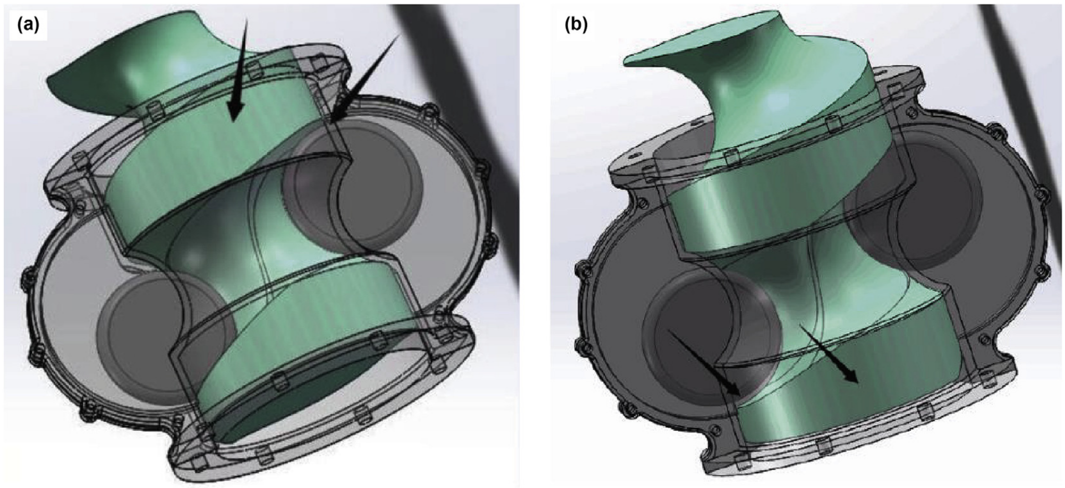

Fig. 2. Backflow leakage channel.

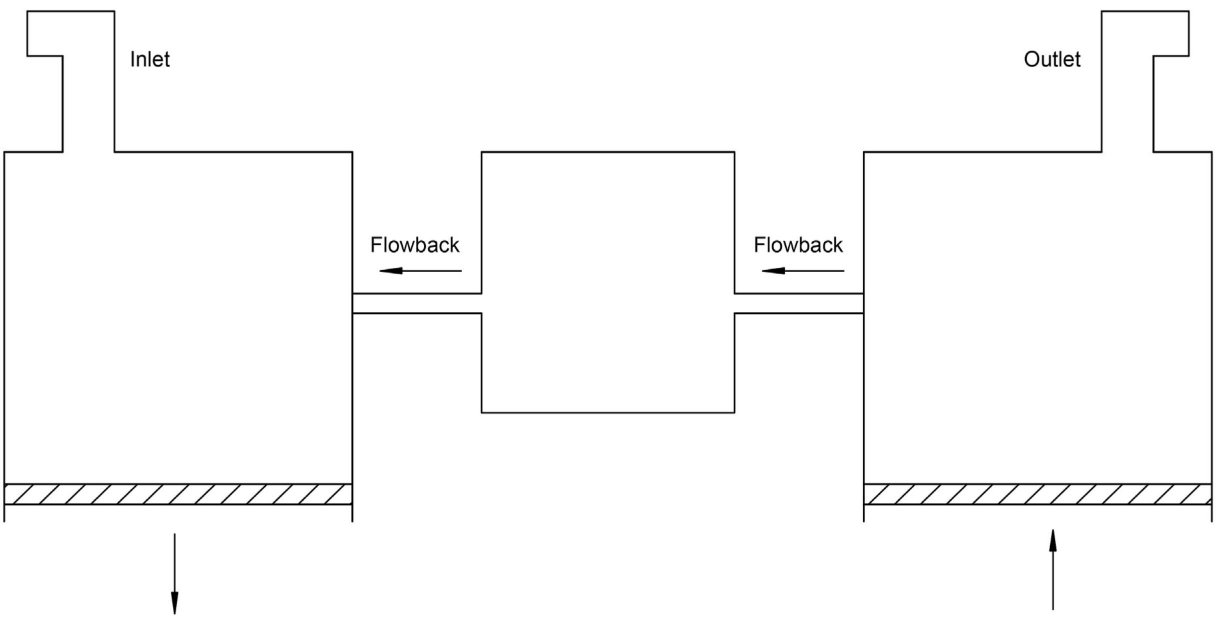

Fig. 3. Schematic diagram of pressure-increasing due to backflow.

