Dynamic Performance and Fatigue Analysis of ASP Pump Rotor

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Abstract. In this paper, static analysis, modal analysis and harmonic response analysis are carried out on the rotor structure of single-stage double-suction centrifugal pump. The fatigue life of the rotating shaft was analyzed with the nCode Designlife module in ANSYS. The results show that the maximum equivalent stress is greater than the allowable stress of the material and is located at the journal of the transmission torque connected with the coupling. The maximum deformation also occurs here at 0.4857mm. The axis of rotation is liable to resonate at the first natural frequency of the non-rigid body. The frequency which has the greatest influence on the vibration of the rotating shaft is near the third and fourth order natural frequencies of modal analysis and should be avoided. The minimum number of circulation at the maximum stress is 1.785e⁷ times, and other positions are above 10¹⁴ times. The results show that the pump group works stably under the design condition and the rotor performance meets the design requirements.

1. Introduction
Shaft parts are essential parts of mechanical systems. The dynamic and static analysis of the shaft includes the study of the deformation, resonance frequency, critical speed and fatigue strength of the shaft[1]. These characteristics have great influence on the stability and reliability of the shaft[2,3].

TP Davis et al. studied the factors affecting the performance of rotor structure and modified the key factors. The relationship between the influence factor and rotor performance is put forward[4]. Urquiza G et al. conducted mechanical behavior analysis on large rotor structures and studied the vulnerable failure sites of rotor structures, which has certain guidance for engineering practice[5]. Uhlmann E et al. analyzed the performance of the rotor structure under various constraints and calculated the stress distribution of the rotor structure. The distribution of thermal stress under multiple constraints is pointed out[6]. Fatigue design research is limited by too little basic data. The analysis of fatigue life of rotating shaft is mainly based on theoretical analysis and simulation analysis.

Based on the finite element knowledge, this paper applies the nCode Design Life module integrated under ANSYS Workbench and its platform. The static analysis, modal analysis, prestressed modal analysis, harmonic response analysis and fatigue life analysis of the rotor were carried out respectively.

2. Centrifugal Pump Rotor Structure
The prototype of the equipment studied in this paper is single stage double suction horizontal centrifugal pump ASP150-355R pump shaft.
The main components and installation positions of the centrifugal pump rotor are shown in Figure 1, which mainly includes the pump shaft, impeller, sliding bearing, mechanical seal, oil retaining ring and coupling.

![Schematic diagram of the centrifugal pump rotor](image)

**Figure 1. Schematic diagram of the centrifugal pump rotor**

The main performance parameters of centrifugal pump are shown in Table 1.

| Parameter          | Numerical value | Unit |
|--------------------|-----------------|------|
| Rotating speed     | 1480            | r/min|
| Entrance diameter  | 250             | mm   |
| Outlet diameter    | 150             | mm   |
| Impeller diameter  | 355             | mm   |
| Outlet pressure    | 2.08            | MPa  |
| Shaft power        | 46              | kW   |
| Head               | 40              | m    |
| Flow               | 350             | m³/h |
| Effectiveness      | 84              | %    |
| Motor power        | 50              | Hz   |
| Cavitation allowance | 2.2         | m    |

### 3. Static Analysis of Rotor

The static analysis of the rotor subsystem mainly investigates whether the rotor strength meets the design requirements. In the actual working process of the rotor, the force is very complex. The main force analysis is as follows:

- The gravity of the pump shaft and impeller itself, as well as the gravity of the coupling;
- The supporting force of a bearing;
- Torque of motor to shaft;
- The radial force Fr generated by the fluid at the impeller outlet.

