Study on Axial Clearance Size and Leakage of Canned Motor Pump under Axial Force Self-Balance State

Faye Jin¹, Ran Tao¹,*, and Ruofu Xiao¹

¹College of Water Resources and Civil Engineering, China Agricultural University, 17 Qinghua Donglu, Beijing 100083, People’s Republic of China

*Corresponding author email: randytao@cau.edu.cn

Abstract. Canned motor pump is widely used in chemical industry. Due to the particularity of its application, it is necessary to ensure that the medium does not leak completely. If the axial force of impeller is too large, it will directly affect the performance of canned motor pump. Therefore, the floating impeller could be used in the pump to balance the axial force. In this paper, the relationship between axial clearance and leakage rate at the key part of canned motor pump is studied by means of numerical calculation and experimental verification. It is found that the fitting curve is highly consistent with the calculated value, which provided a good theoretical basis for further study of axial clearance control axial force and experimental axial force self-balance. In addition, the leakage rate increases with the increase of axial clearance. The static pressure in the axial clearance first increases and then decreases with the decrease of radius, and the maximum static pressure value is about 10.5% ~ 15.8% near the clearance inlet. This study is of great significance to the theoretical research on the self-balance state of axial force of impeller.

1. Introduction

Canned motor pump is used for conveying special media, which mainly includes flammable, explosive, toxic or highly corrosive liquids. Therefore, it is required to have absolutely no leakage during operation. The pump type connects the pump and the motor together, the rotor of the motor and the impeller of the pump are fixed on the same shaft, and the shielding sleeve is used to separate the rotor and stator of the motor. The rotor operates in the transmitted medium, and its power is transmitted to the rotor through the stator magnetic field. Due to this special requirement, the shield pump should not choose thrust bearing to balance the axial force, so the usual practice is to design it in the form of floating impeller [1-2]. The main factors affecting the axial force of impeller are the gap size of shroud or hub, the shape of rear blade, the size of balance hole, the size of hub seal ring and so on. At present, the main research methods are to change the shape of the impeller, such as setting the rear blade, adjusting the gap size, adding balance hole or balance pipe and so on. Lin et al. [3] found that the clearance size has a certain impact on the axial force and hydraulic performance of centrifugal pump. With the increase of clearance, the efficiency and head decrease slightly. Gantar et al. [4] analyzed the effect of rotating fluid in the gap and radial leakage of wear ring on axial force. Zhao et al. [5] studied the relationship between axial force and hydraulic characteristics by adjusting the radial position of balance hole. The results show that when the radial position of the balance hole decreases, the axial force of the centrifugal pump decreases. At the same time, the head and efficiency are reduced. Kong et al. [6] can effectively realize the axial force balance design by adjusting the size of sealing ring and balance hole. The axial force of the shielded electric pump is measured by the axial force measuring device and the axial force testing system. The principle of impeller axial force self-balance can be described as follows: assuming that the axial force in the opposite direction of incoming flow
increases, the impeller moves to the inlet side. Therefore, the axial clearance at the balance hole will increase and the leakage rate will increase. The high-pressure medium flows into the impeller channel through the balance hole. Therefore, the pressure in the balance chamber will decrease, causing the impeller to move to the outlet side again and again. The impeller is always in automatic adjustment. When the total axial force on the impeller is 0, the impeller can achieve self-balance. Therefore, the leakage of axial clearance is of great significance to study the axial force self-balance of floating impeller.

2. Research Objective and Experimental Verification

The computation domain of pump model is shown in Fig. 1(a). It consists of inflow extension, impeller, volute and outlet extension. The outlet radius $r_2$ at the inlet is 136.0 mm, and the radius $r_1$ at the impeller ring is 62.5 mm. The width of outlet $b_2$ is 19.0 mm. And the main parameters were listed in Table 1. Multiple reference frame (MRF) model is used to simulate the impeller rotating. The rotational speeds $n=2950$ r/min is set to the impeller. Reference pressure of the simulation is set as 101,325 Pa. Boundary conditions includes a mass flow inlet, a static pressure outlet. The advection scheme and turbulence numeric scheme were set as high resolution. As for the discretization schemes, the central difference scheme was adopted for the diffusion term, the second-order upwind scheme was used for the convective term. Hence, 2,239,858 elements in total were selected as a compromise between accuracy and computational cost. The experiment for testing the performance of the centrifugal pump was carried out. There were two pressure sensors were fixed at the inlet and outlet of the tested pump. A flow meter was set to monitor the flow rate of pipeline system. Pressure and flow were transmitted to a computer to record. The numerical results of the pump performance are compared with the experimental results in Fig. 2. In addition, a pressure monitoring line was set in the axial clearance, as shown in the Fig. 1(b).