$$\frac{p_{\mathrm{d}}}{\rho g}=\frac{p_{\mathrm{k}}}{\rho g}+\frac{u^2}{2g}+h_{\mathrm{w}}+h_{\mathrm{f}} \tag{1}$$

where u is the leakage velocity, P_{d} is the pressure of the working chamber connected with the outlet part, P_{k} is the pressure of the working chamber connected with the inlet part, h_{w} is the local energy loss caused by the expansion of the flow channel, h_{f} is the resistance loss along the liquid flow.

Considering the leakage through the return channel, the expression of local energy loss can be obtained as follows :

$$h_{\mathrm{w}}=\frac{u^2}{2g}\left(1-\frac{S_1}{S_2}\right) \tag{2}$$

where S_1 represent the cross-sectional area near the leakage channel and S_2 represent the cross-sectional area of the outlet part respectively. According to the geometric modeling process of the backflow leakage channel, the area S_1 of the backflow leakage channel is far smaller than the cross-sectional area S_2 of the outlet part, so S_1 can be ignored in the calculation process. Through the above analysis, the expression of local energy loss can be obtained as follows:

$$h_w=\frac{u^2}{2g} \tag{3}$$

In this paper, only the energy loss of the liquid medium at the leakage channel is considered. According to the analysis, the mathematical expression of the resistance loss along the flow path caused by the liquid can be shown as follows:

$$h_{\mathrm{f}}=\lambda\frac{l_1}{d}\frac{u^2}{2g} \tag{4}$$

where λ is resistance coefficient. It is estimated that the Reynolds number of the liquid at the leakage channel can be expressed

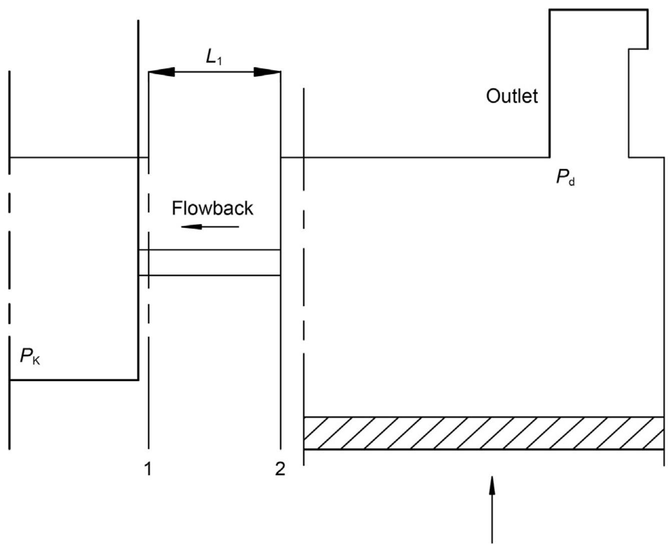

Fig. 4. Model of adjacent working chambers.

as:Re $=\frac{ud}{\nu}\leq 2300$, so the resistance coefficient along the way can be taken as$\lambda=\frac{64}{\mathrm{Re}}$. In the calculation process, the shape of the leakage channel is simplified as a circular ring, d represents the diameter of the circular pipe and l_1 represents the length of the leakage channel.

The leakage model of the meshing clearance leakage channel between the screw rotor and the seal disc can be considered as the flow model of nozzle, by introducing equations (3) and (4) into Bernoulli equation and simplifying, the leakage velocity through this clearance can be obtained as follows:

$$u=\sqrt{\frac{2d(p_{\mathrm{d}}-p_{\mathrm{k}})}{\rho(2d+\lambda l_1)}} \tag{5}$$

Combined with the geometric model of the meshing clearance leakage channel, the theoretical calculation formula of the leakage amount can be obtained as follows:

$$q_1=\begin{cases}2bRu\left(\pi-\arccos\left(\frac{e\cdot\sin\phi}{R}\right)\right),\phi\in(0,\pi)\\2bRu\left(\arccos\left(\frac{-e\cdot\sin\phi}{R}\right)\right),\phi\in(\pi,2\pi)\end{cases} \tag{6}$$

where b is the width of the meshing clearance leakage channel, R represents the radius of sealing disc, e represents the eccentricity of sealing disc, ϕrepresents the screw angle.

