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STUDY OF THE SUCKING AND DISCHARGING PROCESS OF AXIAL PISTON PUMP WITH DIGITAL DISTRIBUTION MECHANISM UNDER RANDOM LOW SPEED INPUT

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ABSTRACT

A kind of hydraulic piston pump with digital distribution mechanism, which is composed of high-speed switching valves and an absolute rotary encoder to replace the traditional mechanical distribution mechanism, may be used directly in power generation system with a random low speed input. However, traditional structures of hydraulic piston pumps which properly run at about 1500 rpm cannot match this situation. Bad suction, lack of pressure building and low efficiency may occur if only the distribution mechanism is replaced by the digital parts but the rests have no changes. So, a new type of axial piston pump with digital distribution mechanism is put forward in this paper. Firstly, the structure and the working principle of this pump are introduced in detail. Secondly, the factors that influence either sucking process or discharging process are analyzed and some traditional structures of axial piston pump, such as the suction channel and swash plate are challenged in low speed situation. Thirdly, the structure reformations are made and the model of the new axial piston pump with digital distribution mechanism is displayed. Lastly, the new model is simulated in AMESim and Ansys/Fluent. The results show that new structures of axial piston pump with digital distribution mechanism are more adaptable to the random low speed input situation than the traditional ones.

KEYWORDS: axial piston pump, digital distribution, random low input, sucking and discharging, structure reformation

1. INTRODUCTION

With the development of digital fluid power, high-speed switching valve (HSV) is also being studied for the combination with piston pump. In short, the traditional distribution mechanism can be replaced by the HSV and thus both the flow distribution and the flow rate control are operated by the HSV.

HSV has the advantages of excellent switching performance, compact structure, anti-pollution and low cost. Frequency response is a key diameter and it mainly depends on the electromagnetic force and the valve core's displacement. The electromagnetic force is determined by the electric circuit and coil [1]. Larger current and lower inductance will lead a better situation. But restricted by the power consumption and the heat generation, the electromagnetic force is always limited in a certain range. The valve core's displacement, or air gap, is designed for a valve's flow rate. For a certain type of HSV (i.e. 2/2 way), lower flow rate means a higher frequency and vice versa. In general, a valve which usually has a peak flow rate more than 5L/min applied in a piston pump has a frequency less than about 100Hz. Thus there exists some contradiction between the traditional piston pump which properly runs at about 1500rpm (25Hz) and our HSV techniques.

On the other hand, nature power generation system (i.e. wind power & ocean power) has a random low input where hydraulic pump can be used as the energy collection and conversion component [2] [3]. To catch this energy, previous method is that a speed increaser is used before the traditional piston pump and proportional valves is applied to control flow rate after the pump. Though it's a good chance for the digital pump to handle this because HSV has the ability of flow control and prefers to run at a low speed, traditional pump's structure cannot meet this situation easily. Bad suction, lack of pressure building and low efficiency may occur if only the distribution mechanism is replaced by the digital distribution part but the rests have no changes.

Therefore, this paper focuses on the structure change of the traditional axial piston pump. Factors that influence either sucking process or discharging process are analyzed in detail in low speed situation. The structure reformations are made and the model of the new axial piston pump as an example with digital distribution mechanism is displayed. The characteristics of the new digital pump which runs at low speed are also studied by the simulations in AMESim and Ansys/Fluent.

2. REVIEW OF DIGITAL DISTRIBUTION

In this paper, the structure of digital distribution pump is based on a 5-piston axial piston pump. Its principle diagram is shown in Figure 1. When the input shaft is driven by the outside torque, the swash plate rotates and the piston is in a reciprocating motion. Meanwhile, the rotation position is recorded by the absolute encoder and the rotation speed is calculated by the controller. Then the controller decides each 3/2 HSVs' performance, which separately controls each piston chamber. In the sucking process, the HSV is de-energized in the right position and the piston chamber is connected to the tank. In the discharging process, the HSV is energized in duty cycle and switches between the left position and the right position adapted to the variation of rotation speed. By these methods, a pump can work adapted to the random input and output a relative stable flux without other mechanics and valves. This model doesn't have the distribution mechanism, which is the difference between the digital distribution and the plate distribution.

