Design Torsional Vibration Damper of Engine based on Classical Optimal Approach

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Abstract. The torsional vibration of crankshaft is an important factor in the vibration caused by the engine and causing fracture. An inline 4-cylinder engine crankshaft system will be the object to research. Finite element model of crankshaft is built by ANSYS. And analysis torsional vibration characteristics of shafting are based on the model, which include natural vibration and forced vibration. It gets main harmonic times, minor main harmonic times and corresponding engine speed in commonly used speed range. Designs parameters of torsional vibration damper (TVD) based on classical optimal approach. And tests the damping effect under different conditions based on the finite element model. The results show that the design meets the requirements of the damper, and has good damping effect. Research methods and conclusions can be applied to actual research. It can be used in similar crankshaft torsional vibration system analysis, design and analysis of torsional damper.

Key words: Engine; Crankshaft; Torsional Vibration; Torsional Vibration Damper; Finite Element Method; Classical Optimal Approach.

1. Introduction

The engine is not only the power source of the whole vehicle, but also the vibration excitation source of the whole vehicle [1], while the crankshaft torsional vibration is the most important component of the whole engine excitation source, and its reliability directly affects the reliability of the engine and the whole vehicle [2]. From the perspective of safety, even if resonance occurs, it is desirable that the resonance point is outside the rotational speed range; if resonance occurs in the rotational speed range, it is desirable that the resonance amplitude is small and the stress at this time is also small. Therefore, it is necessary to analyze and study the torsional vibration characteristics of the crankshaft system, and to explore the reduction method and control measures of torsional amplitude [3].

Taking the crankshaft shafting system of a 30T engineering vehicle in-line four-cylinder engine as the research object, the ANSYS finite element model of the crankshaft system is established, based on inherent characteristics of model analysis of shaft torsional vibration and it’s forced vibration response. The parameters of torsional vibration damper are designed based on the optimal classical design method. Exploring the effect of torsional vibration of the torsional vibration of the torsional vibration damper under different working conditions by the harmonic response analysis which is based on crankshaft shafting finite element model with or without torsional vibration damper. It provides a reference method...
for the study of torsional vibration characteristics of crankshaft shafting of similar engines and the design of damper.

2. Analysis of Torsional Vibration Characteristics of Crankshaft Shafting

2.1. Analysis of inherent characteristics

The three-dimensional solid model of the crankshaft shaft of the hub, crankshaft, flywheel and piston linkage mechanism is established. Based on the three-dimensional model, the ANSYS finite element model of the crankshaft shaft system is established, as shown in Fig. 1; the first two natural frequencies are obtained, as shown in Table 1.

![Finite Element Model of Crankshaft](image)

**Fig.1** Finite Element Model of Crankshaft

| degree | numerical value |
|--------|-----------------|
| 1degree| 483 Hz          |
| 2degree| 1195 Hz         |

2.2. Resonance degrees

When the circular frequency $\omega_0$ of a certain harmonic of the engine excitation torque is equal to a certain order natural circular frequency $p_i$ of the crankshaft axis, resonance will occur. The natural frequency [4] is expressed by the number of free vibration cycles per minute $N_i$, at this time:

$$N_i = \frac{60}{\pi} \frac{60 p_i}{2\pi} = \frac{30 p_i}{\pi}$$

(1)

If the crankshaft speed is $n_c$ and the angular velocity is $\omega_c$ when resonance occurs, then [5]:

$$\omega_0 = \omega_c = \frac{\pi n_c}{30} = p_i, \quad n_r = \frac{1}{v} \frac{30 p_i}{\pi} = \frac{1}{v} N_i$$

(2)
There are infinite theoretical speeds in accordance with equation (2), but there are only a limited number of engine speed ranges. The number of simple harmonics that can cause resonance between the idle speed \( n_{\text{min}} \) and the nominal maximum speed \( n_{\text{max}} \):

\[
\frac{N_i}{n_{\text{min}}} \geq \nu \geq \frac{N_i}{n_{\text{max}}}
\]  

(3)

When the abscissa is \( n \) and the ordinate is \( v\cdot n \), the ray of different \( v \) times is drawn, and the abscissa value corresponding to the intersection of the ray and \( v\cdot n = N_i \) is the \( i \)-order \( v \)-resonance speed of the engine.\(^c\)\textit{nc} [6]. The engine speed range studied is 750 r/min-6000 r/min, and the main harmonic and sub-primary harmonic resonance speeds are shown in Fig. 2.