When the fluid flows through the backflow pressurized leakage channel, the contact between the fluid and the pump body is line contact, so the leakage channel can be regarded as a thin-walled orifice model. According to Bernoulli equation and fluid continuity equation, the leakage amount of the medium through the backflow pressurized leakage channel can be obtained as follows:

$$q_2=\frac{Y\varphi CA}{\sqrt{1-(d/D)^4}}\sqrt{\frac{2\Delta p}{\rho}} \tag{7}$$

where Y is the expansion coefficient of the internal medium, φ is the expansion coefficient of the material on both sides of the leakage channel, C is the flow coefficient of the leakage orifice model, A is the cross-sectional area of the leakage channel, Δp is the pressure difference between the two sides of the leakage channel, ρ is the fluid density, d/D is the ratio of orifice diameter to the equivalent inner diameter of the leakage flow chamber.

All of the leakage channel between the inner wall of the pump shell and the screw rotor/the seal disc can be equivalent to the parallel plate model as shown in Fig. 5. The distance between the two plates is expressed as h, the width of the parallel plate is b, and the length of the parallel plate is L. The relationship between them is:$b\gg h, L\gg h$. According to the theory of fluid flow between parallel plates in fluid mechanics, the leakage flow velocity can be obtained as follow:

$$u=\frac{1}{2\mu}\frac{\mathrm{d}p}{\mathrm{d}x}(h-y)y\pm U\left(1-\frac{y}{h}\right) \tag{8}$$

where μ is the dynamic viscosity coefficient of the fluid, U is the relative velocity of motion between two plates. The first part is the leakage flow caused by the pressure difference between the two sides of the leakage channel, which is called differential pressure flow. The second part is the leakage flow caused by the relative motion between two plates, which is called shear flow.

By direct integral calculation of the above velocity equation, the leakage amount q_3 through leakage channel between the inner wall of the pump shell and the screw rotor and the leakage amount q_4 through leakage channel between the inner wall of the pump shell and the seal disc can be obtained as follows:

$$q=\frac{bh^3\Delta p}{12\mu L}\pm\frac{bhU}{2} \tag{9}$$

In equation (9), when the rotation direction of the screw rotor relative to the inner wall of the pump shell is consistent with the direction of pressure difference, the sum of leakage quantity of shear flow and differential pressure flow is taken; when the rotation direction of the screw rotor relative to the inner wall of the pump shell is opposite to the direction of pressure difference, the difference of leakage quantity of shear flow and differential pressure flow is taken.

According to the above theoretical analysis results, the leakage amount of each leakage channel flowing into and out of the pressurization chamber can be summed up to obtain the variation of the fluid medium quality in the pressurization chamber per unit time:

$$\Delta m=\rho\cdot[(q_{1\mathrm{in}}+q_{2\mathrm{in}}+q_{3\mathrm{in}}+q_{4\mathrm{in}})-(q_{1\mathrm{out}}+q_{2\mathrm{out}}+q_{3\mathrm{out}}+q_{4\mathrm{out}})] \tag{10}$$

Combining with the calculation formula of the volume of the pressurization chamber of the DSSP (Wang et al., 2020a), the change of the physical volume of the pressurization chamber in unit time can be obtained. Then according to the basic principle of backflow pressurization, the continuity equation can be expressed by taking the pressurization chamber as the control volume as follows:

$$\rho\mathrm{d}V_{\mathrm{p}}=\Delta m \tag{11}$$

where $\mathrm{d}V_{\mathrm{p}}$ is the volume increment of the screw chamber in unit time.

The pressure in the pressurization chamber at any screw corner can be obtained by adding the theoretical geometry model, the leakage model and solving the continuity equation and Bernoulli equation iteratively.

