Strength analysis of motor shaft based on ABAQUS

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ABSTRACT: Focusing on the situation of the motor shaft breaking in the work, using finite element software ABAQUS, considering the coupling effect of interference fit, working load, and eccentric load, the strength analysis of the motor shaft is carried out. The results show that the interference fit will cause the stress concentration of the motor shaft, and the stress safety factor of the motor shaft under the working load can meet the use demand, while the eccentric load will significantly change the stress distribution of the motor shaft and reduce its safety factor. The variation of assembly interference and eccentric load will significantly affect the stress distribution of motor shaft.

1. Introduction

As an important component of engine cooling system, the working condition of quantitative motor affects the reliability of the whole engine. As the power output part of the quantitative motor, the power input end of the motor shaft is connected with the hydraulic plunger by ball hinge, and the output end is connected with the cooling fan shaft by hub. According to the different cooling requirements of engine, quantitative motor has complex working conditions. Due to the bearing wear or the misalignment of the shaft hole caused by the nonstandard assembly, the quantitative motor may run under unbalanced load, which seriously affects the bearing capacity of the motor shaft and reduces its service life. Therefore, in the motor shaft strength analysis, the influence of assembly characteristics, working load and eccentric load operation should be considered comprehensively.

Many scholars have studied the dynamic characteristics of the motor shaft during operation. Ma, Z.L. et al. [1] established a finite element model of the motor shaft hole assembly considering the manufacturing characteristics, and carried out research on the impact of the manufacturing characteristics of key parts on the mechanical characteristics of the entire assembly, revealing that manufacturing errors have a significant impact on the normal operation of the motor. Tao, D.F. et al. [2] used explicit dynamics to study the influence of mass scaling on the calculation accuracy of the stress, contact pressure and load torque of the wind power lock disk shaft hole connection structure. Xie, Z.H. et al. [3] studied the influence of interference, key length and external load on the connection strength of shaft and hub. Aiming at the failure of the fan motor shaft and sleeve of the cooling system, Zhou, K.Z. [4] carried out research from the aspects of structure, dimensions, material properties, etc., and determined that the misalignment of the motor shaft and the sleeve was the main reason for the failure. Kang, D. et al. [5] analyzed the influence of interference, friction coefficient and geometric error on the contact stress. Wu, P. [6] took the motor shaft which failed in the verification test of electric reciprocating saw as the research object, and determined that the tooth fracture of motor shaft belongs to fatigue fracture under alternating stress.

Considering the working characteristics of the motor fan combination structure, this paper
summarizes the typical working conditions including speed working condition, torque working condition and eccentric working condition. The load boundary conditions of the combination structure under each working condition are obtained by dynamic analysis and theoretical calculation. The static analysis is carried out by using the finite element software ABAQUS to study the influence of interference fit, working load and eccentric load on the stress distribution of the motor shaft.

2. Determination of load boundary conditions

According to the actual working conditions of the cooling system, the research conditions are determined, including the speed condition and the maximum torque condition.

For the rotational speed condition, because the rotational speed is constant, the load at the impeller end is relatively stable, and the load at the ball joint fluctuates greatly, which has a great impact on the working stress of the motor shaft. Therefore, the load at the ball joint is taken as the boundary condition of the rotational speed condition. In order to obtain the load at the spherical hinge, the multi-body dynamic analysis software ADAMS was used for dynamic analysis.

For torque condition, the output power of the motor shaft is mainly used to overcome the flow air resistance at the impeller end. The resistance torque at the impeller end is equal to the output torque of the motor shaft. Taking the torque at the input end of the motor shaft as the load boundary condition, the value can be calculated by the power relationship.

2.1. Dynamic analysis of motor fan combination structure

When the motor works, the oil pressure acts on the plunger, and the plunger drives the motor shaft to rotate through the ball hinge connection[7]. In order to obtain the load on the ball hinge when the impeller works at the maximum speed and rated speed, it is necessary to carry out the dynamic analysis of the working process of the quantitative motor.

The analysis structure is mainly composed of hydraulic cylinder, plunger, motor shaft, fan shaft, shaft sleeve and impeller. The model is shown in Fig 1.

Fig 1 dynamic simulation model of motor fan combination structure

According to the working principle of the hydraulic motor, the motor fan combination structure is connected and constrained, as shown in Table 1.

Table 1 connection settings of components

| Component pair                  | Connection form               |
|---------------------------------|-------------------------------|
| Impeller and shaft sleeve       | Fixed pair                    |
| Shaft sleeve and fan shaft      | Fixed pair                    |
| Motor shaft and fan shaft       | Fixed pair                    |
| Motor shaft and plunger         | Spherical hinge               |
| Plunger and hydraulic cylinder  | Contact                       |
| Impeller and ground             | Rotating pair                 |
| Hydraulic cylinder and ground   | Rotating pair                 |
| Motor shaft and hydraulic cylinder | Speed ratio correlation     |

A rotating drive is added to the impeller to simulate the rotating process of the cooling fan. The maximum speed and rated speed angular speed are 27000 °/s and 12000 °/s respectively.

