The influence of annular seal clearance to the critical speed of the multistage pump

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Abstract. In the multistage pump of high head, pressure difference in two ends of annular seal clearance and rotor eccentric would produce the sealing fluid force, the effect of which can be expressed by a damping and stiffness coefficient. It has a great influence on the critical speed of the rotor system. In order to research the influence of the annular seal to the rotor system, this paper used CFD method to conduct the numerical simulation for the flow field of annular seal clearance. The radial and tangential forces were obtained to calculate the annular dynamic coefficients. Also dynamic coefficient were obtained by Matlab. The rotor system was modeled using ANSYS finite software and the critical speed with and without annular seal clearance were calculated. The result shows: annular seal's fluid field is under the comprehensive effect of pressure difference and rotor entrainment. Due to the huge pressure difference in front annular seal, fluid flows under pressure difference; the low pressure difference results in the more obvious effect on the clearance field in back annular seal. The first order critical speed increases greatly with the annular seal clearance; while the average growth rate of the second order critical speed is only 3.2%; the third and fourth critical speed decreases little. Based on the above result, the annular seal has great influence to the first order speed, while has little influence on the rest.

1. Introduction
The multistage pump is widely used in the desalination, petroleum, chemical industries, along with the large-scale production plant, the operation stability for the multistage pump as the auxiliary production unit is becoming increasingly stringent, of which the dynamic analysis is very important.

Different with the general rotor system in the rotor dynamic analysis of multistage pump, it not only consider the influence of shaft and journal bearing to the rotor system, but also consider the sealing fluid force generated by the annular seal clearance. Pressure difference and rotor eccentricity would produce sealing fluid force. Matlab software was adopted to slove the film pressure, the dynamic coefficients were obtained by the finite difference method. However, the influence of sealing fluid force to the rotor can not be ignored, of which the effect would be expressed by the sealing dynamic coefficients. Many scholars have carried out research for the annular seal dynamic coefficient. Sun Qiguo [1] researched the dynamic coefficient of clearance annular with the perturbation method. Some scholars [2–4] analyzed the rotor dynamics of gas seal by the two-control volume method.
based on Bulk Flow theory. Also some scholars\textsuperscript{[5–9]} adopt CFD method to calculate the dynamic coefficient of gas or fluid seal.

This paper based on CFD method to obtain the radial and tangential force by the numerical simulation for the annular seal clearance flow filed. Calculation formula of annular seal dynamic coefficient was used to obtain the annular seal stiffness and damping. The rotor system of multistage pump was modeled in ANSYS finite software to conduct the rotor dynamic analysis. The critical speed of rotor system with and without annular seal was calculated and the result was analyzed to provide some reference value for rotor dynamic analysis.

2. Numerical simulation of annular seal clearance

2.1. Grid generation of annular seal and simulation setting
The radial size of the annular seal clearance in this paper of multistage pump is 0.25mm and axial size is 17.5mm. Due to the small annular seal clearance and the relative large size in circumferential, unstructured grid would induce the negative volume to affect the CFX simulation. The grid of annular seal clearance was divided into 8 layers considered the boundary layer effect. The grid of annular seal clearance was shown in figure 1. The grid was checked by ICEM, the grid quality was above 0.9, which ensure the simulation accuracy and convergence. The grid was imported to the CFX software, two ends of the annular seal clearance fluid was set as the pressure inlet and pressure outlet, the inner and external surface was set as the rotating wall. Pressure difference in front annular seal in each stage was set to 2.234MPa; pressure difference in back annular seal was set to 10000Pa within the pressure loss.

![Figure 1. Grid of annular seal clearance.](image)

2.2. Flow field analysis of annular seal clearance
After the simulation of the flow field in the whirling speed of 6500r/min, 8500r/min and 10000r/min, we find that pressure distribution and streamline in the front annular seal field at different whirling speed have little changes. Pressure distribution and streamline at the whirling speed of 10000r/min was shown in figure 2. Pressure distribution and streamline in the back annular seal at different whirling speed have obvious changes, which were shown in figure 3 and 4.