Based on the above analysis, the theoretical drainage volume of DSSP per turn when leakage is considered can be obtained:

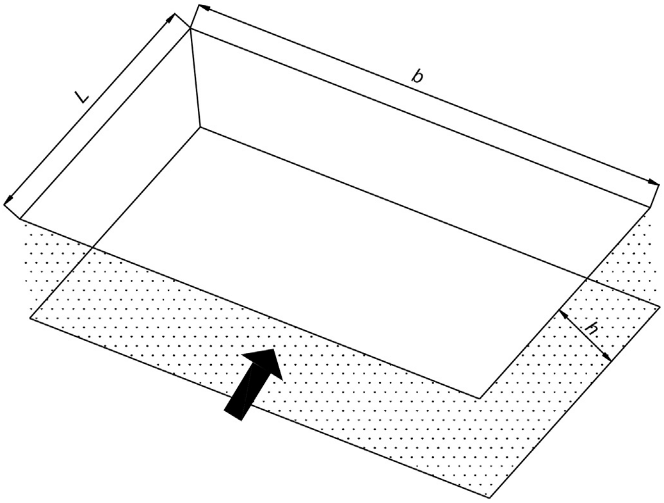

Fig. 5. Parallel plate model.

Table 1
The structural parameters of DSSP.

Screw radius	Center distance	Radius of seal disc	Eccentricity
R_{sr}, mm	a, mm	R, mm	e, mm
100	100	60	30

$$V = V_{\rm p} - \int_0^{2\pi} V_{\rm s} {\rm d}\phi \tag{12}$$

where V_s is the backflow leakage volume from the pressurized chamber to the inlet position in screw unit rotation angle.

4. Results and discussion

In order to solve the pressurization process of DSSP, the structural parameters of DSSP for theoretical research are shown in Table 1, and the working parameters are shown in Table 2. The influence of the working parameters which include the inlet and outlet pressure, screw rotation velocity and the viscosity of the liquid medium on the pressurization characteristics were analyzed based on the mathematical model of pressurization established above.

4.1. Verification of mathematical model of pressurization

In order to verify the correctness of the mathematical model established in this paper , three types of DSSP with the structure parameters shown in Table 3 are used to calculate the theoretical discharge volume when leakage is considered.

The calculation results of the theoretical discharge volume when leakage is considered are listed in Table 3 and are compared with the actual discharge volume of the DSSP used in engineering, the engineering data was obtained from the home page of FOILEX Engineering AB (Shen. 2018).

The pressure of the cavity of the DSSP directly affects its leakage, and it can be seen from the comparison data that the theoretical discharge volume is slightly lower than the actual engineering data when the leakage is considered, and the deviation is not more than 5.6%. This indicates that the mathematical model of the pressurized flow established in this paper is right.

4.2. Analysis of pressurization mechanism

The pressure change law in the pressurization chamber of the DSSP under the standard operating parameters is obtained through theoretical calculation. As shown in Fig. 6, when the screw rotor rotates from the negative angle to the initial position, the pressurization chamber is still connected with the inlet section, so the pressure in the pressurization chamber is basically equal to that at the inlet section. However, during the screw rotation, the high-pressure liquid at the outlet section will flow back to the pressurization chamber through various leakage channels, so the liquid pressure in the pressurization chamber will rise slightly, but not obviously.

With the continuous rotation of the screw rotor, the pressure in the pressurization chamber increases sharply when the screw rotor turns around 0°. This is because a independent chamber will be formed near the rotation angle equal to 0°, and the volume of the independent chamber will gradually decrease, at the same time, the high-pressure medium at the outlet section flow back to the pressurization chamber through the leakage channel. Under the double effects of the pressurization chamber volume reduction and the backflow pressurization, the pressure in the pressurization chamber increases sharply and completes the instantaneous supercharging process. It can be seen that the pressure in the pressurization chamber reaches the maximum value when rotation angle equal to 0° and the instantaneous pressure in the pressurization chamber in this position is even greater than the pressure at the outlet section. This is because the volume of the pressurization chamber before this position continues to decrease and the seal disc continuously squeezes the liquid in the pressurization chamber until the pressurization chamber is basically completely closed at position when rotation angle equal to 0°, and the volume of the pressurization cavity at this position is minimum. Therefore, the liquid in the pressurization chamber is squeezed by the seal disc, and the pressure continues to increase until the pressure value in the pressurization chamber reaches the maximum value when rotation angle equal to 0°.