The simulation time is set to 5s, and the steps are set to 4500 and 2000 respectively. The force curves
of the spherical hinge of the motor shaft are shown in Fig 2.

Fig 2 load at the spherical hinge of motor shaft at maximum and rated speed

It can be seen from Fig 2 that under the maximum speed, the load of motor shaft spherical hinge is obviously higher than the rated speed, and the peak value is about 2.3 times. The load at the two spherical joints is symmetrical, and the load changes periodically in X, Y and Z directions. Fourier transform the time domain data under the maximum speed and rated speed to obtain the load spectrum, as shown in Fig 3.

Fig 3 load spectrum of motor shaft spherical hinge at rated and maximum speed

The results show that the loads at the two spherical hinges have a peak value at 75Hz and 33.3Hz respectively, which are the rotation frequencies of the impeller, indicating that the loads at the spherical hinge have periodicity, with the periods of 0.0133s and 0.03s respectively. Based on this, the load data of one cycle is selected to input the load at the speed condition.

2.2. output torque calculation of quantitative motor

According to the basic formula of hydraulic quantitative motor and the power output requirements of vortex fan, the output torque of motor shaft is obtained according to formula 1.

\[
M = \frac{D \times \Delta p \times \eta_{hm}}{63} [N \cdot m] \quad (1)
\]

Where D is the displacement of the motor; \( \Delta p \) is the pressure difference between the oil inlet and the oil outlet, \( \eta_{hm} \) is the mechanical efficiency.

According to the quantitative motor model, the displacement of the motor is 9.8cm\(^3\)/rev, the maximum continuous pressure difference is 350bar, and the mechanical efficiency is 90% under the limit condition. According to formula 1, the output torque is 49 N·m.

3. Strength analysis of motor shaft

3.1. Establishment of finite element model

Based on the quantitative analysis of the load boundary conditions and the connection relationship between the components in the working process of the motor, the finite element analysis structure is determined, including motor shaft, motor shaft key, fan shaft, shaft sleeve and impeller. The motor shaft and key adopt interference assembly. In order to ensure the simulation accuracy, it is necessary to refine the mesh of the motor shaft keyway and key. Hexahedral elements are used for mesh generation. The number of finite elements is 44993. The finite element model is shown in Fig 4.
The material properties of each component are shown in Table 2.

| Components  | Density kg/m³ | Elastic modulus MPa | Poisson’s ratio | Allowable stress MPa |
|------------|---------------|---------------------|-----------------|----------------------|
| Motor shaft | 7.85e+3       | 2e+5                | 0.3             | 930                  |
| Fan shaft  | 7.78e+3       | 2.11e+5             | 0.277           |                      |
| Key        | 7.89e+3       | 2.09e+5             | 0.269           |                      |
| Shaft sleeve | 7.78e+3     | 2.11e+5             | 0.277           |                      |
| Impeller   | 2.68e+3       | 7e+4                | 0.33            |                      |

The motor shaft and key adopted the initial interference contact to simulate the interference assembly. According to the interference fit standard and the motor shaft size, the interference amount is determined to be 0.02mm. The fan shaft and shaft sleeve are connected by tie, and the connection between the shaft sleeve and the impeller adopts the pre tightening force to simulate the bolt pre tightening.

3.2. Stress analysis of interference fit

The stress nephogram of motor shaft due to interference assembly is shown in Fig 5.

It can be seen from the stress nephogram that the interference assembly will lead to stress concentration around the keyway fillet. The maximum stress is 115.9MPa at the keyway fillet, and the rest of the stress is below 77MPa. The stress of keyway should be paid more attention in the subsequent loading analysis.

3.3. Speed condition

The radial constraint is used to simulate the bearing constraint. Constrain the freedom of impeller end X, Y and Z directions, establish 6 unit loads at the loading point of the ball joint, apply the rated speed and maximum speed single cycle load spectrum obtained by dynamic analysis, and conduct stress analysis under the rotating speed condition. The boundary conditions are applied as shown in Fig 6.
Fig 6 boundary condition setting of speed condition

Through the simulation calculation, the stress nephogram of the motor shaft in a single cycle can be obtained, and the nephogram of the maximum stress moment is shown in Fig 7.

