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Study on Performance of Automobile Gasoline Mechanical Turbocharger Based on Finite Element Analysis

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In order to improve the design quality and benefit of the automobile gasoil mechanical turbocharger, the finite element analysis method is proposed in this paper. Through the analysis of the semi-floating bearing structure, rotor dynamic analysis of small vehicle gasoline engine turbocharger have been done, including the calculation and analysis of critical speed and unbalance response. From the results of fatigue analysis and influence of super high pressure by selecting parts of geometry size as the design variables, in terms of meeting certain intension and toughness demands, the frame volume is selected as objective variable, and by adopting zero rank approach, the structural optimization is developed to prolong fatigue life of the material.

1. Introduction

In recent years, turbochargers have increased in importance and have become indispensable to environmental protection and improved output, accompanied by further tightening of emissions standards. The success is reflected in the fact that almost all heavy trucks use turbo Charged diesel engines and this trend is continued into the small diesel for automotive application, particularly in China where their popularity has boomed due to low fuel consumption, durability and low emissions. The primary design objective of a turbocharger is wider mass flow range at a relatively moderate pressure ratio with emphasis on compactness, low cost and good acceleration characteristics to keep size, weight and inertia small (Hajipour et al, 2014). A turbocharger is a device in which heat energy from engine exhaust gas turns a radial inflow turbine along with a centrifugal compressor on the same shaft and supported by two inboard mounted fully floating shaft sleeve, such that inflow air is pressurized by the compressor and supplied to the engine, thereby improving the engine’s combustion efficiency.

The turbocharger is the use of exhaust emissions from engine to drive a turbine to rotate, so as to drive the impeller rotating air end, will be more air to the cylinders, achieve the goal of pressurization, and has the advantages of energy conservation and emissions reduction and increase engine power. Is the most critical of the turbocharger rotor of the turbocharger parts, belongs to the typical double cantilever rotor structure, complex vibration characteristics. Because of fuel combustion engine produces in the process of colloid and resin dope, uneven deposition on the turbine blades and create a dynamic imbalance, makes the turbocharger vibration under high-speed increase and destroy the parts fit clearance (Nama et al, 2015), scratch and damage to the bearing surface, which resulted in increased engine power loss, even cause the malignant accident such as destruction of the turbocharger, seriously affect the overall comprehensive performance and quality. Therefore we need to analyse the performance of automobile gasoline mechanical turbocharger based on finite element analysis.

2. Overview

The vehicle turbocharging technology requires high pressure ratio, high speed and miniaturization. Therefore, demands for the structural strength and reliability of rotor system are increasing, and dynamic analysis of the bearing rotor system becomes more important. Rotor dynamics mainly study vibration, balance and stability problems of rotor-bearing system in the rotating state. Turbocharger is high-speed rotating machinery. If the

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rotor shaft works at a critical speed for a long time, rotor will suffer various degrees of damages such as the normal gap between impeller and the shell, oil seal and bearing, When these damages happen, the turbocharger can't work normally and will cause the rotor shaft's fracture in severe cases. To ensure safety and reliability of turbocharger, the critical speed of the rotor shaft must be determined and moved from the turbocharger working speed range during design. Balance of the rotor is also an important part to ensure reliable operation of the turbocharger. The higher the turbocharger speed is the higher accuracy the rotor balancing requires. For this reason, the characteristics of unbalance response of the rotor need to be analysed (Varasteh and Goudarzi, 2015).

With the widely application of finite element analysis and structural optimization can shorten the period of production development efficiently, reduce the cost of production manufacture and improve the quality and reliability of the product. The structural of frame for 2500 turbocharger is investigated in this thesis. In view of the properties of the materials and static and fatigue features, taking finite element analysis software ANSYS as the analysis tool, the analysis and optimization of the structural frame are researched, taking fatigue analysis software MSC fatigue as the fatigue analysis tool . It also provides basis and direction for the design and trial of the product. Finite element models primarily based on the solid element and finite element technology are set up with ANSYS Parametric Design Language, and the modelling is simplified necessarily for calculating more convenience. From the static analysis of the frame, the stress and displacement that vary with position are gained and the basis is provided for the next product design and trial. From the results of fatigue analysis and influence of super high pressure, by selecting parts of geometry size as the design variables, in terms of meeting certain intension and toughness demands, the frame volume is selected as objective variable, and by adopting zero rank approach, the structural optimization is developed to prolong fatigue life of the material. With the finite element analysis, the results have a great significance to improve the design quality and benefit. So it has practical application value.