#### 3.1. Computational Model Preprocessing

In this paper, the analysis is carried out under the design condition, so it is necessary to calculate the radial force under the design condition. The empirical function method is used to calculate the radial force of impeller. Formula for calculating radial force of empirical function method:

$$F = \rho gK_r HD_B^2$$  \hspace{1cm} (1)

Where, D2 is the outer diameter of the impeller (mm); B2 is the width of impeller outlet (mm); H is the head (m); Kr is the experimental coefficient. It can be calculated according to Steponff formula:

$$K_r = 0.36[1 - \left(\frac{Q}{Q_y}\right)^2]$$  \hspace{1cm} (2)
Where, $Q$ is the running flow of the pump (m$^3$/h); $Q_N$ is the flow rate at the highest efficiency point (m$^3$/h). For the annular pressurized water chamber:

$$K_r = 0.36 \frac{Q}{Q_N}$$  \hspace{1cm} (3)

The radial force $F = 456$N can be calculated according to the above formula. From the relation between the output torque of the motor and the speed and power of the motor, the following can be obtained:

$$T = 9550 \frac{P}{n}$$  \hspace{1cm} (4)

Where, $P$ is the motor power (KW); $N$ is the output speed of the motor (r/min); $T$ is the output torque of the motor (N\(\cdot\)m). By substituting the motor's power and speed into the above formula, the output torque is $322.6$N\(\cdot\)M.

3.2. Static Calculation Result

Figure 2 is the stress intensity cloud diagram of the rotor structure.

As can be seen from the above figure, the maximum stress of the rotor is located at the first shoulder of the shaft neck connected to the coupling. The stress value is $234.9$MPa, which is far greater than the allowable maximum bending stress value of the material ($52$MPa), but less than the yield strength of the material ($450$MPa). The red area in the figure is the dangerous section. Under this stress, the strength may cause fracture or other forms of failure of the rotating shaft, which is consistent with the actual fracture position. It is necessary to analyze the fatigue reliability of the rotating shaft.

4. Modal Analysis of Rotor

Modal analysis is a method to study the structural vibration characteristics. The method can obtain the natural mode and natural frequency of each order to detect whether the working frequency of the structure is the same as the natural frequency and whether there is resonance. Modal analysis is the basis of dynamics research.

Constraints are the main factors that affect the results of modal analysis. The natural frequencies and modes of the first six orders are extracted from the modal analysis Settings. The natural frequencies are shown in Table 2.

| Modal Order | First-order | Second-order | Third-order | Fourth-order | Fifth-order | Sixth-order |
|-------------|-------------|--------------|-------------|--------------|-------------|-------------|
| Frequency/Hz| 0.1445      | 0.1537       | 64.045      | 64.108       | 235.79      | 334.63      |

Due to the symmetrical structure, the frequency of the first two modes is close to zero. The cloud diagram of the total deformation of the first six modes is shown in Figures 3 to 8:

The first-order mode (0.1445Hz) is the torsional deformation vibration at the journal. The deformation is symmetrically decreasing from the impeller along the X-axis, and the two ends of the rotating shaft swing up and down along the positive and negative directions of the Z-axis, with a maximum value of 16.748mm.
Figure 3. First-order mode

Figure 4. Second-order mode

Figure 5. Third-order mode

Figure 6. Fourth-order mode

Figure 7. Fifth-order mode

Figure 8. Sixth-order mode

The second-order mode (0.1537Hz) is close to the first-order mode (0.1537Hz). The deformation distribution is similar to the first order and the value is equivalent to the rigid body mode. The rotating shaft swings back and forth along the Y-axis direction and at the same time swings up and down along the Z-axis. The maximum value is 16.747mm.

The third-order mode (64.045Hz) maximum deformation occurs at the left end journal. The rotor rotates in the third quadrant, and the axis bends along the positive direction of the Z-axis. The maximum value is 11.56mm.

The fourth-order mode (64.108Hz) and the third-order mode (64.045Hz) have similar deformations and similar values. The rotation axis bends along the positive direction of Y-axis, and the maximum value is 11.548mm.

The deformation of the fifth-order mode (235.79Hz) is very small, which mainly occurs at the outlet edge of the impeller. Due to the radial force generated by centrifugal action, the distribution of impeller deformation gradually increases along the radial direction from the center, with a maximum value of 15.86mm.

The maximum deformation of the 6th-order mode (334.63Hz) occurs at the shaft journal, and the maximum value is 16.413 mm.