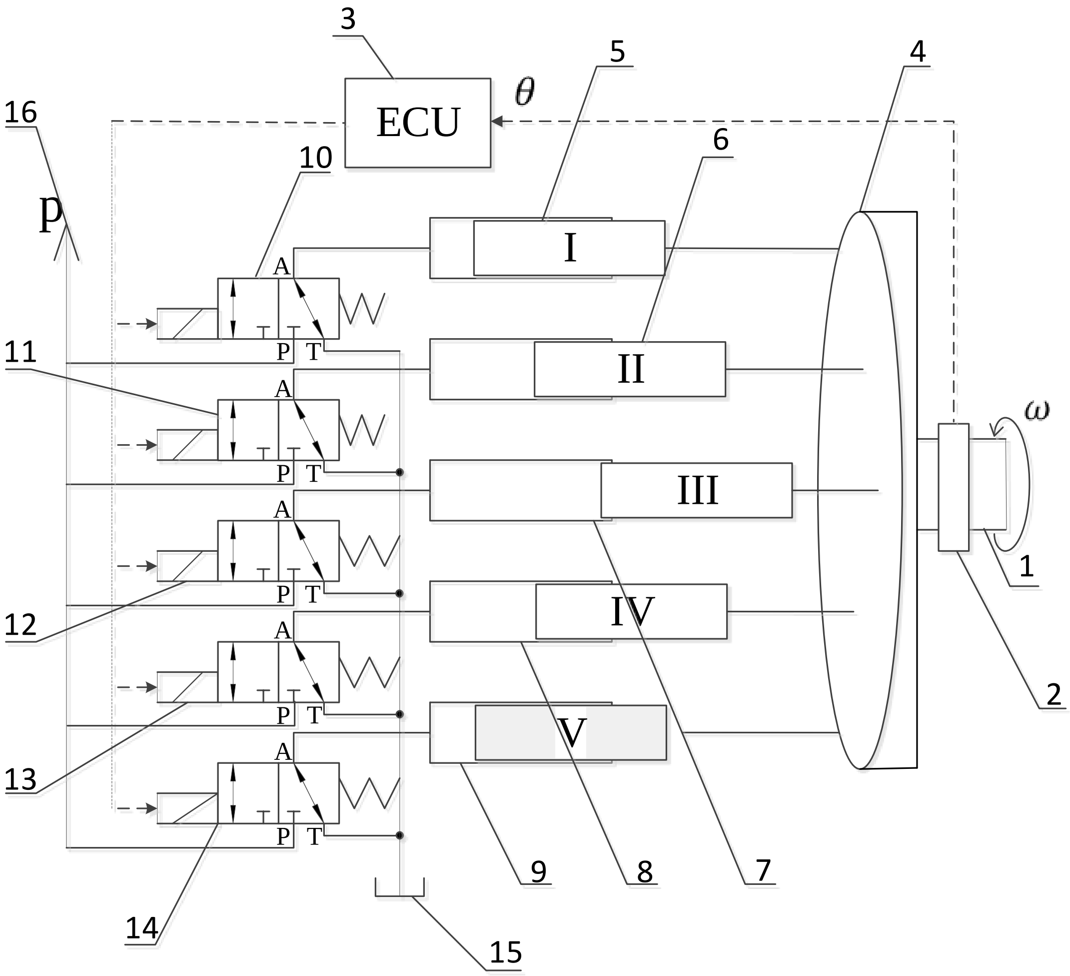


Figure 1. Schematic of the axial piston pump with digital distribution
(1.Input shaft, 2.Absolute encoder, 3.Controller, 4.Swash plate, 5~9.Piston and piston chamber, 10~14.3/2 HSV, 15.Tank, 16.Load)

3. FACTORS INFLUENCE SUCKING OR DISCHARGING PROCESS

When the pump is driven by the random low input, the HSV is easy to control but the pump cannot run properly. The reason is that the traditional structure which runs at high speed cannot meet this situation. Bad suction, lack of pressure building and low efficiency occur if only the distribution mechanism is replaced by the digital distribution part but the rests have no changes. In this part, some main factors will be discussed in detail.

3.1. Self-suction ability

In the process of sucking oil, the volume enlargement of the piston chamber leads to a negative pressure in it. Then the oil fills in the chamber by the pushing of atmosphere. To ensure a smooth and stable sucking, the atmosphere pressure not only transfers to the velocity of fluid, but also conquers the fluid resistance along the flow path and ensures whole pressure is over than the saturated vapour pressure to avoid cavitation erosion.

