Numerical calculation and analysis of radial force on the single-action vane pump

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Abstract: Unbalanced radial force is a serious adversity that restricts the working pressure and reduces service life of the single-action vane pump. For revealing and predicting the distribution of radial force on the rotor, a numerical simulation about its transient flow field was performed by using dynamic mesh method with RNG $k-\varepsilon$-turbulent model. The details of transient flow characteristic and pressure fluctuation were obtained, and the radial force and periodic variation can be calculated based on the details. The results show: the radial force has a close relationship with the pressure pulsation; the radial force can be reduced drastically by optimizing the angle of port plate and installing the V-shaped cavity; if the odd number vanes are chosen, it will help reduce the radial force of rotor and optimize the pressure fluctuation effectively.

1. Introduction
Vane pump is a positive displacement pump with superior characteristics and extensive applications. However, the unbalanced radial force which effects on pump’s rotor and aggravates bearings wear, and reduces the service life of vane pump. It restricts the increasing of vane pump’s working pressure and reliability. Therefore, the radial force on vane pump’s rotor must be checked before application. The calculated methods for the pump chamber’s pressure and rotor’s radial force are currently based on the theoretical analysis of geometric models, and the detail analysis and the trend are still not obtained [1-4].

With the development of Computer Fluid Dynamic (CFD), it makes the analysis which is originally with the help of basic theoretical calculation realized as utilizing the CFD on computers. With the help of CFD technology, the internal flow field can be simulated, and the integrated information can be achieved. It has great referential value. Guillaume, Houzeaux et al. [5] introduced a finite element way to simulate the external gear pump, and monitor the velocity and the pressure distribution with the exact prediction of external gear pump’s operating characteristics. In his article, the transient flow field of single vane pump was performed by utilizing dynamic mesh method with RNG $k-\varepsilon$ turbulent model for the first time, and the pressure variation in chamber can be calculated. Further, the radial force’s distribution on rotor can also be analyzed. It provided theoretical basis that helped improve the operating stability of single-action vane pump.

2. Calculated model
A mini single action vane pump is regarded as the calculated model. The radii of pump’s stator are 33.8mm and 31.5mm, and the eccentric distance between the two cycles is 1mm. The number of
blades is 4, and the width of blade is 3.16mm. The revolving speed of pump is 2900r/min, and the rated pressure is 1Mpa. The structure is shown as Figure 1.

![Figure 1. Structure of single-action vane pump](image1)
1.inlet 2.outlet 3.stator 4.blade 5.rotor

The calculated area includes two parts: the oil in pump’s chamber, the oil between port plate and pipe, the gap between blade and stator can be ignored under the low loading pressure [6]; the dip angle of blade is zero. The wall between blade’s top and stator is assumed as surface contact, and the oil’s temperature on pump is assumed as steady state value.

In order to analyze and calculate the rule of node’s movement, the initial coordinate demands of model’s grid are following:
1) The Z axis is defined as rotating shaft, and the positive direction of Z axis is defined as the direction of rotating angular velocity.
2) The center of rotor is located on the origin of coordinate, and the center of stator is located on the point \((-\varepsilon, 0)\). \(\varepsilon\) is referred to the eccentric distance between rotor’s center and stator’s.

The model is built up by the software Pro/E, and is imported into the software ANASYS-ICEM to make mesh generation: The oil in pump’s chamber is meshed by structural grid, and the oil between port plate and pipe is meshed by unstructured grid. As shown in Figure 3, the grid number is 427970, and the grid skew ratio is 0.95.

The purpose of numerical calculation is to obtain the uneven force on pump’s rotor by calculating the pressure in pump’s chamber that varies with time. The pressure and flow rate in outlet, and the force on the surface of rotor and blade are monitored during unsteady calculation.

3. Numerical Calculation Way

3.1 Control equations and boundary condition
The internal flow field is simulated by applying continuity equation and Reynolds average Navier-Stokes equation, and the RNG \(k-\varepsilon\) turbulence model is used to seal off the system. PISO algorithm is chosen as the coupling way of pressure and velocity [7]. The reference pressure is atmospheric pressure. The wall adopts non-slip condition. The transmission medium is normal hydraulic oil. Its kinematic viscosity is 20×10^{-6} m²/s at 20°C.