As can be seen from Fig. 2, in the range of the rotational speed, the 1, 2, 3, and 4 harmonic rays have no intersection with the first-order natural circular frequency, and the rays within the 12 harmonics have no intersection with the second-order natural circular frequency. Therefore, only the harmonic main harmonics of 6, 8, 10, and 12 and the harmonic secondary harmonics of 5, 7, 9, and 11 are considered. These harmonics cause the first-order resonance of the engine at a specific speed. Since the amplitude of the high harmonic excitation torque is relatively small, the main harmonic and the second main harmonic in the common rotational speed range of the engine are 6 harmonics and 5 harmonics respectively.

3. Torsional Damper Design

3.1. Optimal Classical Design Theory

The mechanical equipment and the torsional vibration damper form a two-degree-of-freedom system, as shown in Figure 3.

The mechanical equipment is the main mass \( m_1 \), and the damper is the sub-mass \( m_2 \). According to the vibration theory, the differential equation as follows [7]:

\[
m_1\ddot{x}_1 + k_1 x_1 + k_2 (x_1 - x_2) + c_2 (\dot{x}_1 - \dot{x}_2) = F_v \sin \omega t
\]  

(4)

\[
m_2\ddot{x}_2 + k_2 (x_2 - x_1) + c_1 (\dot{x}_2 - \dot{x}_1) = 0
\]  

(5)
Let the general solution of the second-order differential equation be \( x_j(t) = X_j e^{i\omega t} \) \((j=1,2)\), and bring back the equations (4) and (5) to obtain the steady-state amplitude:

\[
X_1 = \frac{F_0 ( k_2 - m_2 \omega^2 + i c_2 \omega )}{\left[(k_1 - m_1 \omega^2)(k_2 - m_2 \omega^2 - m_2 \omega^2)\right] + \left[i oc_2(k_1 - m_1 \omega^2 - m_1 \omega^2)\right]} \tag{6}
\]

\[
X_2 = \frac{X_1 (k_2 + ic_2 \omega)}{(k_2 - m_2 \omega^2 + ioc_2)} \tag{7}
\]

**Fig. 3** Schematic Diagram of Torsional Vibration Damper

Study the magnitude of the amplitude. Introducing relative amplitude \( A_1(\varphi) = \frac{X_1}{\delta_1} \), Simultaneous introduction of mass ratio \( \mu = \frac{m_2}{m_1} \), Tuning ratio \( f = \frac{\omega_0}{\omega_2} \), damping ratio \( \xi_2 = \frac{c_2}{\omega_2} \), Frequency Ratio \( g = \frac{\omega_2}{\omega_n} \).

In the equation: \( c_2 = 2m_1 \omega_2 \) - critical damping coefficient, N·m·s/rad; \( \omega_2 = \sqrt{\frac{k_2}{m_2}} \) - Natural frequency of the secondary system, rad/s; \( \omega_n = \sqrt{\frac{k_1}{m_1}} \) - Natural frequency of the main system, rad/s; \( \delta_1 = \frac{F_0}{K_1} \) - Static deformation, m; F0- Maximum amplitude of exciting force, N. (6), (7) Divided by respectively \( \frac{F_0}{K_1} \) is relative amplitude expression:

\[
A_1(\varphi) = \frac{(2j\xi_2) + (g^2 - f^2)}{\sqrt{(2j\xi_2)(g^2 - 1 + \mu g^2) + [\mu f^2 g^2 - (g^2 - 1)(g^2 - f^2)]}} \tag{8}
\]

\[
A_1(\varphi) = \frac{(2j\xi_2) + f^2}{\sqrt{(2j\xi_2)(g^2 - 1 + \mu g^2) + [\mu f^2 g^2 - (g^2 - 1)(g^2 - f^2)]}} \tag{9}
\]
It can be seen from the formula that the relative amplitude is determined by the frequency ratio, the mass ratio, the tuning ratio, and the damping ratio. Using the control variable method [8], let the mass ratio be 0.05 and the tuning ratio be 1 to plot the response curves of the main mass relative amplitudes with different damping ratios. As shown in Fig. 4, two intersection points A and B are obtained.

