

Performance prediction and experimental study of variable displacement vane oil pump

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Abstract: In order to fully predict the performance of variable displacement vane oil pump (VDVP) during concept design phase, one-dimensional simulation software AMESim is used to model the centralised parameters of VDVP. Several important performance indexes of VDVP are predicted, including the flow, pressure, torque and initial variable point, it is proved by the bench test that the simulation results have a good correlation with the test results. In addition, the effect of leakage on volumetric efficiency is studied, external leakage is smaller than internal leakage, and the higher the temperature, the larger the leakage. The force characteristics of VDVP are studied. Under extreme conditions, the maximum pressure of the internal chamber is about 15.9 bar, and the maximum forces of the stator and rotor are 1920 and 1653 N, respectively. The effect of temperature on the frictional power is studied. Under low-temperature conditions, the viscous frictional power is much greater than the dry frictional power, while the high temperature is the opposite.

1 Introduction

The oil pump is the core component of the engine lubrication system, and its function is to improve the oil to the corresponding pressure and transport it to the moving parts to reduce the wear of the parts [1]. As the engine moves towards higher speeds and more energy saving, traditional metering pumps are difficult to match engines with multiple operating conditions. Due to the multi-stage variable, low-pressure pulsation and low noise, the variable displacement vane oil pump (VDVP) has become the most widely used oil pump on passenger car engines [2]. In recent years, the replacement of passenger car products is frequent, and how to comprehensively predict the performance of VDVP is particularly important.

For the analysis of oil pump performance, there are mainly theoretical analysis and simulation. Truong *et al.* [3] introduced a mathematical modelling method based on geometric design and dynamics analysis of VDVP and studied the performance characteristics of the ideal pump. Jayanthamani *et al.* [4] established a mathematical model of VDVP, calculated the flow, leakage and friction under different operating conditions, and verified that the mathematical model can accurately predict the flow within a certain speed. Jenkins and Ivantysynova [5] established a semi-empirical lumped parameter model for the vane pump and analysed its dynamic stability to the system.

Jiang and Perng [6] and Ding *et al.* [7] proposed a computational fluid dynamics (CFD) model for oil pump calculations, later, taking the axial flow water pump as an example, an advanced cavitation model was proposed to simulate the volumetric pump. Bai *et al.* [8] used Pumplinx to analyse the flow, pressure pulsation and cavitation of VDVP, and verified them through experiments. Ham *et al.* [9] studied the relief groove to reduce the pressure pulsation and cavitation of the gerotor oil pump. Rundo *et al.* [10] studied the incomplete filling of the internal chamber at high speed, which is the main cause of volumetric efficiency.

Since the variable mechanism of VDVP changes with the operating conditions of the engine, the fluid software such as Pumplinx cannot simulate the dynamic characteristics of VDVP. Wang *et al.* [11] used the kinetic equation to calculate the eccentricity of the variable mechanism under different operating conditions and then input it into Pumplinx for performance analysis of the fixed speed. The reliability of the method was proved by

experiments. However, this method also cannot directly obtain the dynamic characteristics of the variable mechanism, and it takes time and effort.

One-dimensional (1D) simulation can effectively solve the above problems and can realise electromechanical fluid co-simulation in a short time, which is widely used in performance analysis of the external gear pump, gerotor pump and axial pump [12–16]. Based on AMESim software, this paper studies the performance characteristics of VDVP and applied in practical products.

2 Principle of VDVP

Fig. 1 shows the structural diagram of VDVP studied in this paper. Seven vanes form seven internal chambers, the driving shaft drives the chambers to rotate clockwise, the oil is sucked from the inlet port and discharged from the outlet port. When the rotor rotates one cycle, each chamber completes oil absorption and oil discharge once. When the oil pump is variable, the main gallery oil enters the control chamber from the pilot valve, so that the stator overcomes the preload of the variable spring and rotates anticlockwise around the pivot, thereby reducing the eccentricity and achieving the purpose of the variable.

Fig. 2 shows the overall diagram of the variable control. The solenoid valve and the pilot valve together control the variable of the oil pump. When the engine is in low-speed or low-load condition, the controller has no signal output, the solenoid valve works in the left position, and the oil pump is in the low-pressure mode. When the pressure of the main gallery causes the pilot valve to move to the left, the oil pump starts with variable, and the pressure at this time is called the initial variable point of the low-pressure mode.

When the engine is at high-speed or high-load condition, the controller outputs a signal, the solenoid valve works in the right position, and the oil pump is in the high-pressure mode. Since there is hydraulic pressure on both sides of the pilot valve, and the area on the right side is larger than the area on the left side, the movement of the valve will be slower than the low-pressure mode, so the pressure at the initial variable point will increase.