In whole pump's view, the flow rate only has small ripples under the rated speed. So Bernoulli equation is well to evaluate the pressure in the whole flow field if the oil is considered as ideal fluid,

$$\frac{p_0}{\rho}-\frac{p_{\mathrm{loss}}}{\rho}=\frac{p}{\rho}+\frac{1}{2}v^2+g\Delta h \tag{1}$$

In equation (1), p_{loss} includes the viscosity of the wall, bending of the pipe, and other components' losses in the flow line. In a valve-distribution pump, for example, the check valve's pressure loss on the suction line is unneglectable. Δh is the height difference between the target level and the tank level. For simplification, Δh is thought to be zero in this paper.

In a single piston's view, the flow motion of it is quite unsteady. Equation (2) is used to illustrate the unsteady relationship between two arbitrary points A and B on the flow line and ds is differential length along flow line. The velocity of the end point on piston is shown in equation (3), where ω is the rotation speed of the pump, R is the piston's generic radius, β is the swash-plate's dip angle, and φ is the swash plate's angular position and it only has the relation with time. And through continuity equation (4), the velocity of every point on the flow line is relevant to v_p .

$$\int_A^B \frac{\mathrm{d}p}{\rho}+\frac{{v_{\mathrm{B}}}^2-{v_{\mathrm{A}}}^2}{2}+\int_A^B \frac{\partial v_{\mathrm{s}}}{\partial t}\mathrm{d}s=0 \tag{2}$$

$$v_{\mathrm{p}}=\omega R\tan\beta\sin\varphi \tag{3}$$

$$v_{\mathrm{p}}S_{\mathrm{p}}=v_{\mathrm{A}}S_{\mathrm{A}}=v_{\mathrm{B}}S_{\mathrm{B}}=v_{\mathrm{s}}S_{\mathrm{s}} \tag{4}$$

Combing equations (2)(3)(4), and equation (5) can be get.

$$p_{\mathrm{B}}=p_{\mathrm{A}}-\frac{{v_{\mathrm{p}}}^2}{2}\left(\frac{1}{{S_{\mathrm{B}}}^2}-\frac{1}{{S_{\mathrm{A}}}^2}\right)-S_{\mathrm{p}}R\tan\beta(\omega'\sin\varphi+\omega^2\cos\varphi)\int_A^B\frac{1}{S_{\mathrm{s}}}\mathrm{d}s \tag{5}$$

Equation (5) shows that p_{B} is not only decided by its section area S_{B}, but also affected by the flow line's length L_{AB}. So with the aim of good self-suction, the flow pipe is designed to be wide and short.

3.2. Leakage

For an axial piston pump, leakage occurs at barrel port plate, swash plate, slipper bearing or between piston and barrel. Leakage benefits the relative motion with less friction and heat generation, but sometimes leads to lack of pressure building and low volumetric efficiency. Many researchers studied the leakage in the piston pump with different approaches. Foster and Hannan integrated the dynamic pressure differential equation of the cylinder and evaluated the leakage experimentally [5]. Manring assumed all leakage flows as laminar and used a linear relation between pressure drop and flow, and found the leakage constant for every pump's clearance [6]. Ivantysynova integrated the Reynolds equation of lubrication and the implicit solution was performed through a numerical computer program called CASPAR [7]. Bergada gave a complete analysis of the leakage and output flow ripples considering tilt angle [8]. It is demonstrated in common that over 94% of the leakage in piston pumps is in the slipper-swash plate and the barrel port plate. Since the barrel port plate in our digital distribution pump is replaced by the HSV control, the leakage of slipper-swash plate is the main problem.

The leakage flow rate q_l through a uniform gap height h_l between the shoe land and the swash plate is given by equation (7) [9], where b is the pressure ratio of the recess cavity to the piston chamber, μ is oil viscosity, R_r is the diameter ratio of the recess cavity to the shoe, p_s is the pressure in piston chamber.

$$q_l = \frac{\pi b}{6\mu\left|\ln(R_r)\right|} p_s h_l^3 \tag{7}$$

Leakages of two pumps, which are designed as having different displacement and same geometry characters, are compared in table 1.