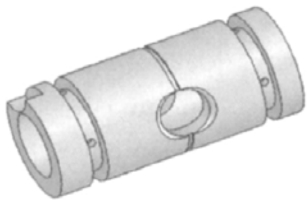
Due to manufacturing errors, the rotor mass centre of all micro-segments generally has a small rotating axis deviation (Quddus and Washington, 2015). When the rotor spins, centrifugal force caused by the above-mentioned deviation from the rotor will produce lateral vibration. The speeds causing extremely strong vibration are called critical speeds. To keep the machine from resonance within the scope of working speed, critical speed should properly deviate from the working speed, for example, over 10%.The critical speed is function of the rotor's flexibility and mass distribution. For a finite number of concentrated mass of discrete rotation system, the number of critical speed is equal to the number of concentrated mass (Pulido et al, 2014; Machuc and Mandow, 2015; Nejadet al, 2016).

For the elastic rotation system which has a continuous distribution of quality, there are an infinite number of critical speeds. During the design and operation of the rotor, it's necessary to know the vibration strength caused by unbalance and other motivating factors within the scope of work speed, and use it as a measure of evaluating the rotor working conditions. To calculate this problem, the critical speed algorithms are often used.

3. Semi-floating bearing structural analysis

Semi-floating bearing can be considered as a plain journal bearing in series with a squeeze film damper. The structure is shown in the figure 1. In order to be effective for it, the inner film must be very stiff to transfer shaft motion to the bearing ring. When this happens, the squeeze film damper becomes effective. The squeeze film damper offers the advantage of nearly zero cross coupling, which is destabilizing. However, the inner film must be stiffened by having a tight clearance, which in turn leads to more power dissipation and heating of the oil film. It is divided into two sub-type ordinary cylindrical sliding bearing according to the effective size during calculation.

Dynamic characteristics factors of bearings are obtained in pre-calculated process and a data file is generated, which are then transferred into final calculation software. At present in small cars, gasoline engine turbocharger commonly uses semi-floating bearings. Fig. 2 shows a rotor-bearing system designed for a small vehicle gasoline turbocharger which includes turbine rotor, compressor impeller, floating bearing, seal kit, thrust sets, and compressor impeller lock nut. In this paper, we establish three-dimensional model of these various components, determine the centroids' data and determine their location on the shaft.



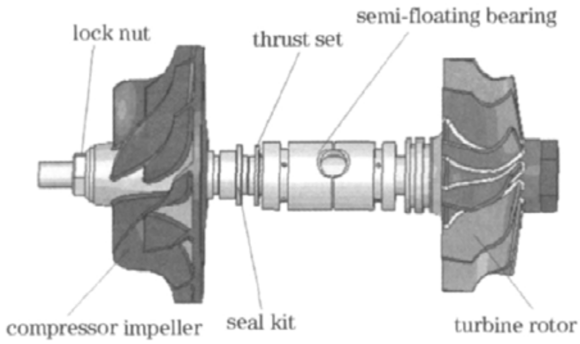


Figure 1: The structure *Figure 2: The rotor-bearing system*

4. Model and algorithm

Using accurate modelling parameters, the turbocharger rotor dynamics model is established (Fig. 3). The physical and mechanical properties of materials of the various component parts of the rotor system were determined. Compressor impeller and turbine impeller have irregular shapes with their mass and moment of inertia concentrated in the centre of gravity at the shaft. Due to that the compressor impeller uses small interference fit mounted on the shaft while the turbo-impeller welded to the shaft; they will affect the deformation of the shaft. Therefore the material elastic modulus and shear modulus are inputted according to the actuality.

In the model, the rotor, seal kits, thrust sets and lock nut are regular rotary bodies and the actual sizes are used as model inputs. Because compressor impeller and turbine impeller are irregular bodies, we put their mass and moment of inertia on the centre of gravity on the shaft. The numbers in the model are nodes which are divided basing on principles of the finite element method. These nodes are usually chosen on positions such as the focus of the impeller, shaft cross-section of any sudden changes, corresponding positions where sealing kits are installed on the shaft, and the shaft end.

The dynamic loads of bearings mainly have three sources: imbalance of rotating assembly, the internal stress caused by the movement of the engine and rotary inertia force of operating organizations. The generation of static load is a common effect of the aerodynamic marginal loads and gear loads of turbine complex. In general, the unbalanced force is the largest of the three as single action. Compressor and turbine impeller are individual balance. So when they are assembled and re-assembled, they won't exceed the limits required for centrifugal force while full assembled, which is in particularly important when using the internal bearings. Balance calculations can be used to meet the largest eccentric moments and acceptable working life that depends on bearing design and planning cycles.

The small vehicle gasoline turbocharger calculated in this paper uses 15 W/40 oil with the low temperature viscosity of 3500 mPa·s at -15°C and the movement viscosity of 12.5-16.3 mm^2/s at 100°C. According to Ref. [10], the oil temperature of turbocharger exit is generally below 120°C. In this paper, we assume the bearing lubricant temperature is 100°C.