The connecting area between the pressurization chamber and the spiral groove at the outlet section increases gradually at the process of screw rotate from position when rotation angle equal to 0° to position when rotation angle equal to 5°. At this time, the liquid medium in the pressurization chamber is transported to the outlet section under the promotion of the pressure difference and the seal disc, so that the pressure in the pressurization chamber gradually decreases until it equals to the pressure at the outlet port.

4.3. Influencing factors of pressurization characteristics

In order to explore the influence of the working parameters on

Table 2
The working parameters of DSSP.

Outlet pressure	Inlet pressure	Screw rotation velocity	Dynamic viscosity coefficient
P_d, kPa	P_s, kPa	n, r/s	μ, Pa·s
800	101.315	10	0.1

Table 3
Comparison between theoretical calculation results and engineering data.

Screw diameter	Rotate speed	Actual displacement	Theoretical displacement
mm	rpm	m^3/h	m^3/h
150	750	37.5	35.4
200	625	70	66.45
250	625	140	133.65

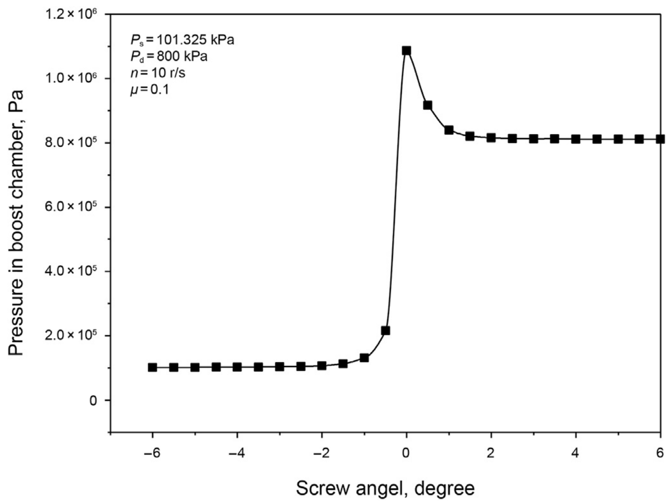

Fig. 6. The law of pressure change in pressurization cavity.

the pressurization characteristics of the DSSP, different working parameters were selected under the condition of keeping other conditions unchanged to analyze the changing law of the pressurization process in the pressurization chamber of the flow-back pressurization area (the rotation angle is between - 5° and +5°).

4.3.1. Effect of screw rotation velocity on pressurization characteristics

When the rotation angle of the screw rotor is between $-5°$ and $+5°$, the influence of the screw rotation velocity on the pressure change law of the pressurization chamber while other parameters are kept unchanged is shown in Fig. 7. It can be seen from Fig. 7a that the pressure change trend in the pressurization chamber is the same under different rotation velocity in the same rotation angle region.

When the rotation angle is in the range of $-5°$ to $-1°$, the pressure in the pressurization chamber increases slightly at different rotation velocities, but the increase trend of the pressure is not obvious, and the pressure in the working chamber has little difference under different rotation velocities.

The pressure in the pressurization chamber increases sharply when the rotating angle is in the range of $-1°$ to $+1°$ and the pressure in the pressurization chamber will increase with the increase of the screw rotation velocity. The reason for this phenomenon is that the high rotation velocity will increase the shear leakage flow in the leakage channel, then the amount of flow-back high pressure liquid will be increased and the pressure in the pressurization chamber also will be increased. In addition, the faster the screw rotation velocity is, the faster the volume of the pressurization chamber decreases near the rotation angle equal to 0°, which leads to the higher pressure in the pressurization chamber. At the same time, it can be seen from Fig. 7b that the maximum pressure that can be achieved in the pressurization chamber increases linearly and proportionally with the increase of the screw rotation velocity. The proportional coefficient between the peak pressure and the screw rotation velocity is 6.29×10^4.

When the rotation angle of the screw rotor is between 0° and $+5°$, the flow area between the pressurization chamber and the outlet section increases rapidly. The liquid in the pressurization chamber is discharged under the push of the pressure difference and the seal disc, so that the pressure in the pressurization chamber decreases gradually. The higher the rotation velocity of the screw is, the faster the high pressure medium in the pressurization chamber is transported to the outlet section, thus the decreasing trend of pressure is more obvious until the pressure in the pressurization chamber drops to the outlet pressure.