| Parameters                          | Symbols | Value     |
|------------------------------------|---------|-----------|
| Design Flow Rate                   | $Q_d$   | 200 [m$^3$/h] |
| Design Head                        | $H_d$   | 92 [m]    |
| Design Rotational Speed            | $n$     | 2950 [r/min] |
| Impeller Blade Number              | $z$     | 6 [-]     |
| Impeller Outlet Radius             | $r_2$   | 136.0 [mm] |
| Impeller Inlet Radius              | $r_1$   | 62.5 [mm]  |
| Impeller Outlet Width              | $b_2$   | 19.0 [mm]  |
| Outer Radius of Labyrinth Ring     | $R_{m1}$ | 125.0 [mm] |
| Inner Radius of Labyrinth Ring     | $R_{m2}$ | 115.0 [mm] |
| Diameter of Balance Hole           | $d_h$   | 12.0 [mm]  |
| Specific Speed                     | $n_s$   | 85.4 [-]  |

Table 1. Main parameters of the pump
3. Pre-processing of Axial Clearance
The model of axial clearance is extracted from the entire pump model for detailed analysis. The RANS turbulence models are used to calculate the flow state of the axial clearance. The boundary condition of the axial clearance was obtained from the entire pump model. The inlet is set to a total pressure of approximately 200 kPa, 180 kPa, 160 kPa, 140 kPa, 120 kPa. However, the outlet of the axial clearance was set to the static pressure. Both sides of this domain were set as rotational periodicity.
The wall near impeller was set rotating wall, the rotational speed of which was 2950 r/min. while the other side wall was stationary.

Figure 3. The axial clearance domain and boundary conditions

4. CFD Results
Figure 4 shows the relationship of the axial clearance $A$ and the leakage flow rate $q$. It was found that the $q$ increases with the increasing of $A$. And $q$ would be larger when the pressure at inlet was higher in the same $A$ calculated domain. An allometric model is fitted to indicate their relationship as following:

$$q(A) = mA^b$$  \(\text{(1)}\)

where $m$ is the constant; $b$ is the index number. The iterative algorithm of this expression adopts Levenberg-Marquardt optimization algorithm, and the number of iterations is 5. The R-square value under all working conditions is greater than 0.9950. The standard error of $m$ is under 1%, and the standard error of $b$ is under 8%. The value of $b$ is larger than 1. The statistical results of the fitting expression were listed in Table 2.

Figure 4. The fitting curves between axial clearance and leakage rate

| Pressure Difference | $m$ Value | $m$ Standard Error | $b$ Value | $b$ Standard Error | R-square | RSS |
|---------------------|-----------|--------------------|-----------|--------------------|----------|-----|
| 120 kPa             | 0.236     | 0.012              | 1.558     | 0.081              | 0.9955   | 0.9944 |
| 140 kPa             | 0.296     | 0.009              | 1.366     | 0.046              | 0.9978   | 0.9972 |
| 160 kPa             | 0.333     | 0.008              | 1.309     | 0.035              | 0.9985   | 0.9981 |
| 180 kPa             | 0.373     | 0.007              | 1.239     | 0.031              | 0.9987   | 0.9983 |
| 200 kPa             | 0.403     | 0.006              | 1.212     | 0.025              | 0.9990   | 0.9988 |
Figure 5 shows the $P$ distribution on coordinate of $Y$. When $A$ is 0.2 mm and 0.3 mm, the highest $P$ is around $Y = 54.4$ mm which is near the clearance inlet. The maximum value of $P$ is about 150 kPa. The position of the maximum $P$ is about 10.5% of the total clearance length. When $A$ is 0.5 mm, 0.7 mm, 1.0 mm and 2.0 mm, the highest $P$ is around $Y = 52.6$ mm. The maximum value of $P$ is about 160 kPa. The position of the maximum $P$ is about 15.8% of the total clearance length.

![Figure 5. The static pressure distribution for the clearance inside](image)

5. Conclusions
In this study, a canned motor pump was investigated by numerical and experimental methods. Furthermore, the relationship of axial clearance and leakage rate was shown as the allometric fitting curves. And the conclusions are drawn as follows:
(1) The canned motor pump adopts the structure of floating impeller, which can realize the self-balance of axial force. The key of design is to control the leakage of axial clearance at the hub;

(2) The leakage increases with the increase of axial clearance, showing the function distribution of allometric model. And the index number of the expression is larger than 1;

(3) The static pressure in the axial clearance first increases and then decreases with the decrease of radius, and the maximum static pressure value is about 10.5% ~ 15.8% near the clearance inlet.

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7. References
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