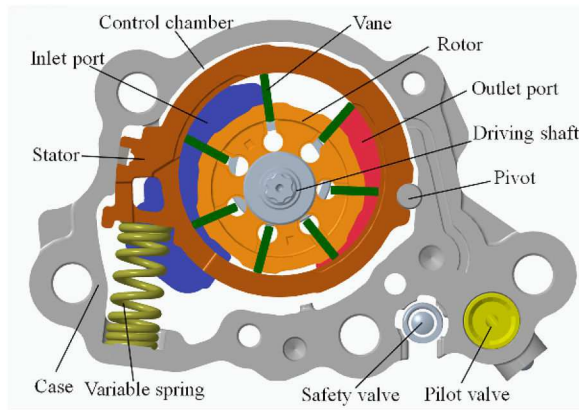


Fig. 1 Assembly identification diagram of VDVP

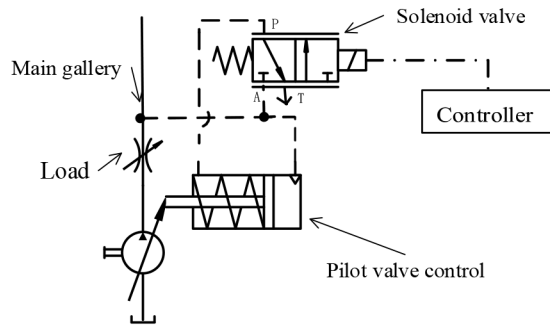


Fig. 2 Overall diagram of the variable control

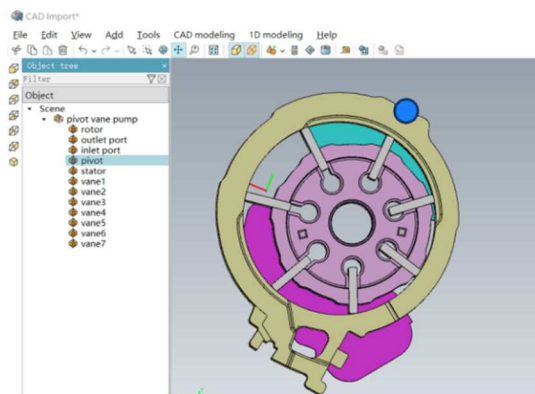


Fig. 3 Partial diagram of 'CAD import' tool

3 Simulation model

According to the structural model of VDVP, selected the rotor, stator, vane, pivot, the inlet port and outlet port, converted them into '.stp' format, and imported into the 'CAD import' tool of AMESim software. Then selected 'pivoting pump' type and associated the corresponding parts in the 3D model. The software will automatically calculate eccentricity, ports and other parameters, as shown in Fig. 3. The tool simplifies the modelling process of the vane pump and more accurately converts the actual model into a vane pump submodel.

The vane pump submodel has a configuration tool that can be used to check the consistency of the parameters, as shown in Fig. 4. Most of the parameters are automatically converted and can be modified, but the relief groove parameter and dead volume value need to be manually filled in. The dead volume is equal to the minimum chamber volume.

The configuration tool also provides a function diagram of groove, ports, chamber volume with rotor angular displacement, which reflects the motion law of the flow field, and its specific value is used for simulation calculation, as shown in Fig. 5.

All the chambers are modelled as capacitive volumes in which the pressure is obtained by integration, from the expression of its

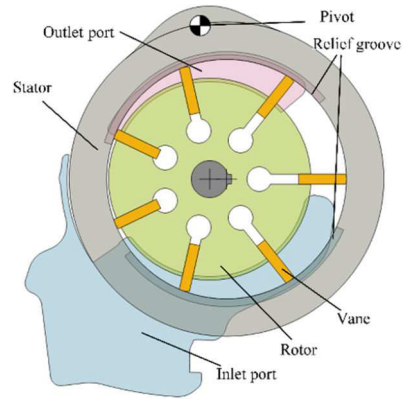


Fig. 4 Visual model of the vane pump submodel

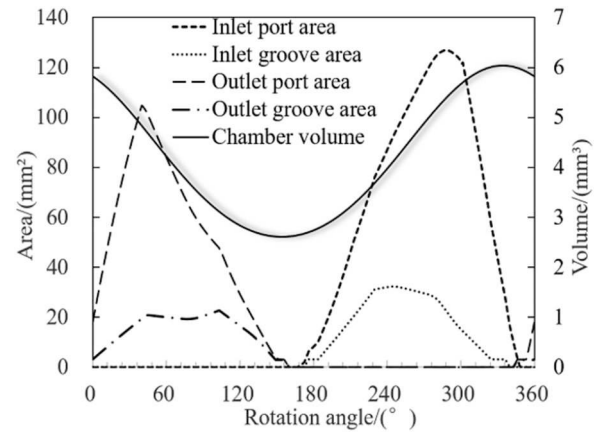


Fig. 5 Grooves, ports and chamber volume versus rotor angular displacement

time derivative. To each volume (i), the mass conservation in isothermal conditions is applied [17]

$$\frac{dp_i}{dt} = \frac{\beta \times ([Q_{in i} - Q_{out i} + Q_{leak in i-1} - Q_{leak in i}] - \sum dvol_i)}{(v_0 + \sum vol_i)} \quad (1)$$

where β is the bulk modulus of the oil, $Q_{in i}$ is the inlet port flow, $Q_{out i}$ is the outlet port flow, v_0 is the dead volume, $\sum vol_i$ is the sum of variable chamber volumes and $dvol_i$ is the time derivative, obtained by the 'vector ray method' [18]. $Q_{leak in i}$ is the internal leakage between chamber (i) and adjacent chambers (these quantities are described in detail in a later section).