When the amplitudes of A and B are equal, the damping effect is best [9], now the formula needs be figured out.

\[ f_{opt} = \frac{1}{1 + \mu} \]  

(10)

A and B are the maximum points of the amplitude, and the derivative of equation (8) is equal to 0.

\[ \xi_A^2 = \frac{3 - \sqrt{\mu}}{8(1 + \mu)^3}, \quad \xi_B^2 = \frac{3 + \sqrt{\mu}}{8(1 + \mu)^3} \]  

(11)

When designing the damper damping, take the average of (11):

\[ \xi_{opt}^2 = \frac{3\mu}{8(1 + \mu)^3} \]  

(12)

The best mass ratio \( \mu \), optimal tuning ratio \( f_{opt} \), and optimal damping ratio \( \xi_{opt} \) of the torsional vibration damper are obtained. At this time, the frequency ratio of A and B is:

\[ f_A = \frac{2 + \mu + \sqrt{(2 + \mu)\mu}}{(1 + \mu)(2 + \mu)}, \quad f_B = \frac{2 + \mu - \sqrt{(2 + \mu)\mu}}{(1 + \mu)(2 + \mu)} \]

The relative amplitude of the main system is \( A_A(g) = A_B(g) = \sqrt{1 + \frac{2}{\mu}} \).

**Fig. 4 Relative Amplitude Curve of Main Quality Under Different Damping Ratio**

### 3.2. Damper Parameter Design

The optimal classical design is derived based on the single-degree-of-freedom equivalent torsion model of the crankshaft shafting system. The shafting system is converted into a single-degree-of-freedom equivalent torsional pendulum system, and the double-torch equivalent system is formed with the shock absorber [11], as shown in Figure 5.
Fig. 5 Double Torsion Equivalent System of Crankshaft with TVD

I’1 - Crankshaft shafting single degree of freedom system moment of inertia; ID- Moment of inertia of the damper inertia ring; KD- Damper stiffness; CD- Damper damping

Tab.2 Parameters of Torsional Vibration Damper (Moment of inertiaI / e-3 kg·m², frequencyf / Hz, StiffnessK / N·m·rad⁻¹, DampingC / N·m·s·rad⁻¹)

| Parameter                  | Value Parameter | Parameter Value | Parameter Value |
|----------------------------|-----------------|-----------------|-----------------|
| CrankshaftI₁              | 21.986          | Mass ratio u    | 0.1537          |
| Main qualityf₁            | 483             | Frequency ratio f | 0.8668         |
| Critical dampingCₑ        | 23.661          | Damping ratio ξ | 0.1937          |
| Shock absorberI₀          | 3.379           | Damper dampingC₀ | 4.5839          |
| Shock absorberf₀          | 557             | Damper stiffnessK₀ | 41421           |
| Resonance frequencyfₐ     | 384.96          | Resonance frequencyf₉ | 506.19          |

The relative amplitude of the first-order equivalent pendulum of the crankshaft is shown in Fig. 6. The resonant excitation frequencies of the two resonance peaks A and B are 384.96 Hz and 506.19 Hz, respectively. The relative amplitudes of the equivalent axes of the crankshaft are equal at 3.7435, which is consistent with the reduction. The optimal design theory of the vibrator.

Fig. 6 Frequency Response of 1st Order Equivalent Pendulum with TVD

4. Finite element analysis of torsion vibration of crankshaft shafting

4.1. Harmonic response analysis of shafting without damper

Based on the 1 cylinder, the initial phase is 0, and the other cylinders are loaded according to the phase difference. The harmonic response analysis has a main harmonic order of 6, and a sub-primary harmonic
of 5. Load finite element model, the vibration torque excitation is loaded at the center point on the central interface of crank journal. The excitation frequency ranges from 340 Hz to 580 Hz, with 24 load steps and 10 Hz step length. The harmonic response analysis under 5 and 6 orders vibration torque excitation is carried out, and the frequency response curve of rotation angle is obtained, as shown in Figure 7 (a). In order to obtain the rotation speed and the crankshaft front rotation angle curve, the excitation frequency is converted into the rotation speed, as shown in Fig. 7(b):

As can be seen from Fig. 7(a), the shafting is about 477 Hz under the action of the 5 and 6 order vibration torque excitations, that is, resonance occurs near the first-order natural frequency. The peak amplitude of the front of the crankshaft under the order of 6 is 0.17494°, and 0.06549° of 5. The 6 order is the main harmonic order. Although the amplitude is smaller than the 5 order, the resonance amplitude is much larger than the 5 order, reaching 0.17494°. This resonance amplitude has exceeded the allowable torsion amplitude (the allowable torsion amplitude of the crankshaft front is 0.1 degrees), and a damper is a must to reduce its resonance amplitude. It can be seen from Fig. 7(b) that the resonances caused by the 6 order and the 5 order occur at 4600 r/min and 5520 r/min respectively, and the resonance amplitude of the 6-order occurring at the engine speed of 4600 r/min is large. This speed is about a common operating speed for the engine, so dampers are needed to improve the response of the engine shafting.