![Figure 2. The diagram of the front annular seal’s pressure distribution and streamline](image)
Contrast above graphics, fluid in the front annular seal clearance from the high pressure side to low pressure side is almost in linear flow, this shows that whirling speed has little influence on the flow field. High pressure difference in two ends of annular seal has greater influence than entrainment on flow field caused by the rotor. Fluid streamline in back annular seal clearance is spiral, which is more obvious with the decrease of whirling speed. This indicates that whirling speed and pressure difference have obvious effect on the flow field of annular seal. With the increase of whirling speed, the effect of entrainment on flow field caused by rotor is gradually larger than low pressure difference, when whirling speed reaches to 10000r/min, fluid is almost in peripheral circulation in annular seal clearance. The radial force and tangential force in different whirling speed of front and back annular seal clearance obtained from CFX was shown in table 1.

| whirling speed/(r/min) | 5000 | 6500 | 8000 | 10000 |
|------------------------|------|------|------|-------|
| radial force in front annular seal/N | 4.434 | 4.9686 | 6.1192 | 7.2114 |
| tangential force in front annular seal/N | 62.249 | 64.2137 | 67.0193 | 69.2916 |
| radial force in back annular seal/N | 0.831 | 2.6274 | 8.2655 | 15.8598 |
| tangential force in front annular seal/N | 69.5678 | 91.0635 | 24.0177 | 151.8596 |

2.3. Calculation of dynamic coefficients of annular seal
The relation of damping coefficient, stiffness coefficient, added mass with sealing fluid force are as follows:

$$
\begin{align*}
\mathbf{F}_r &= \left[ \begin{array}{c}
K & X(t) \\
-k & Y(t)
\end{array} \right] + \left[ \begin{array}{c}
0 & X(t) \\
0 & M
\end{array} \right] \dot{X}(t) \\
\mathbf{F}_\tau &= \left[ \begin{array}{c}
-C & c \\
-c & C
\end{array} \right] \left[ \begin{array}{c}
\dot{X}(t) \\
\dot{Y}(t)
\end{array} \right] + \left[ \begin{array}{c}
0 & M \\
0 & M
\end{array} \right] \ddot{Y}(t)
\end{align*}
$$

(1)

The initial conditions are: \( X(0) = r_0, Y(0) = 0, X(0) = 0, Y(0) = r_0 \Omega, \dot{X}(0) = -r_0 \Omega^2, \dot{Y}(0) = 0 \) and the initial conditions were applied into the formula (1):
\[ F_r / r_0 = -K - c\Omega + M\Omega^2, \]  
\[ F_r / r_0 = k - C\Omega, \] 

(2) 

(3) 

\( r_0 \) is whirling amplitude and 10% of eccentricity; the force in whirling speed of 6500r/min, 8500r/min and 10000r/min were applied to calculate the dynamic coefficient according to the equation (2); the force in whirling speed of 5000r/min and 10000r/min were applied to calculate the dynamic coefficient according to the equation (3); the result are shown in table 2.

| Table 2. The dynamic coefficients of the annular seals |
|-------------------------------------------------------|
| Dynamic coefficient | Front annular seal | Back annular seal |
|----------------------|--------------------|-------------------|
| K(N/m)               | -154669            | -78396            |
| k(N/m)               | 2208256            | -51040            |
| C(Ns/m)              | -538               | -6289             |
| c(Ns/m)              | 30.4               | 2597              |
| M(kg)                | 0.1592             | 2.339             |

3. Calculation of dynamic coefficients for journal bearing

Pressure distribution could be obtained from the Reynolds equation

\[ \frac{1}{R^2} \frac{\partial}{\partial \varphi} \left( \frac{H^3}{\varphi} \frac{\partial p}{\partial \varphi} \right) + \frac{\partial}{\partial z} \left( \frac{H^3}{\varphi} \frac{\partial p}{\partial z} \right) = 6\alpha \left( \frac{\partial h}{\partial \varphi} \right) + 12 \left( \frac{\partial h}{\partial t} \right) \]  

(4) 

\( h \) is oil film thickness, \( p \) is oil film pressure, dynamic coefficient of journal bearing could be calculated by integrating the \( p \) according to the equation (5).