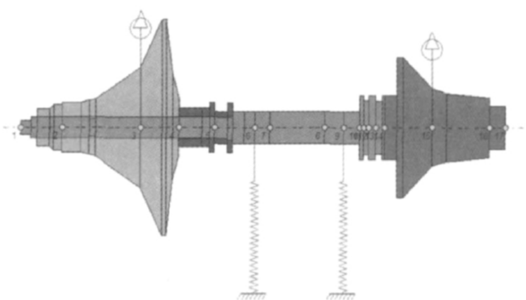


Figure 3: Turbocharger model built in the DyRoBeS software

The basic equation of key algorithm is shown as the equation (1):

$$(N, sk) \leftarrow Key(1^k) \tag{1}$$

This formula is used to generate file checksum parameter which is denoted by:

$$r \leftarrow \{0,1\}^k; sk \leftarrow \{e, d, r\}; \\ Output\{N, sk\}; \tag{2}$$

The Euler function is:

$$\phi(N) = (p-1)(q-1) \tag{3}$$

Then choose an integer e to satisfy the following equation 4:

$$\begin{cases} 1 < e < \phi(N) \\ \gcd(e, \phi(N)) = 1 \end{cases} \tag{4}$$

Then finally export (N, sk) in Tag algorithm, we can get the optimization equation (5):

$$(T_0, T_2, ...T_{n-1}) \leftarrow Tag(pk, sk, m) \tag{5}$$

The formula generates labels for each file block.

$$for(j = 0; j \le n-1; j++); \tag{6}$$

$$\{W_j = r*(j+1); T_i \\ = [h(W_j)*m_j]^c \bmod N\}; \tag{7}$$

$$Output(T_0, T_2, ...T_{n-1}); \tag{8}$$

And local fractional integral of f(x) defined by Eq.9.

$$_aI_b^{(\alpha)} f(t) = \frac{1}{\Gamma(1+\alpha)} \int_a^b f(t)(dt)^{\alpha} \\ = \frac{1}{\Gamma(1+\alpha)} \lim_{\Delta t \to 0} \sum_{j=0}^{j=N-1} f(t_j)(\Delta t_j)^{\alpha} \tag{9}$$

Its local fractional Hilbert transform, denoted by $f_{x}^{H,\alpha}$ is defined by

$$H_{\alpha}\{f(t)\} = \hat{f}_H^{\alpha}(x) \\ = \frac{1}{\Gamma(1+\alpha)} \oint_R \frac{f(t)}{(t-x)^{\alpha}} (dt)^{\alpha} \tag{10}$$

Where x is real and the integral is treated as a Canchy principal value, that is,

$$\frac{1}{\Gamma(1+\alpha)} \oint_R \frac{f(t)}{(t-x)^{\alpha}} (dt)^{\alpha} \\ = \lim_{\varepsilon \to 0} [\frac{1}{\Gamma(1+\alpha)} \int_{-\infty}^{x-\varepsilon} \frac{f(t)}{(t-x)^{\alpha}} (dt)^{\alpha} + \\ \frac{1}{\Gamma(1+\alpha)} \int_{x+\varepsilon}^{\infty} \frac{f(t)}{(t-x)^{\alpha}} (dt)^{\alpha}] \tag{11}$$

To obtain the inverse local fractional Hilbert transform, write again Eq. (11) as

$$\hat{f}_H^{\alpha}(x) = \frac{1}{\Gamma(1+\alpha)} \int_{-\infty}^{\infty} \frac{f(t)}{(t-x)^{\alpha}} (dt)^{\alpha} \\ = \frac{1}{\Gamma(1+\alpha)} \int_{-\infty}^{\infty} f(t) g(x-t)(dt)^{\alpha} \\ = f(x) * g(x), \tag{12}$$

The equation of motion is as follows:

$$\partial_j (C_{ijkl} \partial_k u_l + e_{kij} \partial_k \varphi) - \rho \ddot{u}_i = 0 \tag{13}$$

Under the linear theory, that is:

$$\partial_j (e_{ijkl}\partial_k u_l - \eta_{kij}\partial_k \varphi) = 0 \tag{14}$$

The experiment and result discussion

In accordance with the actual working conditions of the turbocharger, we concretely analyze third-order critical speed and mode of vibration of the rotor when oil temperature is 100°C and bearing gap is 0.011 mm. The first order critical speed, the mode of the rotor and the corresponding energy distribution is shown in Fig. 4. From Fig. 4, the first order critical speed is 24 834 r/min in the vibration mode curves, the vibration mode is a rigid-body diagram cone model, and the compressor impeller and turbine impeller have out of-phase movements. From the energy distribution curves, the turbine-side has the highest energy level because the turbine impeller is very heavy (more than half of the quality of the entire rotor). Therefore, the nature of bearing at turbine-side has great influence on the first order critical speed of the rotor. Figure 5 shows the FEM result. The second-order critical speed and energy distribution is a flexible shaft bending model. Fig. 6 shows that the critical speed of the turbocharger rotor is 54 318 r/min. This mode is mixed mode with the shaft bending energy accounting for 26.83% of the total energy and the compressor impeller and turbine impeller having the same-phase movement. Critical speed map is used to check the design of the shaft and the bearings from a stiffness point of view. The critical speed for a range of bearing stiffness is calculated, assuming that the bearings are connected to a rigid foundation.