Table 1 Leakage comparison of two pumps in similar geometry characters

Swash-plate dip angle β			16.2°		
Recess ratio R_r			0.7		
Pressure ratio b			0.92		
Displacement (mL/r)	Film height(μm)	Leakage flow rate (L/min)	Proportion (1000rpm)	Proportion (100rpm)	Proportion (20rpm)
80	5	0.0069	0.02%	0.21%	1.07%
	10	0.055	0.17%	1.72%	8.60%
	15	0.186	0.58%	5.80%	29.02%
200	10	0.055	0.07%	0.69%	3.44%
	15	0.186	0.23%	2.32%	11.61%
	20	0.44	0.55%	5.5%	27.51%

In the high speed condition (over 1000rpm), the leakage takes up a small proportion in both pumps whether the film height is 5 μm or 20 μm.

In the low speed condition (20rpm~100rpm), the same leakage accounts for more proportion. The volumetric efficiency is poor especially in the lower displacement pump. So the film height needs to be controlled about 5 μm in the 80mL/r pump, while 10 μm in the 200mL/r pump is feasible. So the full lubrication is only possible in the large displacement pump and mixed lubrication is a general solution in the low speed condition.

4. STRUCTURE REFORMATION

The structure diagram of the new 5-piston axial piston pump with digital distribution mechanism is shown in Figure 2. It is designed to work at 20rpm~100rpm, 25MPa, and to have a displacement of 200mL/r.

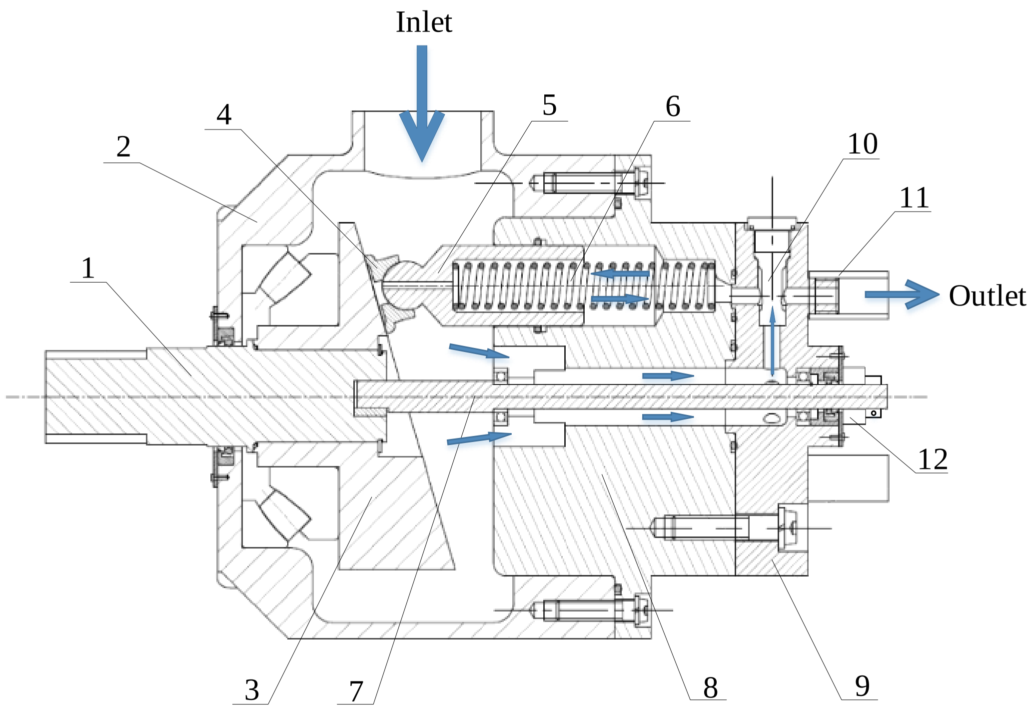


Figure 2. Structure diagram of the new axial piston pump with digital distribution (1.Input shaft, 2.Low-pressure body, 3.Swash plate, 4.Swash shoe, 5.Piston, 6.Spring, 7.Secondary shaft, 8.Barrel, 9.HSV body, 10.2/2 way HSV, 11.Check valve, 12.Absolute encoder.)

