Structural Optimization of Vehicle Turbocharger Rotor

Zhihao Liu1, *, Renren Wang2, Fang Cao1, * and Pidong Shi1

1 School of Mechanical and Automotive Engineering, Qilu University of Technology (Shandong Academy of Sciences), Ji’nan 250300, Shandong, P. R. China
2 School of Electrical Engineering and Automation, Qilu University of Technology (Shandong Academy of Sciences), Ji’nan 250300, Shandong, P. R. China

*Corresponding author e-mail: caofang@qlu.edu.cn, liuzhihao_t@163.com

Abstract. A solid-body model of the rotor system of the turbocharger for light trucks is built. The influence of the oil film bearing support and the material internal damping of the rotor system is studied by using finite element method (FEM), the parameters of the rotor shaft diameter are optimized. The results show that the material internal damping of rotor and the oil film bearing support have complex influence on the dynamic behavior of the rotor and which reduces the running stability of the rotor.

1. Introduction

As the core component of turbocharger, stable operation of high-speed rotating rotor is very important and necessary. At present, the optimization of stable operation of turbocharger rotor is mainly concentrated on oil film, and there is little research on structural optimization. Xin Yida [1] established a finite element model of the oil film of the floating ring bearing and analyzed the influence of the pressure of the supplied lubricant on its lubrication performance, and further analyzed the influence of the floating ring bearing on the stability of the rotor. The simulation results show that both the inner and outer oil films have obvious convergence and divergence regions and the divergence region will cause cavitation in some areas of the oil film and the oil film has higher negative pressure. Finally, it was concluded that increasing the pressure of the supply of lubricating oil can reduce the divergence of the inner and outer oil film, stabilize the oil film pressure, and thus improve the bearing lubrication performance and rotor stability. Zhang Hong [2] established a non-linear oil film support dynamic model for a self-designed marine turbocharger rotor. It is found that the nonlinear force of the oil film dominates the vibration. Yan Wenzhong [3] optimized the structure of the low-pressure rotor of the turbofan engine and analyzed the critical speed of the rotor. After structural optimization, the critical speed is higher than the design working speed. Ren Yongsheng [4] proposed a dynamic model of SMA filamentous composite shaft-rigidly supported rotor to study the stability and vibration of the rotor. The study found that the SMA filamentous improve the static and dynamic natural frequencies of the rotor, which increase the critical speed, and makes the rotor run stably in a certain speed range. According to the current rotor instability factors, the rotor operation stability is improved around the rotor structure. The optimized rotor is analyzed for damping ratio coefficient, stable limit speed and unbalanced harmonic response.
2. Optimization method and Finite element model

2.1. Rotor geometry model
This model is the turbocharger rotor of light truck. As shown in figure 1, the components of the rotor. The red dashed line box is an optimized shaft segment, and the optimization purpose is achieved by changing the diameter of the shaft segment. Turbine Wheel’s material is nickel alloy (K418). Compressor Wheel and Nut’s material is cast aluminium alloy (ZL105). Rotating Shaft’s material is alloy steel (42CrMo). Thrust Bearing and Seal sleeve’s material is stainless steel (316 Stainless Steel). Three different diameter shafts are compared with the original diameter shaft, as shown in figure 2.

![Figure 1. Components of rotor](image1)

![Figure 2. Three kinds of optimization and original size of shaft](image2)

2.2. Finite element model
Rotor dynamic equation:

\[
(M_t + M_r)\ddot{q}(t) + (G + C_c + C_b)\dot{q}(t) + (K_b + K_s)q(t) = f(t)
\]

(M<sub>t</sub> is translation inertia, M<sub>r</sub> is rotational inertia, G is gyro torque, C<sub>b</sub> is bearing support damping, C<sub>c</sub> is material damping, K<sub>b</sub> is bearing support stiffness, K<sub>s</sub> is shaft stiffness, q(t) is total displacement vectors, \dot{q}(t) is total velocity vectors, \ddot{q}(t) is total acceleration vectors, f(t) is global force vectors.

As shown in figure 3, the rotor finite element model and the bearing support and constraint position.
2.3. Bearing and Material parameters data

Table 1 shows the material properties required for the calculation. Figure 4 shows the stiffness and damping parameters of the floating ring bearing. This data is provided by turbocharger manufacturer. It is used for subsequent simulation analysis.

Table 1. Parameters of materials in each part

| Material name   | Density  | Young's modulus | Poisson's ratio |
|-----------------|----------|-----------------|-----------------|
| K418            | 8000 kg/m$^3$ | 2.11E+11        | 0.26            |
| ZL105           | 2680 kg/m$^3$ | 7E+10           | 0.33            |
| 42CrMo          | 7850 kg/m$^3$ | 2.12E+11        | 0.29            |
| 316 Stainless Steel | 7954 kg/m$^3$ | 1.95E+11        | 0.25            |
3. Results and discussion

3.1. Critical speed analysis
As shown in figure 5, this figure is the first-order speed frequency diagram (whirl frequency diagram) of the rotor. From the curve as a whole, the rotor frequency increases with the rotating speed, which is called forward whirl. However, the change of shaft diameter will have certain influence on whirling frequency curve. Before the 40800 rpm speed, the frequency of the original size of the shaft is higher than the optimized size of the shaft. Due to the rotor is in rigid body mode and bearing damping plays a main function. After the speed of 40800 rpm, the frequency of the optimized size is higher than that of the original size. At this time, the rotor is in a flexible state (rotor softening), and the material damping of the rotor plays a main function.

![Figure 5. Rotor first-order whirl frequency](image)

3.2. Damping ratio coefficient and Stability analysis
Figure 6 shows the curve of the damping ratio coefficient of the rotor changing with the speed. The region of rotor rotation stability is above the zero line, and below the zero line is the region of instability. It can be seen from the local enlarged view that the rotor of the original size starts to lose stability at 56568 rpm, and it is harmful to the rotor to enter the unstable region prematurely. However, the stable limit speed of the optimized rotor increases with the shaft diameter. At least 1% increase in the limit steady speed. At the same speed, the damping ratio coefficient increases with the optimal size, which means that the material damping of the rotor decreases gradually. It is verified that reducing the material damping can improve the operation stability of the rotating machinery, which is the goal to be achieved.
3.3. Unbalanced Mass Harmonic Response Analysis

Figure 7 shows the frequency response amplitude of the turbine side bearing center position (P5). The two peaks correspond to the steering frequency of the rotor. It can be observed from the figure that the steering frequency increases with the diameter of the optimization shaft, which increases the response amplitude. This shows that increasing the shaft diameter reduces the damping of the rotor. It means that the shaft diameter cannot be increased blindly, but the vibration displacement caused by its unbalanced response should be considered. According to the response standard, the shaft optimized size was finally locked at 7.50 mm to 8.00 mm.
4. Conclusion

1) The frequency of rotor whirl increases with the rotating speed. The whirl frequency curve shows that the rotor is in rigid and flexible modes. The frequency of shaft optimization will be slightly higher than the original size.

2) The finite element analysis shows that the stability limit speed of the rotor will increase with the increase of the shaft diameter. The damping of the rotor will be reduced accordingly.

3) Through the analysis of the unbalanced response of the rotor, the steering frequency of the rotor will increase with the shaft diameter, and the displacement response amplitude will also increase. Finally, the optimization range of shaft diameter is determined by response standard.

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