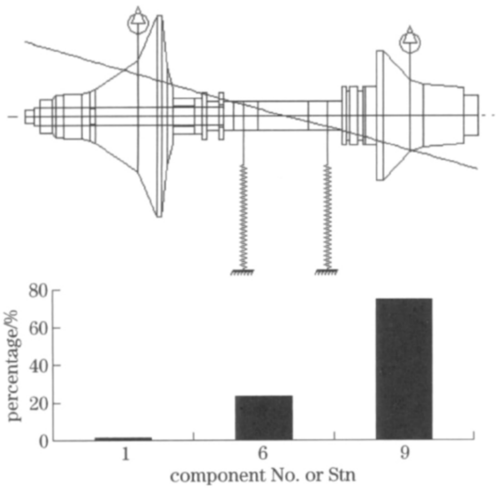


Figure 4: The first-order critical speed and the energydistribution

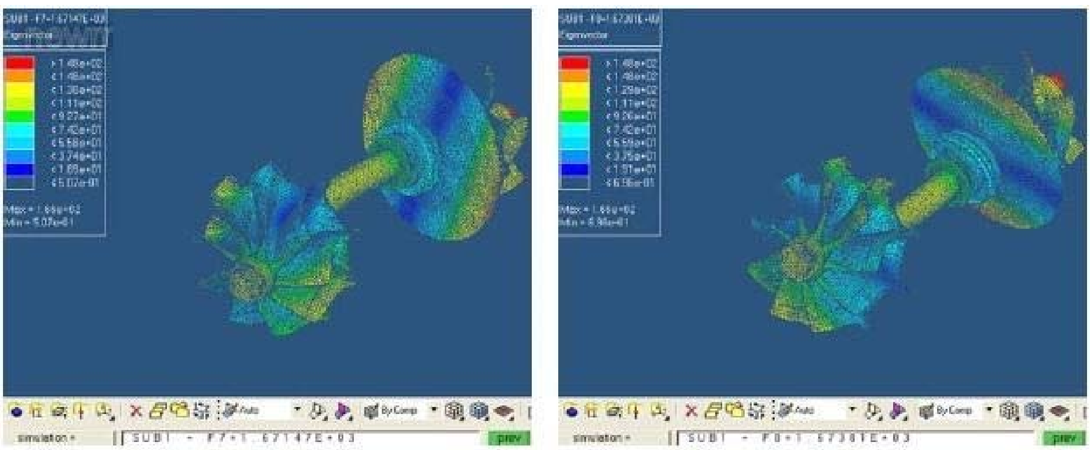


Figure 5: The FEM result

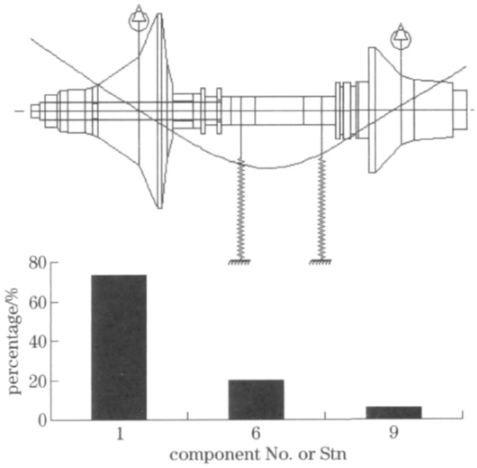


Figure 6: The second-order critical speed and the energydistribution

5. Conclusions

Through the analysis of the semi-floating bearing structure, rotor dynamic analysis of small vehicle gasoline engine turbocharger have been done, including the calculation and analysis of critical speed and unbalance response. The conclusions are as follows.

The first three order critical speeds respectively are 24 857 r/min, 54 346 r/min and 299 563 r/min. Vehicle turbocharger work speed ranges from 70 000 to 180 000 r/min. The flexibility rotor work speeds are normally between the seas and the third critical speed. In the actual work, the first critical speed can be easily obtained because the model of the first order critical speed is rigid and the shaft bending energy only accounts for 1.7% of the total energy. Therefore the response of the shaft is very small. It's not obvious in the test. The first critical speed doesn't have so much impact on the turbocharger. The second-order critical speed is most noticeable important, it must be away from the work speed when design the rotor system. The third-order critical speed is always much higher than the work speed range. Our calculation results approve this rule. And the theoretical predictions match the experimental data.

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