4.1. Sucking process

Firstly, the new model changes the topology of suction channel that the oil in the piston chamber is not sucked directly from the outer tank but from the cavity of low-pressure body, where the swash plate, swash shoe and the piston's head work in it. The increasing volume of the piston chamber in the sucking process equals to the decreasing volume of the low-pressure body's cavity for a single piston. Thus the oil is considered to be stored in the low-pressure cavity in discharging process before it is sucked into the piston chamber, which benefits to the self-suction. Though this advantage is weakened if multi- pistons are considered, it offsets the peak flow rate of suction indeed.

Secondly, the 3/2 way HSV is separated to two valves and a 2/2 way HSV is on the suction way. Compared to the check valve used in other valve distribution pistons, HSV driven by the electrical signal has less pressure difference. And 2/2 way HSV also has a quicker response compared to 3/2 HSV.

Lastly, with the whole pump's size as small as possible, the suction channel is designed to be short and wide according to equation (5).

4.2. Discharging process

Since our pump has a displacement of 200mL/r, both the full lubrication and the mixed lubrication are used and compared in the experimental prototype. For full lubrication, b>0.9 is recommended and the height of the film is controlled. For mixed lubrication, friction heat generation and the mechanical energy loss cause concerns. So R_r is designed larger than that in full lubrication to have a bigger fluid support force. The shoe geometry changes from the original one in Figure 3. The friction power loss of each is designed less than 50W in the prototype.

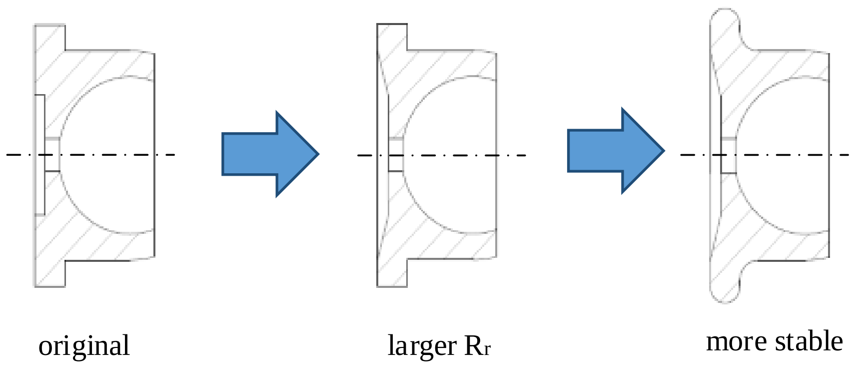


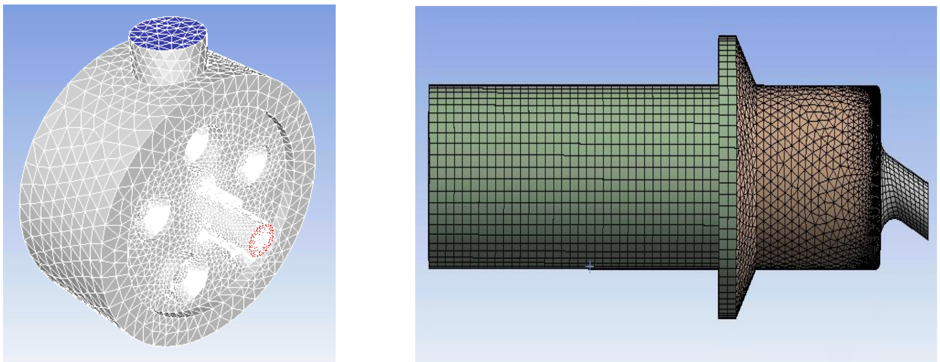
Figure 3. Shoe changes from original one

5. SIMULATION

5.1. Suction simulation

The sucking process is simulated in Ansys/Fluent to find the pressure drop in the pump. The rated flow rate of the prototype pump is 4L/min. The velocity of each piston is calculated according to equation (3) and $\omega = 20\text{r/min}$, $\beta = 16.2^\circ$, $R = 40\text{mm}$.

The whole suction flow can be divided into three parts in simulation. The first part is the public flow field includes the low pressure cavity and the region around secondary shaft shown in Figure 4 (a). It is a relative stable and can be solved by steady state method. The second part is the flow field through the HSV, its flow characters can be find in the product features and is not mentioned in this paper. The third part is each piston chamber and it is simulated in dynamic mesh.