Through the analysis of the first six modes of the rotor, it can be known that the modes are relatively complex, including both forward and backward oscillation and left and right oscillation, as
well as torsional vibration and bending vibration. The maximum vibration displacement deformation is in the journal, which will affect the strength of the structure. During operation, the mode with close frequency should be avoided as far as possible to avoid resonance. The rotor's torsional deformation force is provided by the rotating shaft at the third or fourth natural frequency. The rotating shaft is prone to resonance near the frequency of 64Hz, and it should be avoided to operate at this frequency.

5. Rotor Harmonic Response Analysis
In this paper, the harmonic response of rotor is studied by means of modal superposition method. According to statics and modal analysis, the dangerous section of the rotor is located at the journal where the torque is transferred, and the maximum value of the modal displacement deformation is also at the journal. The main load of the shaft is the torque transferred by the motor. Since the natural frequencies of the first six orders in the modal analysis range from 0 to 334.63Hz, the maximum value in the analysis setting (the sixth natural frequency) should be 1.5 times less than the calculated value of the modal. Therefore, the frequency variation Range of harmonic response analysis is set as 0 to 200Hz, the Range Maximum is set as 200, and every 5Hz is set as one step for a total of 40 steps for mode superposition. Other fixed constraints are the same as for modal analysis.

The axial segment at the left and right journal was selected as the observation point for harmonic response analysis. The response spectrum is shown in Figures 9 to 14.

![Figure 9. X-direction displacement response at the right end journal](image)

![Figure 10. Y-direction displacement response at the right end journal](image)

![Figure 11. Z-direction displacement response at the right end journal](image)

![Figure12. X-direction displacement response at the left end journal](image)

From Figures 9 to 14, it can be seen that there is a peak near 65Hz in all directions of the right end journal. The x-direction peak value is 5.015e-3mm, the y-direction peak value is 0.1357mm, and the z-direction peak value is 4.8419mm, which is very close to the third and fourth order natural frequencies. There was also a peak in all directions of the left end journal around 65Hz, in which the x-direction peak value was 1.6036e-2mm, the y-direction peak value was 0.13147mm, and the z-direction peak value was 5.1876mm, and the frequency was close to the third and fourth natural frequencies. The rotating shaft is prone to resonance near 65Hz. The working frequency of the pump should be avoided near the resonance frequency to avoid safety accidents.
6. Shaft Fatigue Analysis
Among various failure modes, structural fatigue failure is one of the most important failure modes. In practical work, the rotating shaft is subject to complex forces and unstable impacts, and the overall load tends to be white noise wave or other loads with drastic changes. Under this kind of cyclic action, the shaft system tends to produce large stress concentration at the structural mutation. This section uses the nCode Design Life fatigue analysis module in ANSYS Workbench to calculate the fatigue Life of the rotating shaft.

6.1. Basic Steps of Fatigue Analysis
Firstly, the type of fatigue load spectrum and its collection and analysis are determined. Then perform fatigue analysis on the shaft, mainly including the choice of analysis method and correction method. Finally, the fatigue life is predicted. The basic process of fatigue analysis is shown in Figure 15.

6.2. Axis Model Analysis
On the basis of statics, nominal stress method is used to analyze the fatigue of rotating shaft. The S-N curve of the material is shown in Figure 16. From the previous analysis, it can be seen that the maximum stress of the rotor is at the journal where the torque is transferred, so the fatigue analysis can only be carried out for the rotating shaft.

6.3. Fatigue Analysis of Rotating Shaft Based on nCode Module
Input the static analysis results into nCode SN Time Series (DesignLife). It is known from static analysis that at least two of the three principal stresses of the structure are non-zero, which is obviously
a multi-axial fatigue problem. Run nCode SN TimeSeries (DesignLife) for fatigue analysis, select the load Combination as a Text Combination, and Display the results by XY Display. The display results are shown in Figure 17.

![XY Display](image)

**Figure 17. Load-time history**

The calculation engine S-N Method was set to be MultiRRatioCurve, the stress combination mode was SignedVonMises, the correction mode was FKM, and the survival rate was 50. The cloud diagram of the life of the rotating axis is shown in Figure 18. In the figure, the color on the right side from low to high represents the number of cycles from low to high, that is, the life span from short to long. From the figure, it can be seen that the minimum number of cycles occurs at the stress concentration journal of the transferred torque, corresponding to the stress concentration of the static analysis, and its value is $1.785 \times 10^7$. Most of the other parts have more than $10^{14}$ cycles.

