Research on the effect of wear-ring clearances to the axial and radial force of a centrifugal pump

W G Zhao¹, M Y He², C X Qi² and Y B Li²

¹ School of Energy and Power Engineering, Lanzhou University of Technology, Lanzhou, Gansu 730050, China
² Supervision Center of Lanzhou City Wastewater Treatment, Lanzhou, 730050

E-mail: zhaowg@zju.edu.cn

Abstract. Varying of the wear-ring clearance not only has a distinct effect on the volumetric loss of the centrifugal pump, but also on the performance of the centrifugal pump including the axial and radial forces. Comparing with the experimental studies, numerical simulation methods have some special advantages, such as the low cost, fast and high efficiency, and convenient to get the detailed structure of the internal flow characteristics, so it has been widely used in the fluid machinery study in recent years. In order to study the effect of wear-ring clearance on the force performance of the centrifugal pump, based on the Reynolds Time-Averaged N-S equations and RNG k-ε turbulence model, a centrifugal pump with three variable styles of the wear-rings was simulated: Only the clearance of the front wear-ring was changed, only the clearance of the back wear-ring was changed and both were changed. Comparing with the experiment, numerical results show a good agreement. In the three changing styles of the clearance, the variable of the clearance of front wear-ring has the most influence on the axial force of the centrifugal pump, while has tiny effect on the radial force for all the conditions.

1. Introduction

For pump design and operation, the balance of axial and radial force must be considered. Axial force appears due to the unequal pressure distribution on both sides of the impeller, and it directs to the pump suction and parallels to the pump shaft; while the total force in radial direction as a result of uneven pressure distribution is radial force. If these two forces are not to be balanced, collision and abrasion between the static and dynamic components will be caused, and the bearing load increase, resulting in unit vibration, so it is harmful for the normal operation of the pump. So in pump design process, the balance of these two forces must be considered. The main cause of axial and radial force creation is the pressure distribution and the different area on which the pressure acts. Wear-ring seal is the most common form for centrifugal pump impeller seal, and the main role is to limit the leakage of high pressure fluid in impeller chamber to the low pressure area in the inlet or center area of the impeller. When pump operates, the wear-ring clearance will change due to abrasion and other reasons. The change of wear-ring clearance will not only directly affect the impeller sealing effect, but also has a major impact on the overall performance [1-2]. While the leakage of the sealing affects the pressure distribution on both sides of the impeller and the impeller and internal fluid, the force on the impeller will be changed.
Up to now, research on the impact of the ware-ring clearance to pump performance is very little. And research on wear-ring clearance is mainly depends on experimental methods. References [3-5] had a deep study of the wear-ring of a centrifugal oil pump. The clearance size of wear-ring is smaller, and mesh generation and numerical calculation have some difficulties, so numerical simulations of the centrifugal pump often can be simplified, not accounting for the clearance of the wear-ring; as the wear-ring clearance has an important effect on the volumetric loss and also the disk friction losses, the calculation must be corrected using some empirical formulas, thus decreased its credibility. With the development of computational technology, numerical simulation can be carried out in large-scale computation, and numerical simulations with the clearance effect have been widely accepted. Reynolds averaged N-S equations and the RNG k-ε turbulence model are used to simulate a centrifugal pump, in order to have a better understanding of the clearance dimension changes on the axial and radial force of a centrifugal pump.

2. Numerical model
The computational model is an IS80-50-220 centrifugal pump, which has six blades and the specific speed is 59; the design parameters are: the flow is 54m³/h, head 61m, and rotational speed is 2900rpm.

In order to investigate the impact of wear-ring clearance change on the pump performance, three types of calculation model with changed wear-ring clearance were performed: only the clearance of the front wear-ring is changed, only the clearance of the back wear-ring is changed and both are changed. For simplicity, b denotes both the front and back wear-ring clearance are changed; f denotes only the radial dimension of the front wear-ring is changed; h denotes only the radial dimension of the back wear-ring is changed. The original design value of the front and back wear-ring clearance of the pump is 0.15mm. In the calculation, the dimension of the wear-ring on the pump body is retain ed constant, and the clearance is changed by reducing the wear-ring dimension on the impeller to increase the radial clearance value. The wear-ring radial clearance dimension is 0.3, 0.5, and 0.7 separately. For comparison, computational model without considering the wear-ring is simulated, and it is indicated by b0.

![Figure 1. Computational model and mesh.](image)

Hybrid meshing methods of hexahedral and tetrahedral mesh are adopted: for the suction pipe and discharge pipe of the pump, hexahedral mesh is used; while for the wear-ring clearance, because the geometric dimension is relatively smaller, while the ratio of axial to radial size is relatively larger, structured hexahedral mesh is adopted; for the impeller and the volute, unstructured tetrahedral grid is used. Computational model and the grid are shown in Figure 1. The total grid number is 1.2~1.4 million.

3. Numerical method
Based on the non-compressible Reynolds time averaged N-S equations and the RNG k-ε turbulence model, the finite volume method is used for the equation discretization; in which second-order central difference scheme is used for the diffuse term discretization, and second order upwind discretization for the convection term. k-ε model is a semi-empirical turbulence model, and through solution of two transport equations of the turbulence kinetic energy k and turbulence dissipation ε separately to decide
the turbulent velocity and turbulence scale \cite{7}. Separated implicit method is used for the calculation, and SIMPLEC algorithm is adopted for the pressure-velocity coupling.