Fig 7 maximum stress nephogram under rated speed condition and maximum speed condition

It can be seen from Fig 7 that at rated speed, the maximum stress is 219.2MPa on the Right side of keyway (viewed from the motor end), and the rest of the stress is 70MPa-140MPa, while the maximum stress at the maximum speed occurs on the right side of the keyway (from the motor end), reaching 480.4MPa, and the rest of the stress distribution is 160MPa-320MPa. The maximum stress value at the maximum speed condition is far greater than that at the working speed condition. Compared with the stress nephogram of the motor shaft under interference fit, the stress distribution around the keyway is changed by loading, and the stress concentration at the fillet of the keyway is intensified.

By observing the stress nephogram of each time in a single cycle, it can be seen that the maximum stress is always distributed at the fillet of keyway (far away from the blade end). For the convenience of research, the maximum stress value of each stress nephogram is counted. The stress value on the left side of the fillet is marked as negative, and that on the right side is marked as positive. The maximum stress distribution curves of the maximum speed and rated speed in a single cycle are obtained, as shown in Fig 8.

Fig 8 single-cycle maximum stress curve under rated speed condition and maximum speed condition

It can be seen from the stress curve that when the motor shaft is running at rated speed and maximum speed, the stress distribution around the keyway is constantly changing. Because the motor shaft rotates in one direction, the stress distribution on both sides of the keyway of the motor shaft is asymmetric, that is, the maximum stress on the right side is always higher than that on the left side, and the maximum stress on the right side is significantly higher than that on the left side in the single rotation cycle. The peak stress of the two speed conditions is shown in Table 3.
3.4. maximum torque condition
Constrain the degrees of freedom in the three directions X, Y and Z of the motor shaft end, and apply the maximum torque 49N \cdot m at the impeller loading point. The boundary conditions are set as shown in Fig 9.

![Fig 9 boundary conditions setting of rated speed working condition](image)

The stress nephogram of motor shaft under maximum torque condition is shown in Fig 10.

![Fig 10 cloud diagram of maximum stress under maximum torque condition](image)

According to the stress nephogram, the maximum stress occurs on the right side of the keyway (viewed from the motor end), reaching 464.1MPa. The safety factor is 2.004, which meets the requirements of normal operation.

3.5. Eccentric load condition
Under the ideal condition, the motor axis and fan axis coincide, which is the normal working condition. However, due to nonstandard assembly, serious wear and shaft deformation, the motor axis and fan axis do not coincide, which is the eccentric load working condition. The eccentric load operation will make the load condition of the components worse, which is easy to make the components fail. On the basis of rated speed condition, maximum speed condition and maximum torque condition, the finite element analysis of eccentric load condition is carried out.

Based on the boundary conditions of speed condition and maximum torque condition, the displacement constraint is added to the constraint position of fan shaft bearing to simulate the offset between motor shaft and fan shaft axis. The setting of boundary conditions of eccentric load condition is shown in Fig 11.
According to the fracture detection report of the motor shaft, the runout between the motor shaft and the fan hub is about 0.2 mm, and the radial displacement is 0.1 mm. Taking the angle of extruding the side wall of the keyway as the starting angle, 36 angles are selected with 10° as the interval, and the single rotation cycle eccentric load simulation is carried out.

The load imposed by the maximum stress nephogram is selected as the load boundary condition under the rated speed condition and the maximum speed condition.

The maximum stress nephogram of single cycle under three eccentric load conditions is obtained, as shown in Fig 12.

Compared with the stress nephogram under normal operation condition and eccentric load condition, the stress under eccentric load condition increases significantly, and the maximum stress under three conditions increases by 441.4 MPa (201.4%), 404.8 MPa (91.3%), and 441 MPa (95.0%). It can be seen that under the eccentric load condition, the eccentric load and the eccentric load angle have a significant effect on the stress distribution of the motor shaft, while the load boundary condition has little effect on the maximum stress value of the motor shaft. The maximum stress is always on the right side of the keyway (far away from the impeller end) under the three eccentric load conditions, which indicates that this part is easy to fracture. Under the three eccentric load conditions, the safety factors are 1.408, 1.012, and 1.028 respectively, which are less than 1.5, and the safety margin is insufficient. Compared with the normal operation conditions, the safety margin is reduced by 75.30%, 47.73%, and 48.70%, which is significantly reduced.

In a single cycle, the maximum stress position of motor shaft under three working conditions is around the fillet of keyway, and the stress variation curve of single rotation cycle under three working conditions is shown in Fig 13.
According to the stress results under three kinds of eccentric load conditions, the eccentric load changes the stress distribution of the motor shaft. Under normal conditions, the maximum stress of the motor shaft is almost located on the right side of the keyway. For the eccentric load condition, the maximum stress of the motor shaft is located on the left side of the keyway in a single eccentric load cycle of about 150 degrees, and the maximum stress is obtained under the three kinds of eccentric load conditions when the angle of the eccentric load is 360 degrees, which indicates that there is no significant difference. Compared with the working load, the effect of eccentric load on the stress distribution of motor shaft is more significant.