The inlet port flow and outlet port flow for each chamber are calculated by an orifice law. The critical flow number (default value 1000) and the maximum flow coefficient (default value 0.7) supplied as parameters are used together with the opening area A to define the volumetric flow rate as

$$Q = C_q \times A \times \sqrt{\frac{2|\Delta P|}{\rho}} \times \frac{\rho}{\rho_0} \times \text{sign}(\Delta P) \quad (2)$$

where C_q is the flow coefficient, A is the sum of the port area and groove area (as shown in Fig. 5), ΔP is the pressure difference, ρ is the oil density and ρ_0 is the density of the oil at standard atmospheric pressure.

Selecting the correlative submodel in the AMESim software, including 'hydraulic', 'hydraulic component design', 'signal, control', 'mechanical', combined with the load control during the bench test, the centralised parameter simulation model as shown in Fig. 6 is established. The main parameter values are shown in Table 1.

In the simulation model, the pump body module is established according to the parameters of the vane, stator, rotor and pivot, which is used to calculate the pressure, flow, force etc. Friction leakage module is established according to the parameters of

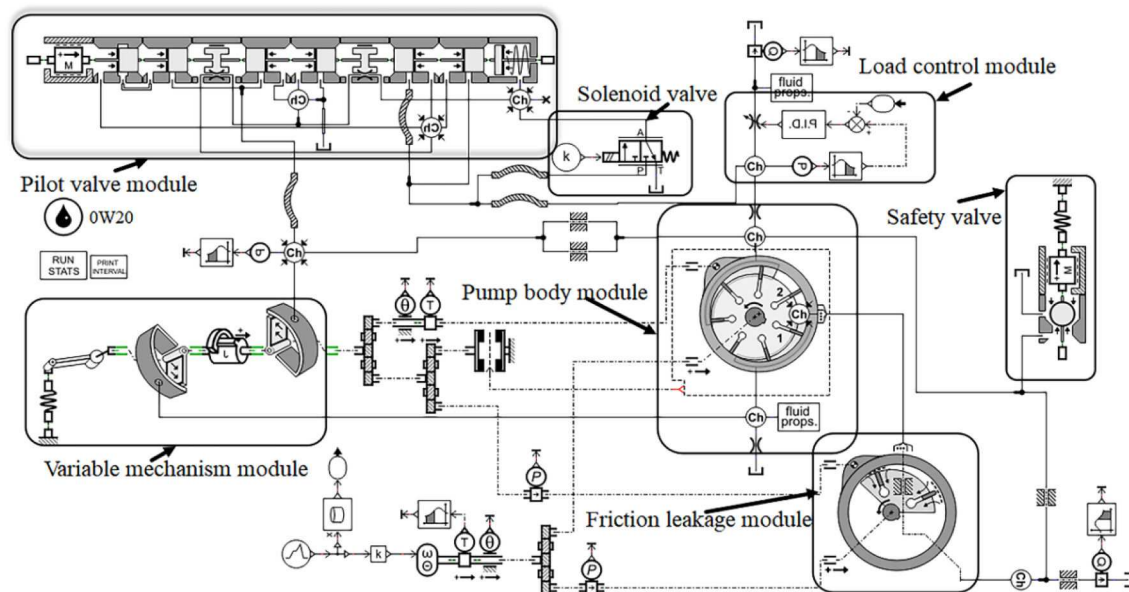


Fig. 6 1D simulation model of VDVP

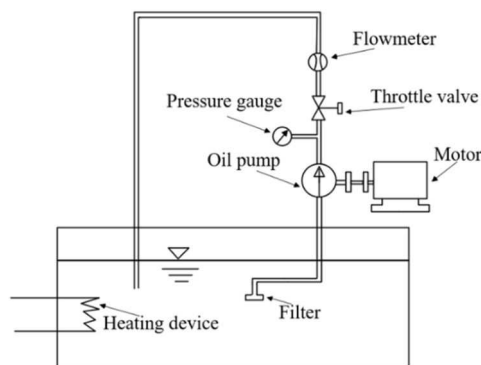


Fig. 7 Principle of the test

leakages, oil film, torques etc., which is used to calculate the friction and leakage between parts. Variable mechanism module is established according to the parameters of the variable spring and stator, which is used to calculate the deflection of the stator. Pilot valve module is established according to the valve structure, clearance and spring parameters, which simulates the actual working process of the pilot valve. The load control module is established according to parameters of the throttle valve of the bench test, which is used to simulate the load condition of the engine. The parameter setting of the switch solenoid valve satisfies the actual working principle of the solenoid valve. Safety valve controls the maximum pressure at the pump outlet.

4 Simulation model verification

4.1 Test principle

In order to verify the reliability of the simulation model for performance prediction, a bench test was carried out. The principle of the test is shown in Fig. 7. The computer can collect pressure, flow, temperature and speed of the motor, and accuracy of the sensor are controlled within 0.5%. The test principle is to adjust the diameter of the throttle valve to change the pressure of the load, so as to simulate different conditions of the engine. The installation layout of the bench test is shown in Fig. 8. The main gallery pipe is connected to the branch pipe of the outlet pipe, and the solenoid valve controls the on and off between it and the oil pump (manual switch to different modes during the test).