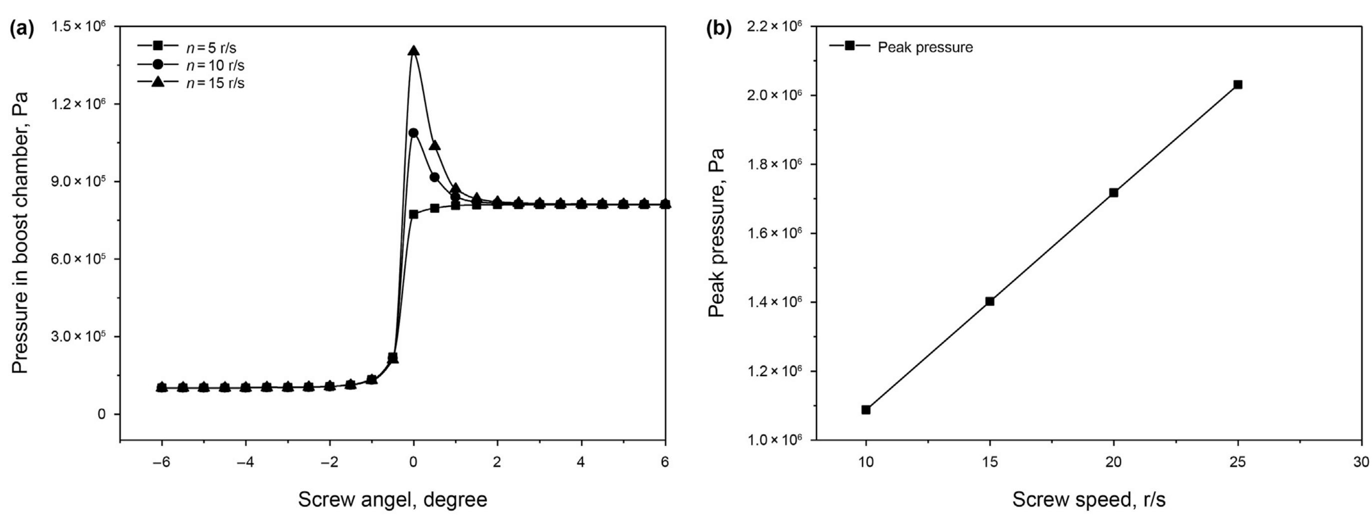

Fig. 7. The influence of screw rotation velocity on pressurization characteristics.

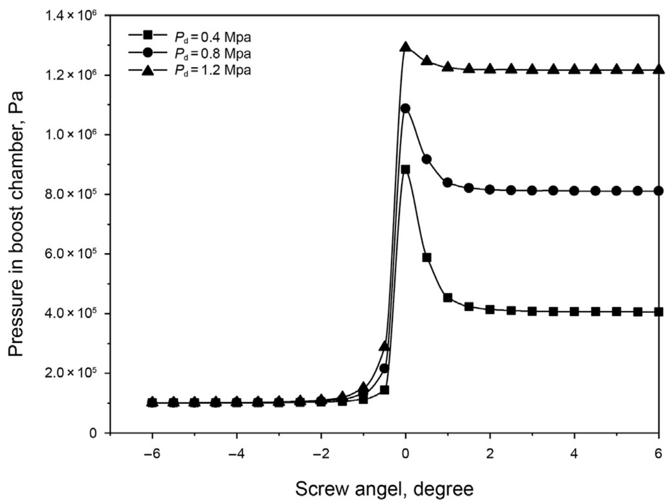

Fig. 8. The influence of pressure at outlet on pressurization characteristics.

4.3.2. *Effect of outlet pressure on pressurization characteristics*

The influence of the outlet pressure on the pressure change law of the pressurization chamber is shown in Fig. 8. It can be seen from Fig. 8 that when the rotation angle is in the range of $-5°$ to $-1°$, the outlet pressure has no obvious effect on the pressure trend in the pressurization chamber. However, when the rotation angle is in the range of $-1°$ to $+1°$, the pressure in the pressurization chamber will increases sharply, and the larger the outlet pressure is, the pressure in the pressurization chamber will increases more obvious. This is because the pressure increment in the pressurization chamber mainly depends on the back-flow pressurization and the volume reduction of the pressurization chamber. The larger the outlet pressure is, the greater the pressure difference between the two ends of the leakage channel is, so the back-flow medium to the pressurization chamber will increase, so the pressure growth trend in the pressurization chamber will be more obvious.