(a) Steady state region (b) Dynamic boundary region

Figure 4. Mesh model of the suction channel

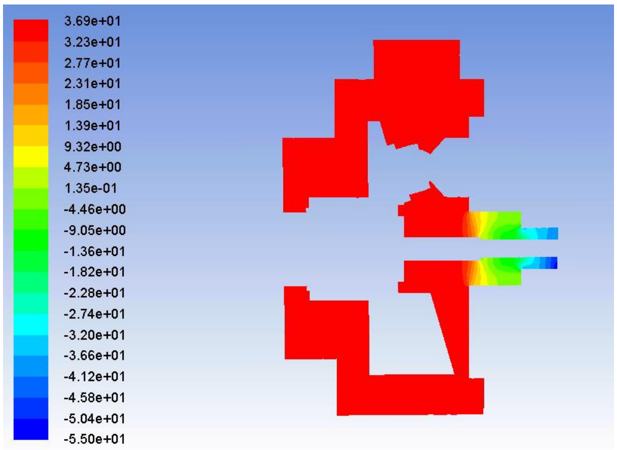


Figure 5. Pressure distribution in public region

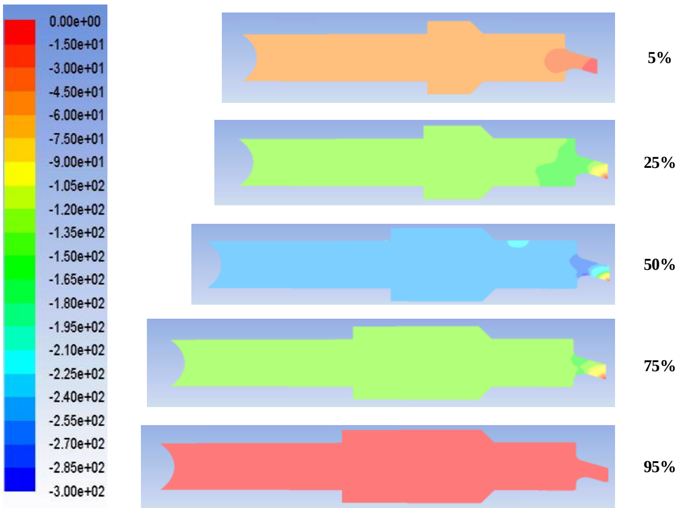


Figure 6. Pressure distributions in piston chambers at different time

Results shown in Figure 5 indicate the pressure drops is small (less than 100Pa) and the drop most occurs around the secondary shaft. Results shown in Figure 6 indicate that the maximum pressure drop (275Pa) occurs around the nozzle's exit in half time. And the pressure loss in first half period is bigger than in second half because the piston accelerates in first half period. In general, the prototype pump in low velocity has no suction problem in the public region and the piston chamber.

5.2. Simulation of full lubrication film

The simplified model of a fully lubricated swash film is built in AMESim shown in Figure 7 and its main parameters are the same with the prototype.