4.2 Steady characteristic verification

Steady performance test of the oil pump is to study the relationship between pressure and flow at fixed speed. The diameter of the

Table 1 Main parameter values

Name	Value
oil type	0W20
temperature analysed, °C	40 and 140
speed range, rpm	0–6000
pilot valve preload, N	2.69
variable spring preload, N	131.4
eccentricity range, mm	0.96–3.27

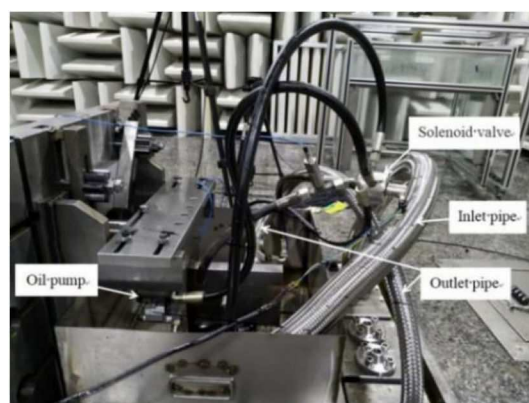


Fig. 8 Installation arrangement of test bench

throttle valve was changed from large to small (maximum diameter of 18 mm) by the computer so that the pump load gradually increased, then the pressure and flow at different speeds were collected. Fig. 9 is a comparison of the simulation results with the test results for VDVP in the low-pressure mode.

It can be seen from the figure that when the pressure reaches 2.1 bar of the initial variable point, the oil pump starts with variable and the flow decreases, when the pressure reaches 10 bar, the safety valve opens, and the flow decreases again. When the oil pump is in high-speed condition, before the variable, the flow of 4000 rpm is greater than the flow of 6000 rpm and the flow after the variable is normal. Owing to the leakage, the curve will not remain level. Simulation and test are in good agreement.

Fig. 10 shows the comparison of the simulation results with the test results in the high-pressure mode. When the pressure reaches the initial variable point of 5.2 bar the oil pump starts with variable and the flow decreases. As in the low-pressure mode, the flow of 4000 rpm is larger than the flow of 6000 rpm, and the flow after the variable is normal. The simulation curve agrees well with the

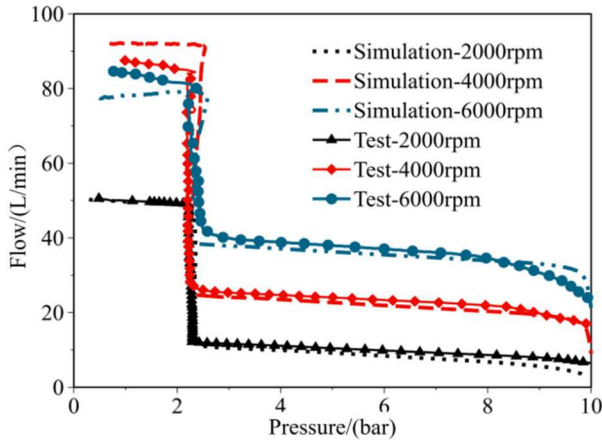


Fig. 9 Pressure flow characteristic curve of the low-pressure mode

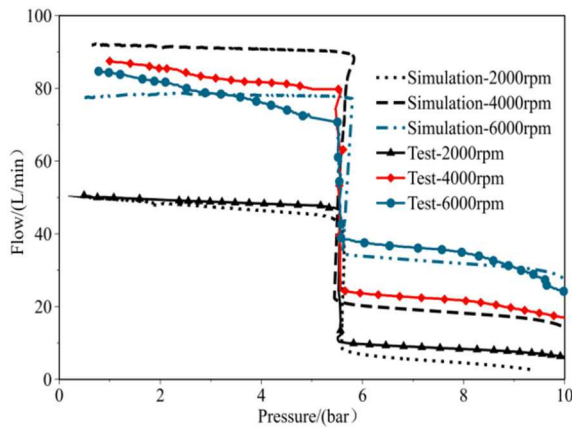


Fig. 10 Pressure flow characteristic curve of the high-pressure mode

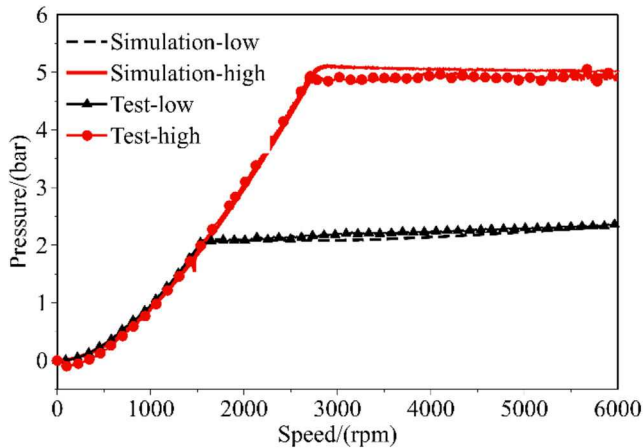


Fig. 11 Acceleration pressure curve

test curve, and the simulation model can reflect the overall characteristics of the oil pump.