![Life diagram of the shaft](image)

**Figure 18. Life diagram of the shaft**

The damage cloud image of the rotating axis is shown in Figure 19. Red to blue in the figure represents the damage degree from high to low. The maximum damage occurred at the stress concentration of the journal where the torque was transferred, which corresponds to the static analysis and lifetime cloud map.

According to the cloud map of life and damage, the position with the shortest life is the position with the maximum stress of the journal, which is the same as the result of static analysis. Before high cycle fatigue failure, the alternating stress is often much smaller than the yield limit of the material, and the cycle times are high, generally $10^5$-$10^7$ times [7]. The life span reached $10^7$ times at the stress concentration, which is satisfactory. Combined with the results of fatigue analysis, it can be seen that the fatigue failure of the rotating shaft will occur at the shortest fatigue life. However, the number of failure cycles in this position also exceeds the service life of the general shaft, so the design of this shaft is reasonable.
7. Conclusion

In this paper, the static analysis, modal analysis, harmonic response analysis and fatigue Life analysis of the rotor are carried out by ANSYS Workbench and nCode Design Life.

The results of static analysis show that the maximum equivalent stress is concentrated in the shaft neck which is connected with the coupling, which is consistent with the actual fracture position. The maximum equivalent stress is 234.9MPa, which is much greater than the allowable maximum bending stress of the material (52MPa), but less than the yield strength of the material (450MPa). Therefore, it is necessary to analyze the reliability of the rotating shaft. In the modal analysis, the first six orders of natural frequencies and modes are extracted, and the comparison with the external excitation shows that the rotation axis is prone to resonance under the first order of non-rigid natural frequencies. Through harmonic response analysis, it is found that the frequency with the greatest influence on the vibration of the rotating shaft is around the third and fourth order natural frequencies of the modal analysis, and its value is 65Hz. Vibration near this frequency should be avoided to avoid resonance phenomena. According to the fatigue analysis, the maximum damage position and the minimum fatigue life cycle number of the rotating shaft are located at the journal of the transmission torque connected with the coupling, which are corresponding to the results of the static analysis. The minimum number of fatigue life cycle at the maximum stress is 1.785e7 times, and other positions are above 1014 times. For the high cycle fatigue analysis and the fatigue life limit of the ordinary shaft, the cycle number of 107 times is satisfactory. In summary, the pump shaft is reasonable in design.

References

[1] Cheng Feng. Study of Vibration Characteristic of Spindle System in a Axial Flow Fan[J]. Coal Mine Machinery, 2016, 37(3), 71-73.
[2] Chang Hao, Liu Jianrui and Li Wei et al. Modal analysis of high flow self-priming centrifugal pumps shaft and crank-shaft[J]. Journal of Drainage and Irrigation Machinery Engineering, 2016, 34(12), 1035-1039+1076.
[3] Wang Zhenghao and Zhang Dujuan. Vibration Dynamics Analysis of CJ190Z4 Machine Tool Spindle Experimental Model Based on Ansys Workbench[J]. Journal of Shenyang Jianzhu University ( Natural Science), 2018(1), 141-149.
[4] Belwal T, Giri L and Bhatt I D, et al. An improved method for extraction of nutraceutically important polyphenolics from Berberis jaeschkeana C.K. Schneid. fruits[J]. Food Chemistry, 2017, 230, 657-666.
[5] Urquiza G, García J C and González J G, et al. Failure analysis of a hydraulic Kaplan turbine shaft[J]. Engineering Failure Analysis, 2014, 41(5), 108-117.
[6] Uhlmann E and Hu J. Thermal Modelling of a High Speed Motor Spindle[J]. Procedia Cirp, 2012, 1(1), 313-318.
[7] Sun Xue, Dai Ying and Wang Xiaofei, et al. Fatigue Characteristic Analysis of the Shaft of High Speed Induction Motor[J]. Motor and Control Application, 2018, 45(3), 97-102.