3.2 Dynamic grid setting
The Clinder model is used to define the revolving velocity and time step. The irrelevant layers and grid which are regenerated during the course of realizing dynamic grid. The node’s movement is the function of eccentric distance and revolving velocity, rotor’s radius. It is controlled by the user define function(UDF) in every time step [8]. The blade revolving 0.1° is defined as a time step.

3.3 User define function algorithm
The oil’s movement can be disposed as a two dimension problem considering the node’s movement is zero in the Z axis direction. The oil that is two dimension is divided into \((n+1)\) parts by \(n\) mini cycles, and every center of the cycle is spread over the segment between the point \((0,0)\) and the point \((-\varepsilon,0)\). Any node’s track can be expressed by the mini cycle’s equation. The radii of stator and rotor can be defined as \(R, r\) respectively.

The node’s revolving angle during the time \(dt\) is

\[
d\theta = \frac{n \times 2\pi}{60} \times dt
\]

(1)

One node’s coordinate is defined as \((x_i, y_i)\) at one time, and the expression of its movement track is following.

\[
(x + \varepsilon_i)^2 + y^2 = r_i^2
\]

(2)

and \(\rho_i = \sqrt{x_i^2 + y_i^2}\)

Figure 4. Relation between radius vector of node and radius of placed circle

Figure 5. Comparison of ave. flow rate in experiment and simulation

Figure 6. Curves of transient flow rate and pressure in outlet

According to the geometric relationship, following expressions can be obtained.

\[
\varepsilon_i = \frac{\rho_i - r}{\rho - r} (\rho_i = \sqrt{x_i^2 + y_i^2})
\]

(3)

\[
\rho = \sqrt{R^2 - (\varepsilon \sin \phi_i)^2} - \varepsilon \cos \phi_i, \phi_i = \arctan \left( \frac{y_i}{x_i} \right)
\]

In this article, the eccentric distance \(\varepsilon = 0.001\)mm can be imported into the expression (3), and the function about eccentric distance can be obtained after connecting with the expression (2)

\[
\varepsilon_i = \Phi_i(x_i, y_i)
\]

The new node’s coordinate \((x'_i, y'_i)\) meets following expressions after the node \((x_i, y_i)\) revolving the angle \(d\theta\).

\[
\begin{cases}
\arctan \left( \frac{y_i}{x_i} \right) - \arctan \left( \frac{y_i}{x_i} \right) = d\theta \\
(x'_i + \varepsilon_i)^2 + y_i^2 = r_i^2
\end{cases}
\]

(4)

4. The analysis of the calculated results

4.1 The prediction of vane pump’s performance
The average flow rate can be tested under different loading pressure according to the rules in JB/T7039-2006, and it is calculated by numerical simulation through 3600 time steps also under different loading pressure. Because of ignoring the gaps, the simulated results’ variation is small in different working condition. The declining point emerges on vane pump’s performance curve in the pump’s performance test. As shown in Figure 5: when the loading pressure ranges from 0 to 0.73Mpa, the pump’s volume loss is lesser. The relative error between calculated result and tested result is lower than 6.6%, and it proves that the results of numerical simulation can predict the performance of flow rate accurately. The relative error will be enormous as the loading pressure exceeding 0.73Mpa. The reason for this phenomenon is that the tested flow rate will reduce rapidly as relief valve generating flow rate leakage.

4.2 The analysis of flow rate an pressure

The uneven variation of vane pump’s flow rate and outlet pressure is primarily caused by the volume variation in pump’s chamber. The blade number is 4 in Figure 6, and this figure show the simulated results that the transient flow rate \( Q_t \) and the outlet pressure \( p \) vary with rotor’s revolving angle \( \phi \). When the blade passes through closed area, the volume in the working chamber between two blades will be compressed. As a result, the internal pressure will increase rapidly. When the high pressure oil is linked with outlet, the high pressure in working chamber will be load-off and the flow rate will increase rapidly at the same time. The high pressure in port plate increasing rapidly leads to the pressure oscillation and the flow rate fluctuation, and it brings a enormous shock to working chamber and blade. Figure 7 is the transient pressure nephogram as the blade revolve 52.0° and 52.5°. As shown in Figure 7, there is a pressure oscillation in outlet.