4. Research on influencing factors
According to the research, the interference and eccentric load will affect the maximum stress around the keyway of the motor shaft. Taking the motor shaft model under the maximum torque condition as the research object, the influence law of the interference and eccentric load on the maximum stress of the motor shaft is deeply studied.

4.1. Influence of assembly interference on stress distribution of motor shaft
Five groups of research variables with initial interference of 0.01-0.05mm and interval of 0.01 are set for regular simulation analysis. The initial interference stress around the keyway of motor shaft is shown in Fig 14.

It can be seen from Fig 14 that the stress distribution around the keyway of the motor shaft is significantly affected by the initial assembly interference. With the increase of the initial interference, the influence range of the stress around the keyway increases significantly. In all cases, the maximum stress of the motor shaft is always around the fillet, and the maximum stress increases significantly with the increase of the interference.

Under eccentric load condition, the maximum stress of motor shaft in single rotation cycle is always around the fillet of keyway. The maximum stress of motor shaft under different initial interference in single rotation cycle is shown in Fig 15.
It can be seen from the Fig 15 that the initial interference has little effect on the variation trend of the maximum stress of the motor shaft in the single eccentric load cycle. With the increase of the initial interference, the maximum stress increases slightly. With the increase of the initial interference, the change of the maximum stress position of the motor shaft lags behind, and the angle of the peak stress on the left side of the keyway is slightly ahead of time. The simulation results show that the maximum stress in a single cycle increases by 2.88%, 2.89%, 3.86% and 4.34% with the increase of 0.01mm, which indicates that the initial interference will have a certain impact on the maximum stress of motor shaft, and the increase of maximum stress is obviously accelerated with the increase of initial interference.

**4.2. Influence of eccentric load on stress distribution of motor shaft**

Five groups of research variables were set with the bias load of 0.06-0.14mm and the interval of 0.02. Through regular simulation analysis, the maximum stress of motor shaft under different eccentric loads in a single rotation cycle is shown in Fig 16.

It can be seen from Fig 16 that the eccentric load has a great influence on the variation trend of the maximum stress of the motor shaft, and with the increase of the eccentric load, the maximum stress increases significantly. With the increase of eccentric load, the angle range of the maximum stress on the left side of the keyway increases gradually from about 75° to 170° and the peak stress on the left side of the keyway appears near 195° under each eccentric load. The simulation results show that the maximum stress of a single cycle increases by 15.29%, 17.05%, 16.00% and 14.09% with the increase of 0.02mm of eccentric load, which indicates that the eccentric load will significantly increase the stress of motor shaft and reduce its service life.
5. Conclusion
This paper aims at the fracture problem of motor shaft during operation. The load boundary conditions are obtained by theoretical calculation and dynamic simulation. Considering the interference fit and eccentric load, the following conclusions can be drawn:

1. Under the condition of speed and maximum torque, the strength of motor shaft meets the requirements of normal operation.
2. Under the condition of eccentric load operation, the stress of motor shaft increases significantly, about 440MPa, which has more significant effect on the stress of motor shaft than the working load, and will significantly reduce the service life of motor shaft.
3. The stress concentration around the fillet of keyway of motor shaft will be caused by interference fit, and with the increase of interference, the affected area of stress around keyway will become larger obviously, and the maximum stress increase will accelerate with the increase of interference under eccentric load condition.
4. With the increase of eccentric load, the angle range of stress peak on the left side of keyway increases, and the maximum stress increases significantly.

Reference
[1] Ma, Z.L., Jin, X., Xiao, M.Z., etc. Finite element analysis of motor shaft hole assembly based on manufacturing characteristics [J]. Navigation and control, 2018, 017(003):26-32.
[2] Tao, D.F., Wang, J.M., Tang, L., et al. Application of mass scaling in wind power locking plate assembly analysis [J]. Journal of Taiyuan University of science and technology, 2013, 34 (1): 32-36.
[3] Xie, Z.H. Study on torque transfer characteristics of heavy duty hub composite connection [D]. Zhejiang University .
[4] Zhou, K.Z. Failure analysis of fan motor shaft and shaft sleeve in cooling system of fracturing truck [J]. Management and technology of small and medium enterprises, 2016, 06 (V.28): 172-173.
[5] Kang, D., Li, Y., Yang, Y.W., et al. Finite element analysis of axle fit based on ABAQUS [J]. Henan science and technology, 2019, 663 (01): 59-62
[6] Wu, P. Fracture failure analysis of motor shaft of electric reciprocating saw [J]. Modern manufacturing technology and equipment, 2017, 000 (005): 83134
[7] Yang, C.Y., Li, Z.L. Analysis of load characteristics of variable displacement pump and quantitative motor volumetric speed control system [J]. Modern machinery, 1999, 000 (002): 22-24