![Frequency response curve](image1)

![Curve with rotation speed](image2)

**Fig.7** Torsion Angle Curve of Crankshaft Front Without TVD

### 4.2. Harmonic response analysis of shafting with damper

Based on the 1 cylinder, the initial phase is 0. The harmonic response analysis has a main harmonic order of 6, and a sub-primary harmonic of 5. Load finite element model, the vibration torque excitation is loaded at the center point on the central interface of crank journal. The first two natural frequencies of the finite element model of the damper crankshaft are 342 Hz and 565 Hz, respectively, and the load step is set accordingly. Set different load steps, view in Table 3. According to this load step, a total of 71 substeps are set. The method performs a harmonic response analysis operation with a step size of 1 Hz at about 342 Hz and 365 Hz, and performs a harmonic response analysis operation with a step size of 10 Hz in other ranges.

| Substep | Range (Hz) | Step size (Hz) | Steps |
|---------|------------|----------------|-------|
| Substep 1 | 250–330 | 10             | 8     |
| Substep 2 | 330–350 | 1              | 20    |
| Substep 3 | 350–550 | 10             | 10    |
| Substep 4 | 550–570 | 1              | 20    |
| Substep 5 | 570–700 | 10             | 13    |

According to the above settings, the harmonic response analysis under 5 and 6 orders vibration torque
excitation is carried out, and the frequency response curve of rotation angle is obtained, as shown in Figure 8 (a). In order to obtain the rotation speed and the crankshaft front rotation angle curve, the excitation frequency is converted into the rotation speed, as shown in Fig. 8(b).

It can be seen from Fig. 8(a) that under the action of 5 and 6 order vibration torque excitation, resonance occurs at excitation frequencies of 346 Hz and 370 Hz, that is, the first two-order torsion natural frequencies of the damper with damping effect and the crankshaft shafting. Due to the damper, the frequency response of the front of the crankshaft changes from a single peak to a double peak, which means frequency modulation. In the figure, the two amplitude peaks of the front of the crankshaft under the action of 5 order vibration torque excitation are 0.0252°, 0.0281°, and the maximum amplitude is 0.0281°. Compared with no TVD, the maximum displacement of the front is 0.06549°, which is reduced by 57.093%. In Figure 8(b), the peak amplitudes of the front under 6 order are 0.0606° and 0.0855°, and the maximum amplitude is 0.0855°, which is reduced to less than 0.1°. It meets the international allowable standard, and the maximum front displacement of 0.17494° is reduced by 51.123% compared to no TVD. It can be seen from the figure that the resonance caused by the 6 main order occurs at 3400 r/min and 5700 r/min respectively, and the resonance caused by the 5 main order occurs at 4080 r/min, and the other peak exceeds the engine speed range.

Therefore, the designed damper reduces the amplitude of the front of the crankshaft shafting by 51.123% and 57.093% under the action of 6 main order and 5 main order vibration torque excitations respectively. The torsion vibration of the shafting is greatly reduced, that is, the damper plays a good role in reducing the amplitude.
5. Conclusion
The torsion characteristics of the crankshaft shafting of a four-cylinder engine for a construction vehicle are analyzed comprehensively. Based on the simplified ANSYS finite element model of crankshaft system, the torsion vibration characteristics of the crankshaft shafting are analyzed and the inherent characteristics are obtained. According to the first two torsion natural frequencies of the crankshaft shafting, the primary and sub-primary orders in the range of engine speed are determined. The influence of the parameters of the damper on its performance is discussed, and the parameters of the damper are designed based on the optimal classical design method. Do harmonic response analysis based on the finite element model of crankshaft system with and without dampers, the results show that the damper has excellent damping effect and can reduce the maximum amplitude of 5 and 6 order torsion vibration which affects crankshaft torsion vibration most by more than 51%. The results show that it is feasible to design the torsion damper based on the optimal classical design theory, which provides reference for the torsion vibration analysis of engine crankshaft system, the design of torsion vibration damper and the analysis of damping effect.

Acknowledgments
Fund from science and technology research project of Chongqing in 2017 "Investigation and Research on Automobile Consumption Demand in Chongqing in the New Era" (No.: KJ1729412);
Fund from Professional capacity building project (key majors) of Chongqing higher vocational college. (No.:120).

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