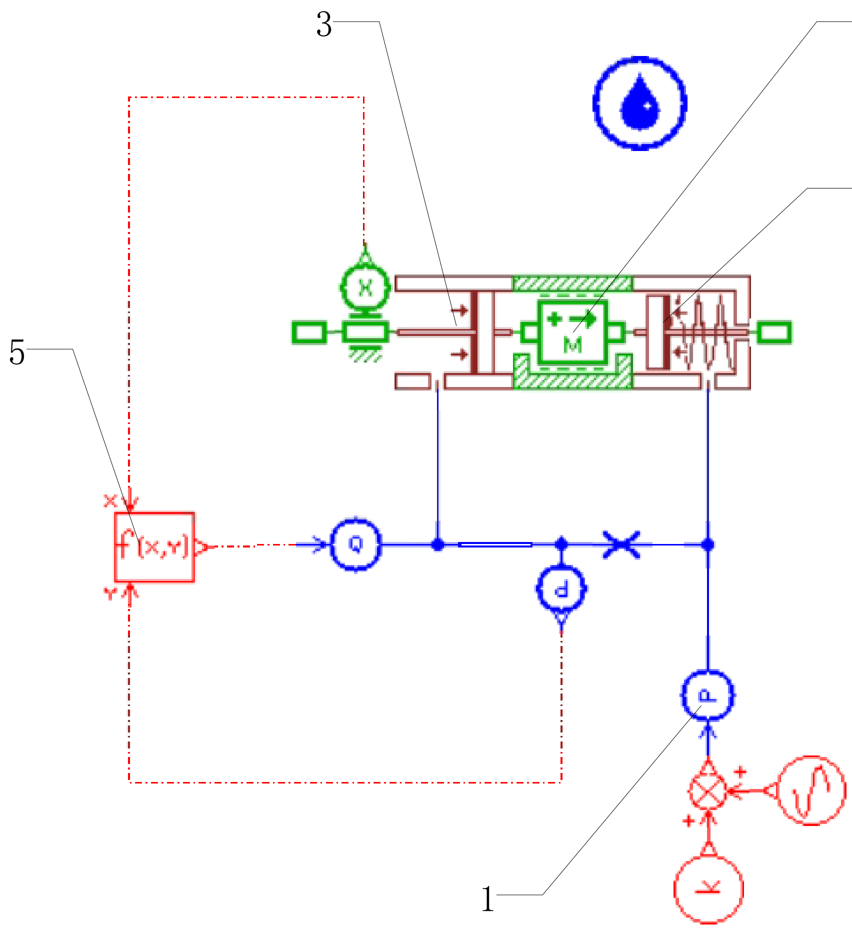
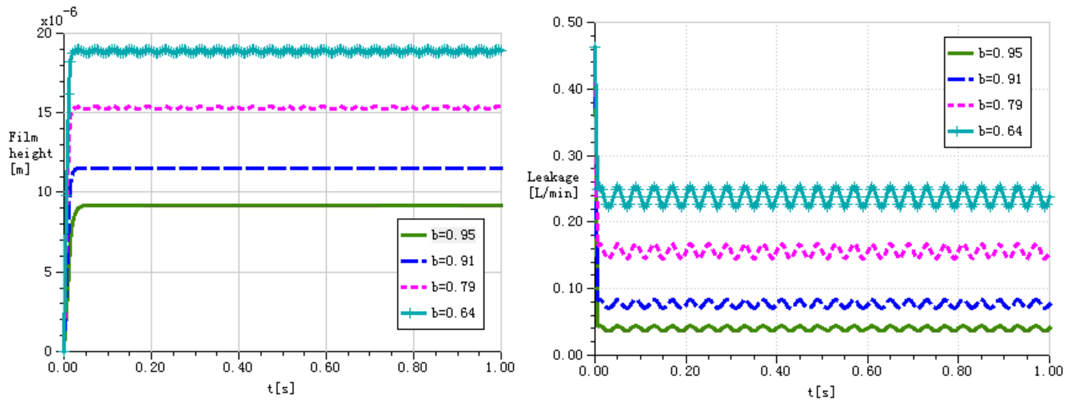


Figure 7. Simplified model of a fully lubricated swash film

(1.Pressure in piston with vibration, 2.Piston model with spring, 3.Equavelent recess cavity model, 4.Mass model with viscous friction, 5.Leakage model)

The film properties shown in Figure 8 indicate that less pressure ratio *b* will thicken the film height and cause more leakage. For the prototype run at 20r/min and having a displacement of 200mL/r, *b* is recommend to be over 0.9 and film height is controlled below 10 μm.

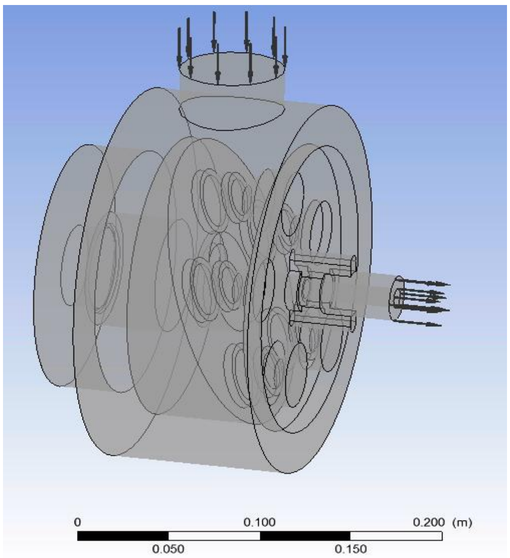


(a) Film height (b) Leakage flowrate

Figure 8. Film properties in different pressure ratio *b*

5.3. Simulation of heat transfer

The heat transfer simulation is done in Workbench/CFX. The simulation domain and its settings are shown in Figure 9. Steady state for the heat analysis is used. Heat transfer is only considered in the flow field. And buoyancy is not included in this model.