Under the condition of large displacement of the oil pump, with the increase of the speed, the oil pump is prone to insufficient oil absorption, resulting in cavitation. When cavitation is severe, the outlet flow will be substantially constant or reduced. After the oil pump variable, the cavitation phenomenon will disappear due to the decrease of flow and the flow curve is normal. In the actual operating conditions of the engine, when the speed reaches 4000 rpm, the oil pump has been variable, so the time of cavitation is short.

4.3 Transient characteristic verification

The transient test is to study the change of pressure during continuous acceleration. The speed of the oil pump was set to

1500 rpm by the computer, and then the diameter of the throttle valve was adjusted to make the pressure of the main gallery reach 1.43 bar, then the diameter of the throttle valve was fixed. Within 60 s, the speed accelerated uniformly from 0 to 6000 rpm and the speed and pressure values were recorded.

Fig. 11 is a comparison of the simulated curve and the test curve in different modes. When the oil pump accelerates in different modes the pressure increases as the speed increases. When the speed reaches about 1500 rpm, the pressure reaches 2.1 bar and the oil pump starts with the variable. As the speed increases, the pressure is basically stable. In the high-pressure mode, the speed reaches 2600 rpm and the pressure reaches ~5.2 bar of the initial variable point. The simulation curve agrees well with the test curve and the simulation model can reflect the acceleration characteristics of the oil pump.

Fig. 12 shows the torque variation curve for the low-pressure mode, which reflects the power consumption of the oil pump during continuous acceleration. At 1500 rpm, the oil pump starts with variable and the torque decreases. When the speed reaches about 3000 rpm, the torque increases again due to the speed, cavitation and friction. The simulation curve can basically reflect the changing rule of torque.

5 Leakage characteristics analysis

Due to the pressure difference in the internal flow field of VDVP, part of the flow will leak from the gap, which reduces the volumetric efficiency of the oil pump.

Fig. 13 shows the gap leakage at different locations. Q_{tip} is the leakage between the stator and the rounded tip of the vane; Q_{side} is the leakage between the casing and the sides of the vanes, Q_{tip} and Q_{side} are called internal leakages of the chamber. Q_{slot} is the leakage of vane slot; Q_{rim} is the leakage between the casing and the sides of the rotor and Q_{slot} and Q_{rim} are called external leakages of the chamber.

Internal leakage between chambers at vane tips and sides are taken into account, those leakages are expressed as a sum of a Poiseuille contribution ($Q_{leak in p}$) and a Couette contribution ($Q_{leak in c}$), with the following description:

$$Q_{leak in} = Q_{leak in p} + Q_{leak in c} \quad (3)$$

$$Q_{leak in p} = \left[H \times cl_{tip}^3 + 2h_{ve i+1} \times cl_{side}^3 \right] \times \frac{P_i - P_{i+1}}{12\mu \times b_v} \quad (4)$$

$$Q_{leak in c} = \frac{\omega}{2} \times \left[H \times cl_{tip} \times \delta_i + ((R_r + h_{ve i+1})^2 - R_r^2) \times cl_{side} \right] \quad (5)$$

where H is the axial width of the vane, cl_{tip} is the vane tip clearance, $h_{ve i}$ is the vane lift, cl_{tip} is the vane side clearance, p_i is the chamber pressure, μ is the dynamic viscosity, b_v is the vane thickness, ω is the speed of the shaft, R_r is the radius of the rotor and δ_i is the length of the vector ray for the vane [18].

External leakage is the sum of Q_{slot} and Q_{rim} , those leakages are expressed from the following Poiseuille equations, for each chamber (i):

$$Q_{leak ext} = 2 \cdot Q_{slot} + Q_{rim} \quad (6)$$

$$Q_{rim} = \left[L_{clh} \times cl_{rim}^3 \right] \cdot \frac{P_1 - P_i}{12\mu \times \log(d_r/(d_r - 2l_{rim}))} \quad (7)$$

$$Q_{slot1} = \left[H \times cl_{slot}^3 \right] \cdot \frac{P_1 - P_i}{12\mu \times (h_v - h_{vane i})} \quad (8)$$

$$Q_{slot2} = \left[H \times cl_{slot}^3 \right] \cdot \frac{P_1 - P_i}{12\mu \times (h_v - h_{vane i+1})} \quad (9)$$

where L_{clh} is the angular section of the chamber, cl_{rim} is the clearance between rotor rim and casing, p_1 is the leakage outlet