Then the screw rotor continues to rotate to $+5°$, the pressure in the pressurization chamber of different control groups will gradually reduce to the outlet pressure under their respective working conditions.

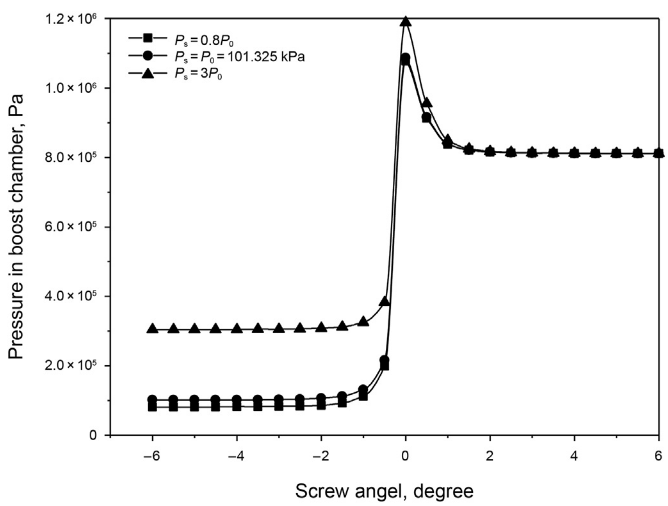

Fig. 9. The influence of pressure at inlet on pressurization characteristics.

4.3.3. *Effect of inlet pressure on pressurization characteristics*

In the flow-back pressurization area, the law of pressure change in the pressurization chamber is shown in Fig. 9 by change the inlet pressure. It can be found that the pressure in the pressurization chamber basically equal to the pressure at the inlet section under their respective working conditions when the rotation angle is in the range of $-5°$ to $-1°$.

In the rotation angle range from $-1°$ to $+1°$, the pressure in pressurization chamber will increase sharply, and with the increase of inlet pressure, the increasing degree of pressure in the pressurization chamber will also increase, but it is not obvious. This is because increasing the inlet pressure will reduce the leakage rate of the pressurization chamber towards the inlet section, thus making the relative volume reduction rate of the pressurization chamber greater, so that the increasing degree of the pressure in the pressurization chamber is greater. Then as the rotation angle continues to rotate to the $+5°$, the pressure of each group in the pressurization chamber will gradually decrease until it is equal to the outlet pressure.