Velocity boundary condition is used for the inlet, and according to different working conditions, the velocity of the inlet flow is given; the full developmental outflow boundary condition is used for the outlet; the rotating coordinate system is performed on the rotor region and the rotational speed is given; standard wall function method is used near the wall region. When the calculation error is less than $10^{-4}$ or the monitoring static pressure in outlet is almost unchanged, the calculation convergence is considered to be achieved.

| Table 1. Comparison of computational with experimental results. |
|---------------------------------------------------------------|
| Q/Q_0 | $H$(m) | $P$(kw) | $\eta$(%) |
|-------|--------|--------|-----------|
| 0.6   | 64.2   | 65.89  | 65.71     | 10.66 | 10.36 | 9.24 | 52.0 | 54.37 | 61.40 |
| 0.8   | 63.0   | 65.67  | 65.80     | 11.52 | 11.82 | 10.49 | 63.0 | 63.98 | 72.21 |
| 1.0   | 61.0   | 63.97  | 65.32     | 13.30 | 13.87 | 12.47 | 66.0 | 66.39 | 75.38 |
| 1.1   | 60.0   | 63.02  | 65.12     | 14.45 | 14.39 | 69.0  | 67.81 | 76.43 |
| 1.2   | 59.0   | 61.94  | 64.26     | 15.38 | 14.43 | 69.5  | 68.49 | 76.91 |
| 1.3   | 58.0   | 60.41  | 62.83     | 15.51 | 15.32 | 70.0  | 68.72 | 76.76 |
| 1.4   | 57.0   | 58.98  | 61.09     | 16.65 | 16.16 | 69.0  | 68.93 | 76.20 |

4. Results and analysis

For every wear-rings combination, numerical simulation is performed on seven operating points. Through post processing of the calculation results, the external characteristic curve and the internal flow structure was obtained. Table 1 shows the numerical and the experimental results comparison, and there is not efficiency correction for the numerical results. It can be seen from table 1 that the numerical result has a relatively large error compared with the experiment, while the maximum error in designing clearance dimension is less than 5%, within the acceptable range.

![Figure 2. Axial force variation with the flow rate (b0.15)](image)

![Figure 3. Axial force variation with the flow rate when the front and back wear-ring clearance change simultaneously.](image)

4.1. Axial force effect

When the wear-ring clearance is 0.15mm, which is the original design clearance, the axial force variation with the flow rate is shown in Fig 2. As can be seen from the figure, the axial force increases when the flow rate is lower than the design flow rate, while when it is bigger than the design flow rate, it decreases with the flow rate. But the change of the axial force is smaller.

Figure 3 to Figure 5 are the axial force change when the curve respectively the front and back wear-ring clearance changed simultaneously, only the front wear-ring clearance change and only the
back wear-ring clearance change respectively. As can be seen from the figure, when both the front and back wear-ring clearances change at the same time, the impact on the axial force is distinct, and the axial force increases rapidly with the clearance increasing; when only the front wear-ring clearance becomes larger, the axial force variation is larger, but the growth rate is slower than the rings change simultaneously; when only the back wear-ring clearance change, the impact on the axial force is smaller, and variation of the force is not large, moreover when the clearance increases from 0.3mm to 0.7mm, the axial force changes very little.

Figure 4. Axial force variation with the flow rate when the front wear-ring clearance changes.

Figure 5. Axial force variation with the flow rate when the back wear-ring clearance changes.

Figure 6 and Figure 7, are the static pressure distribution in the shroud and hub of the impeller in design condition change with the clearance variation. As can be seen from the figure, the static pressure at the center area increases with the clearance increasing, while the pressure near the impeller rim decreases; and the impact on the static pressure distribution is minimal with only the back wear-ring clearance change.

Thus, the front wear-ring clearance has a greatest impact on the axial force, wherein the axial change when both the front and back wear-ring change mainly depends on the change in the front wear-ring. Therefore, axial force balance by changing the dimension or structure of the wear-ring should be mainly performed by changing the front wear-ring.
4.2. Radial force effect
Calculation of the radial force is more complicated. The radial force is obtained by integrating the pressure on its action area.

Figure 8 to Figure 10 are the radial force change when the curve respectively the front and back wear-ring clearance changed simultaneously, only the front wear-ring clearance change and only the back wear-ring clearance change respectively. As can be seen from the figure, the radial force is small at low flow rate condition and design condition (54m³/h), and the maximum radial force occurs at the large flow rate condition. On the whole, the impact of wear-ring change on the radial force variation is not great, and for the studied pump, the radial force is very small.
5. Conclusions
Numerical simulations were performed in three clearance change programs on a centrifugal pump. Effect of the clearance change of wear-ring on the force performance was analyzed. The following conclusions can be obtained:

1) The wear-ring clearance change has an obvious effect on the axial force. For the three conditions, the effect of the wear-ring clearance on the axial is most significant when both the front and back wear-ring clearance change, while the effect is smallest when only the back wear-ring clearance changes. The front wear-ring clearance change plays a major role in the change of the axial force;

2) The effect of the wear-ring clearance is mainly concentrated in the front and back chamber of the pump, also in the exit of the clearances, while the impact is not significant in other parts of the pump. As the clearance increases, the low pressure area in the chamber expands to the direction of the volute and the high-pressure zone in exit of the clearance has the tendency to expand toward the inlet of the impeller.

3) For the studied pump, the radial force is small; and wear-ring clearance changes have little effect on the radial force.

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