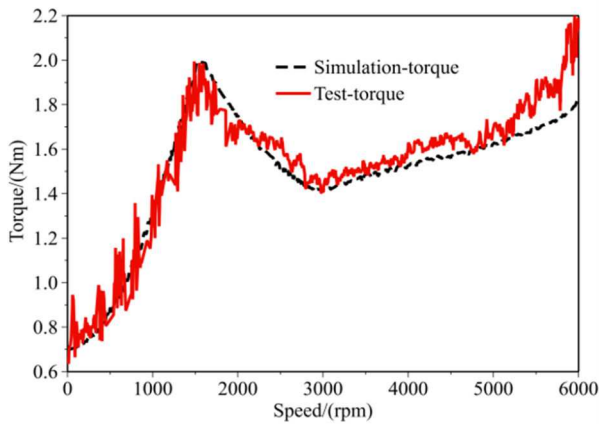


Fig. 12 Acceleration torque curve

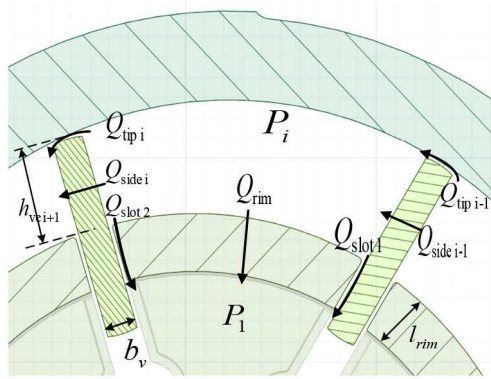


Fig. 13 Gap leakage

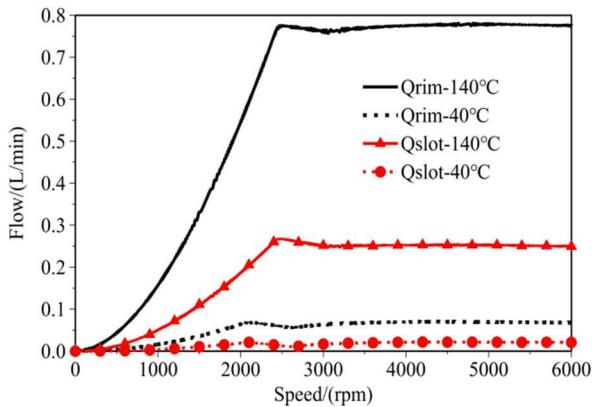


Fig. 14 External leakage

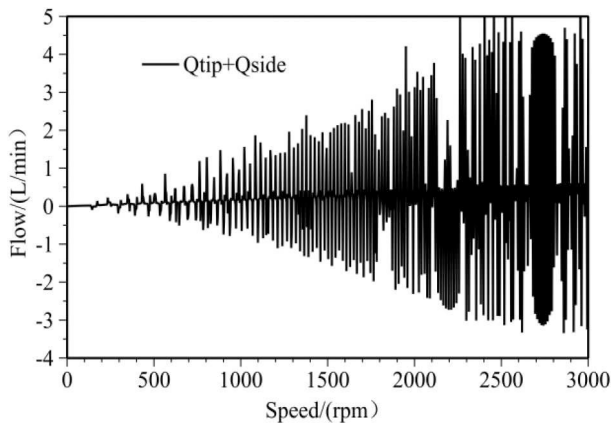


Fig. 15 Internal leakage

pressure; d_r is the diameter of the rotor, l_{rim} is the leakage length of the rotor rim and cl_{slot} is the vane slot clearance.

It is assumed that the gap size does not change with factors such as temperature and load condition. The gap value is measured according to the oil pump, taking the maximum value of the gap. cl_{tip} is 0.063 mm, cl_{side} is 0.084 mm, cl_{rim} is 0.089 mm and cl_{slot} is 0.05 mm.

Fig. 14 shows the relationship of external leakage with the speed at different temperatures. When the speed reaches 2600 rpm, the leakage does not increase because the pressure is stabilised. At $140^{\circ}C$, Q_{side} is 0.8 l/min, accounting for 1.58% of the total flow (total flow is 50.5 l/min), while Q_{slot} is 0.26 l/min, accounting for 0.5% of the total flow, both account for 2.08% of the total flow. At $40^{\circ}C$, those leakages are <0.1 l/min due to the decrease in viscosity of the oil, which is negligible.

Fig. 15 shows an internal leakage curve of a chamber at $140^{\circ}C$, which is selected from 0 to 3000 rpm due to the intensive curve fluctuations. The maximum leakage is stable at 5 and -3.9 l/min (positive and negative is due to the pressure difference between adjacent chambers), the total leakage is about 1.1 l/min, so the leakage of seven chambers is 7.7 l/min, accounting for 15.25% of the total flow.

Since the relief groove will increase internal leakage, and the vane tip is rounded, the leakage length is short, so internal leakage has a great influence on the volumetric efficiency, and the external leakage has little effect. So, reducing internal leakage can effectively improve the volumetric efficiency.

6 Force characteristics analysis

During the working process of the oil pump, the oil generates centrifugal force and hydraulic pressure. Centrifugal force acts on the stator and hydraulic pressure acts on the stator and rotor, as shown in Fig. 16. Those forces will disturb the behaviour of the pump in two ways:

- The forces will push on the rotor and if they are too strong they might make early wear of the pump shaft bearing.
- The forces, if improperly balanced with regard to pivot, will disturb the stator position and the regulated pressure.