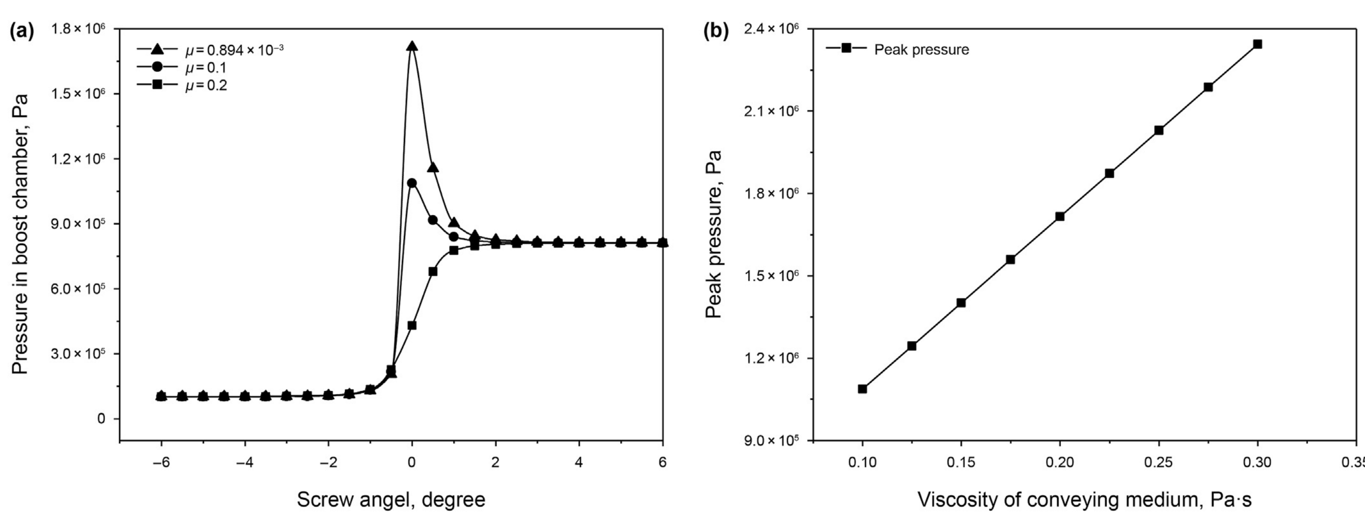

Fig. 10. The influence of viscosity of conveying medium on pressurization characteristics.

4.3.4. Effect of viscosity of conveying medium on pressurization characteristics

The pressure change law in the pressurization chamber of the DSSP under different media conditions when other conditions remain unchanged is shown in Fig. 10. It is can be seen from Fig. 10a that the pressure in the pressurization chamber is not obviously different under different media conditions and is basically equal to the pressure at the inlet when the rotation angle is in the range of -5° to -1°.

When the rotation angle between -1° and $+1^{\circ}$, the pressure in the pressurization chamber will increase sharply, and the larger the viscosity of the medium is, the larger the pressure increase will be. This is because the larger the medium viscosity is, the lower its flowing property is. Thus, when the seal disc rotates to squeeze the liquid in the pressurization chamber, high viscosity media leak slowly, a small volume reduction can cause a large increase in pressure. Then as the screw rotor continues to rotate to the $+5^{\circ}$, the pressure of each group in the pressurization chamber will gradually decrease until it is equal to the outlet pressure.

At the same time, it can be seen from Fig. 10b that the maximum pressure that can be achieved in the pressurization chamber increases linearly and proportionally with the increase of the dynamic viscosity of the conveying medium. The proportional coefficient between the peak pressure and the dynamic viscosity of the conveying medium is 6.28×10^{6}.

5. Conclusions

The pressurization mathematical model of DSSP was established in this paper. The pressurization characteristics of DSSP was analyzed and the influence of various working parameters on the pressurization law of DSSP was explored. The results are obtained as follows.

(1) The pressurization trend of theoretical calculation results is essentially in agreement with the numerical simulation results, which proves that the mathematical model established in this paper can accurately predict the pressurization law in the working chamber.
(2) The instantaneous pressurization process could be realized depending on the reflux pressurization from the outlet chamber to the pressurization chamber and the squeezing effect of the seal disc on the liquid in the pressurization chamber when the screw rotor rotation angle is located at -5° to $+5^{\circ}$. The pressure in the pressurization chamber reaches its maximum value near the position when rotation angle equal to 0°.
(3) The pressure in the pressurization chamber will increase with the increase of working parameters which include inlet pressure, outlet pressure, screw rotation velocity and dynamic viscosity of fluid medium in the area of flow-back pressurization (rotation angle from -5° to $+5^{\circ}$). Comparing the influence of those working parameters on the supercharging process, it can be found that the screw rotation velocity and conveying medium viscosity are the main factors affecting the pressure change of the pressurization chamber.
(4) The screw rotation velocity and the viscosity of the conveying medium have a significant influence on the peak pressure in the pressurization chamber. The peak pressure in the pressurization chamber is proportional to the screw rotation velocity and the dynamic viscosity coefficient of the conveying medium. The proportional coefficient between the peak pressure and the screw rotation velocity is 6.29×10^{4}. The proportional coefficient between the peak pressure and the dynamic viscosity of the conveying medium is 6.28×10^{6}. In practical application, the ultimate bearing capacity of the screw pump should not be neglected while considering the drainage volume and the adaptability to the conveying medium.

According to these obtained results, the pressurized flow process of the disc seal single screw pump has been clearly researched. However, experimental research is a direct means to explore the working performance of the pump. In the future, the pressurization characteristics of the DSSP will be further analyzed through experiments.

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