![Figure 7. Static pressure distribution at the load pressure of 0.25MPa](image)

4.3 The analysis of radial force

The radial force on single action vane pump is consisted of two parts. The hydraulic force acts directly on rotor’s cycle, and the other force working on the blade in seal oil. The force action will transform directly to rotor as the rotor in connection with blades.

Figure 8 shows the variable curve between the radial force \( F \) and direction angle \( \alpha \) during one cycle. The blade number is 4, and the working pressure in outlet is 0.25Mpa in this figure. As shown in Figure 8, the radial force on rotor varies periodically with the revolving angle during one cycle, and the variable periodic is \( 2\pi/4 \). It is consistent with the fluctuation periodic of flow rate and pressure in pump’s outlet. The total radial force mostly ranges from 87.1N to 122.5N, but some moments’ forces exceed 300N, and these moments are as the rotor revolves to the angles 48.3°, 138.3°, 238.8°, 318.3°. When the two blades pass through one closed area near those positions, the volume in the working chamber between two blades will be compressed. The pressure in working chamber increases rapidly; the closed volume between the other two blades increases at the same time, and as a result the pressure reduces. The transient radial force will be enormous in this condition.

5. The measures for reducing radial force
The radial force and periodic variation have a close relationship with the pressure fluctuation, and the transient variation of flow rate can reflect the pressure variation in pump’s chamber. Therefore, some measures such as installing port plate and V-shaped cavity, choosing odd blade number, can be used to reduce the radial force.

5.1 Port plate and V-shaped cavity
In theoretically, when the chamber’s oil is linked with outlet after being compressed, the oil’s pressure is equal to the working pressure. The transient flow rate is the most stable, and the pressure fluctuation is the minimum one in this time. As shown in Figure 9, the pressure peak can be controlled in some degree by the regulation of port plate’s angle $\gamma$ [9]. V-shaped cavity is installed in the inlet window near port plate, and it can be used to increase the flow area. As a result, the pressure in working chamber between two blade can rise gradually, and it will reach to loading pressure at last. With the validation of numerical simulation, when port plate’s angle $\gamma$ is 2.2°, and V-shaped cavity’s wrap angle is 7, the fluctuation of flow rate and pressure in outlet will reduce to the minimum one as shown in Figure 10.

5.2 The selection of blade number
The radial force can be reduced by choosing odd blade number. When the chamber is unsymmetrical, the radial peak force will be smaller during the course of compressing and expanding. As shown in

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**Figure 8.** Radial force distribution at the load pressure of 0.25Mpa  
**Figure 9.** The corner angle of port plate and V-shaped groove  
**Figure 10.** Curves of flow rate and pressure of optimized model  
**Figure 11.** Radial force distribution of optimized model
Figure 12, in the model of 5 blades, some radial force in the compressed chamber will balanced by the other one that is symmetrical with the compressed one as the radial force increase to the maximum one. So the radial peak force will be smaller than the one with four blades. The Figure 13 shows the curve between rotor’s angle and radial force that with 5 blades. The average radial force is 93.66N, and it decreases by 7% than the 4 blades.

6. Conclusions
1) The variable rule of internal transient flow rate and pressure can be represented clearly by the numerical simulation with the UDF and dynamic grid technologies.
2) When the loading pressure ranges from 0 to 0.73Mpa, the relative error for flow rate between numerical simulation and experiment will be lower than 5.6%. The simulated results are credible.
3) The radial force has close relationships with the pressure pulsation; the radial force can be reduced drastically by optimizing the angle of port plate and installing the V-shaped cavity;
4) The radial peak force with five blades will be obviously smaller than the one with four blades, and the average radial force is 93.66N, and it decrease by 7% than the 4 blades. Odd number blade should be chosen.

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