The force applied to the stator is calculated by the hydraulic pressure F_{sci} and the centrifugal force F_{spi} [19]:

$$F_s = \sum_{i=1}^7 F_{spi} + \sum_{i=1}^7 F_{sci} \quad (10)$$

The hydraulic pressure in each chamber is acting on a surface of influence defined by the contact points of vane (i) and vane ($i+1$) for the axial width of the vane. The equivalent force F_{spi} is applied in the middle of the contact points line and is perpendicular to it. Its magnitude is

$$|F_{spi}| = p_{ni} \times H \times \sqrt{\delta_i^2 + \delta_{i+1}^2} - 2\delta_i \times \delta_{i+1} \times \cos(\Delta\psi) \quad (11)$$

where p_{ni} is the relative pressure, the direction of the effort could be centripetal if the chamber pressure is lower than the atmospheric pressure and $\Delta\psi$ is the angle difference between adjacent contact points.

The centrifugal force F_{sci} is applied at the centre of gravity of each chamber, which is always radial, its magnitude is

$$|F_{sci}| = \rho \times V_i \times \omega^2 \times r_{gi} \quad (12)$$

where V_i is the volume of the chamber and r_{gi} is the distance between the centre of gravity of the volume and the centre of the rotor.

The method of defining the hydraulic pressure acting on the rotor is similar to that of the stator, its magnitude is

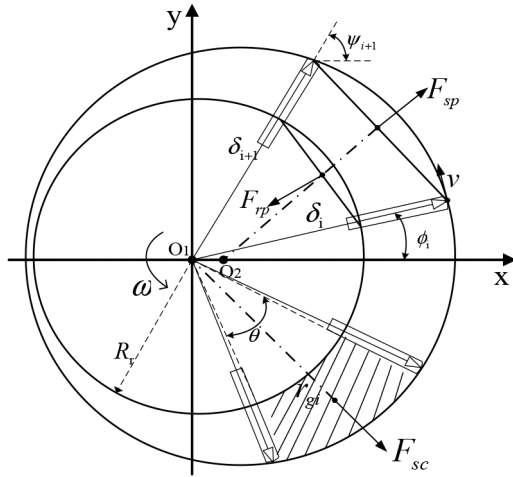


Fig. 16 Force diagram

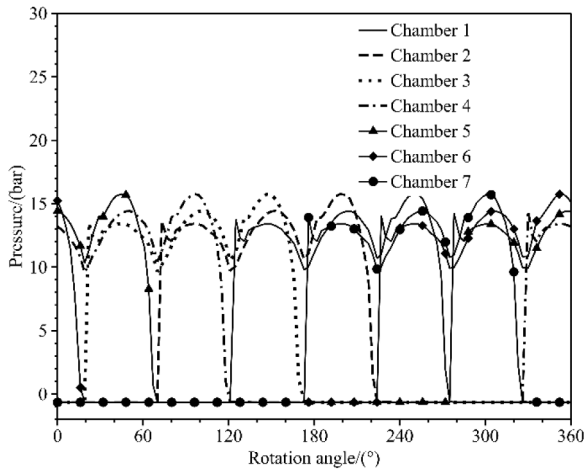


Fig. 17 Chamber pressure curve

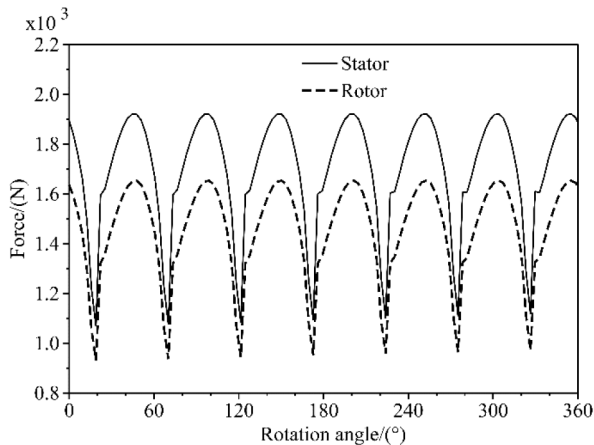


Fig. 18 Stator and rotor force curves

$$|F_{rp}| = p_{ni} \times H \times R_r \times \sqrt{2 - 2\cos(\phi_2 - \phi_1)} \quad (13)$$

where ϕ is the angle between the vane and the x -axis.

Fig. 17 shows the hydraulic pressure curve of the chambers under the extreme conditions (40°C and 6500 rpm). When the chamber absorbs oil, the minimum pressure value is about -0.66 bar, the negative pressure reflects the oil absorption capacity of the oil pump and affects the cavitation of the oil pump. When the chamber discharges oil, the maximum pressure is about 15.9 bar. The maximum pressure has an influence on the strength of the oil pump and cannot exceed the bearing range. Since the outlet port covers three chambers and the outlet port area decreases with the rotation angle, therefore the pressure curve of each chamber has

three peaks, the pressure of the peak gradually increases. The seven-chamber pressures alternate periodically.

The hydraulic pressure acts on the stator and rotor, making the force on the stator and rotor periodic, as shown in Fig. 18. The maximum force of stator is about 1920 N and the minimum force is about 1083 N, the maximum force of rotor is about 1653 N and the minimum force is about 957 N, these forces have changed seven times during one revolution. Since the forced area of the stator is larger than that of the rotor, it can be concluded that the centrifugal force of oil has little influence on the stator.

The radial force applied to the rotor is unbalanced. The unbalanced force increases with increasing pressure and is transmitted to the shaft or bearing, which limits the working pressure and service life of the pump.