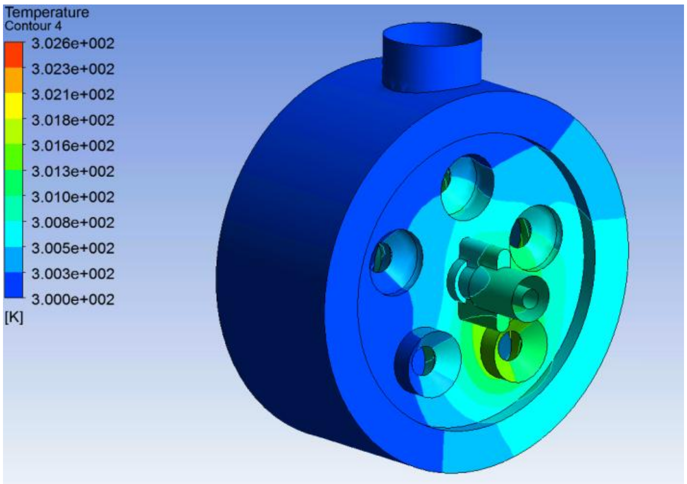


Density	850 [kg m^-3]
☑ Specific Heat Capacity	
Option	Value
Specific Heat Capacity	2000 [J kg^-1 K^-1]
Specific Heat Type	Constant Pressure
☐ Reference State	
Transport Properties	
☑ Dynamic Viscosity	
Option	Value
Dynamic Viscosity	0.0368 [Pa s]
☑ Thermal Conductivity	
Option	Value
Thermal Conductivity	0.12 [W m^-1 K^-1]

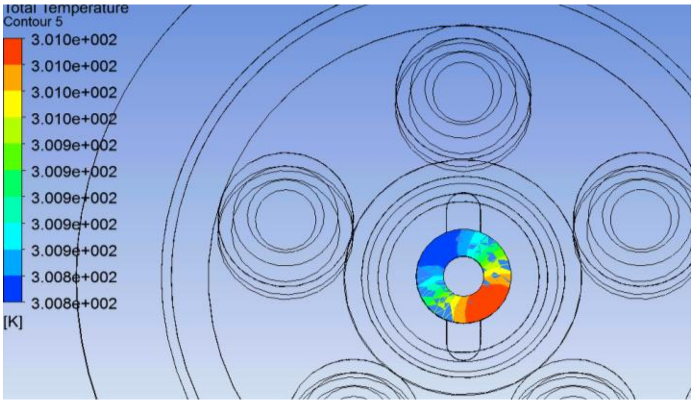
Figure 9. Heat transfer simulation domain and its settings

At one moment, three pistons are assumed in the discharging process with heat generation 50W each. The inlet has a flow rate of 4L/min and the temperature is set to be 300K. The temperature result in Figure 10 shows that the highest temperature occurred around the pressurized shoes is 302.6K, only having an increase of 2.6K. The temperature raise of the outlet is about 1K.

In general, with the suction of cold oil from the tank, this channel has a high cooling effect. The swash shoe, the swash plate and any other bearings in the low-pressure cavity can be protected from temperature rise.



(a) Whole fluid domain



(b) Outlet surface

Figure 10. Temperature field

6. CONCLUSION

Traditional pump's structure has some problems in the random low speed condition if only the distribution mechanism is replaced by HSVs but others have no change. Factors which influence either sucking or discharging process are discussed in this paper and structure reformations are made to overcome these problems.

The sucking channel is changed and optimized and the suction becomes easier. These changes benefit on lubrication and cooling as well. Also 3/2 way HSV is separated into a 2/2 HSV on sucking channel and a check valve on discharging channel, which will cause less pressure loss on valve.

As the main source of leakage, the swash shoe characteristics are analyzed and simulated. Results show that the full lubrication film with pressure ratio b might larger than 0.9, and the mixed lubrication with large recess ratio R_r is recommended for a large displacement pump. But for a small displacement pump, only the mixed lubrication works.

7. ACKNOWLEDGEMENT

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