\[
\begin{align*}
K_{xx} &= \int_{-1}^{1} \int_{-1}^{1} \left( \nabla H \cdot \nabla p \right)_z d\phi d\lambda \\
K_{yx} &= \int_{-1}^{1} \int_{-1}^{1} \left( \nabla H \cdot \nabla p \right)_\varphi d\phi d\lambda \\
K_{xy} &= \int_{-1}^{1} \int_{-1}^{1} \left( \nabla H \cdot \nabla p \right)_z d\phi d\lambda \\
K_{yy} &= \int_{-1}^{1} \int_{-1}^{1} \left( \nabla H \cdot \nabla p \right)_\varphi d\phi d\lambda \\
C_{xx} &= \int_{-1}^{1} \int_{-1}^{1} \left( \nabla H \cdot \nabla c \right)_z d\phi d\lambda \\
C_{yx} &= \int_{-1}^{1} \int_{-1}^{1} \left( \nabla H \cdot \nabla c \right)_\varphi d\phi d\lambda \\
C_{xy} &= \int_{-1}^{1} \int_{-1}^{1} \left( \nabla H \cdot \nabla c \right)_z d\phi d\lambda \\
C_{yy} &= \int_{-1}^{1} \int_{-1}^{1} \left( \nabla H \cdot \nabla c \right)_\varphi d\phi d\lambda 
\end{align*}
\]  

(5)

Obtained directly by the finite difference method of the perturbation pressure on the partial derivative, the dynamic coefficients could be obtained by the numerical simulation according to the equation (5) and were shown in table 3.

\[
\frac{\partial}{\partial \varphi} \left( H_0 \frac{\partial p}{\partial \varphi} \right) + \frac{D}{L} \frac{\partial}{\partial \lambda} \left( H_0 \frac{\partial p}{\partial \lambda} \right) + \begin{cases} 
3 \left( \cos \phi - \sin \phi \frac{\partial H_0}{\partial \phi} \right) \\
-3H_0 \left( \cos \phi \frac{\partial p_0}{\partial \phi} \right) + 3H_0 \left( \sin \phi \frac{\partial p_0}{\partial \phi} \right)
\end{cases} = 6 \sin \phi \sin \phi (i = 1, 2, 3, 4)
\]  

(6)
Table 3. Dynamic coefficients of journal bearing

| Dynamic coefficient | Kxx  | Kyx  | Kxy  | Kyy  | Cxx  | Cyx  | Cxy  | Cyy  |
|---------------------|------|------|------|------|------|------|------|------|
| Value               | 0.1752 | 0.3107 | -0.1307 | 0.1471 | 0.3183 | 0.1257 | 0.1257 | 0.4697 |

4. Rotor dynamics analysis

The rotor structure of multistage pump is modeled in ANSYS. The rotor was modeled with BEAM189 Element, impeller was modeled with MASS21 Element, journal bearing and annular seal was modeled with COMB214 Element.

The whirling diagram of first order critical speed of rotor calculated by ANSYS with and without annular seal are shown in figure 5; the Campbell diagram with and without annular seal are shown in figure 6; the value of critical speed in different rotation speed are shown in table 4 and table 5.

![Figure 5. The whirling diagram of first critical speed](image)

(a) with the annual seal  (b) without annular seal

![Figure 6. The Campbell diagram](image)

(a) with the annual seal  (b) without annular seal

Table 4. The critical speed with annular seal in different rotation speed

| Rotation speed(r/min) | First order | Second order | Third order | Fourth order |
|-----------------------|-------------|--------------|-------------|--------------|
| 2000                  | 1880        | 8881         | 15651       | 19127        |
| 3000                  | 1882        | 8920         | 15690       | 19152        |
| 4000                  | 1884        | 8961         | 15725       | 19716        |

Table 5. The critical speed without annular seal in different rotation speed

| Rotation speed(r/min) | First order | Second order | Third order | Fourth order |
|-----------------------|-------------|--------------|-------------|--------------|
| 2000                  | 3854        | 9172         | 15608       | 18967        |
| 3000                  | 3855        | 9218         | 15643       | 18990        |
| 4000                  | 3587        | 9256         | 15678       | 19013        |
Contrast the result in table 4 and table 5, the first order critical speed increases greatly with the annular seal clearance; while the average growth rate of the second order critical speed is only 3.2%; the third and fourth critical speed decrease a little. Based on the above results, the annular seal has great influence on the first order speed and would make the first order speed increase, while has little influence on the rest.

5. Conclusion
This paper make a conclusion of the multistage pump rotor dynamics analysis based on considering the annular seal.

(1) annular seal's fluid field is under the comprehensive effect of pressure difference and rotor entrainment. Due to the huge pressure difference in front annular seal, fluid mainly flow under pressure difference; the low pressure difference result in the more obvious effect on the clearance field in back annular seal.

(2) the first order critical speed increases greatly with the annular seal clearance; while the average growth rate of the second order critical speed is only 3.2%; the third and fourth critical speed decrease a little. Based on the above result, the annular seal has great influence on the first order speed and would make the first order speed increase, while has little influence on the rest.

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