7 Frictional power analysis

As mentioned above, the force applied by the stator and the rotor influences the durability of the oil pump. However, the friction causes local cracks and crack growth of the part, which is also a factor in increasing power consumption, so the prediction of friction power is very important.

When the oil pump is running at high speed, the vane tip will stick to the stator ring due to centrifugal force, causing dry friction. Due to the oil film at the gap, there will be viscous friction at the vane tip and the rotor side. Owing to the fact that relative speed between the parts is large, these frictions consume more power.

For the calculation of friction power, the calculation of friction is mainly involved:

$$P_R = F_f \times v \quad (14)$$

where F_f is the friction force applied to the solid and v is the speed of the contact position.

The formula of the dry friction force F_{fb} and the viscous friction force F_{fv} :

$$F_{fb} = m \times P_a \times \text{sign}(dv) \quad (15)$$

$$F_{fv} = \tau \times (A - A_a) \times \text{sign}(dv) \quad (16)$$

where m is the coefficient of friction (default 0.7), P_a is the load carried by the rough surface estimated by the Greenwood-Williamson model [20], dv is the sliding speed or relative tangential velocity between parts, τ is the fluid shear stress; A is the surface of the contact between the vane and stator ring and A_a is the area of asperity contact [19].

The viscous friction of the rotor side is approximated by Newton's law. Based on the double integral with polar coordinates, the following formula is computed for each vane angular sector:

$$F_{trim} = 2 \frac{\mu \cdot \omega_a}{3 \cdot c l_{rim}} \times \theta \times (R_r^3 - (R_r - l_{rim})^3) \quad (17)$$

where θ is the minimum angle between adjacent vanes.

Figs. 19 and 20 show the frictional power curve with speed at different temperatures. It can be concluded that the frictional power increases as the speed increases. At 40°C, because the viscosity is large, the oil film thickness is greater than the average height of the surface roughness, so the viscous frictional power of the vane tip and the viscous frictional power of the rotor side are the most consumed and the dry frictional power is consumed very little. However, when the temperature rises to 140°C the dry frictional power of the vane tip increases and the viscous frictional power decreases due to the decrease of oil film thickness.

8 Conclusion

The purpose of this paper is to demonstrate the ability of 1D simulation to predict the pressure flow, leakage, force and frictional power of VDVP, and draw the following conclusions:

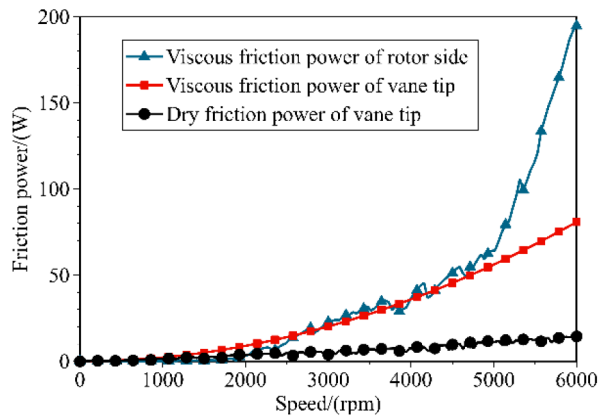


Fig. 19 40°C friction power

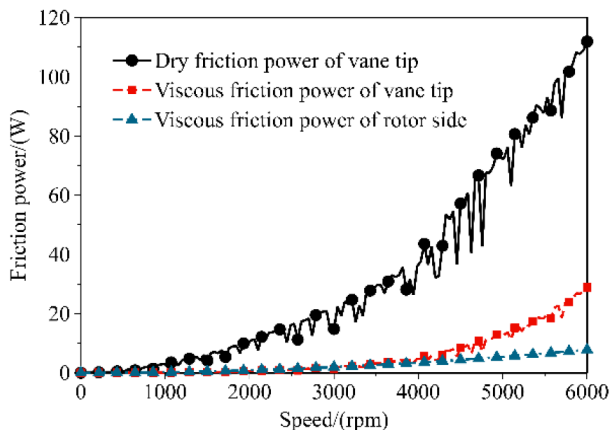


Fig. 20 140°C friction power

(i) 1D simulation model meets the requirements of the bench test and accurately predicts the steady characteristics and transient characteristics of the oil pump.

(ii) When the oil pump is under high speed and large displacement conditions, it is prone to cavitation. When cavitation is severe, the flow no longer increases or decreases, the cavitation disappears after the variable.

(iii) As the speed increases, the leakage of the oil pump increases. External leakage accounts for 2.08% of the total flow, and when the temperature increases, leakage increases. Internal leakage accounts for 15.25% of the total flow, which has the greatest impact on volumetric efficiency.

(iv) When the oil pump is working under extreme conditions, the chamber pressure alternates with the rotation angle and three peaks appear at the outlet port. The chamber pressure causes the force of the rotor and stator to change periodically, the centrifugal force of the oil has less influence on the stator.

(v) Temperature is the most important factor affecting the frictional power. At low temperature, the viscous friction power is much

larger than dry frictional power, and high temperatures are opposite.

9 Acknowledgments

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10 References

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