

Analysis of Performance Parameters of Torsional Vibration Damper Under Various Operating Conditions

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ABSTRACT

The performance parameters of torsional vibration damper, including stiffness and damping, have great influence on the torsional vibration of automobile driveline. At present, the research on torsional vibration damper mainly concentrates on the torsional stiffness, but rarely on the torsional damping characteristics. This paper systematically studied the effect of torsional stiffness and damping on torsional vibration of automobile driveline under uniform speed conditions, accelerated and decelerated conditions, idling conditions and resonance conditions. The requirements on stiffness and damping of various operating conditions were summarized. The effect and requirements researched were useful to performance match design of torsional vibration damper.

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INTRODUCTION

Torsional vibration of transmission system is an important vibration mode of vehicles. It couples with vehicle longitudinal vibration, transmission bending vibration and drive axle suspension distorted vibration. Therefore, a complex problem of vibration and noise comes out, which has direct impact on vehicle ride comfort and the durability of the parts [1][2][3][4]. The transmission torsional vibration reduces significantly using suitable torsional damper.

At present, the majority of literature basically aims at the selection and research of the torsional stiffness of torsional damper, but few aims at research of damping characteristic [5][6].

A multi-degree-of-freedom torsional vibration model of driving system was established in this paper. The effects of stiffness and damping characteristics of torsional damper on torsional vibration during condition of constant speed, acceleration, deceleration, idling, resonance were researched. The requirements of stiffness and damping characteristic of torsional damper during different condition were summarized. Theoretical support for the design of the torsional damper was provided.

MULTI DOFS MODEL OF TRANSMISSION SYSTEM

The actual structure of transmission system is very complex, and it is difficult to calculate the rotational inertia,

torsional stiffness and damping characteristic directly. Therefore, it is necessary to simplify the actual transmission system. The simplified model was a discrete model with limited DOFs that contains inertial sensors, elastic element and damping element.

According to the simplifying principle [7][8], the equivalent model of transmission system was built. Figure 1 shows the 7 DOFs torsional vibration model of transmission system under idling condition; Figure 2 shows the 12 DOFs torsional vibration model under driving condition.

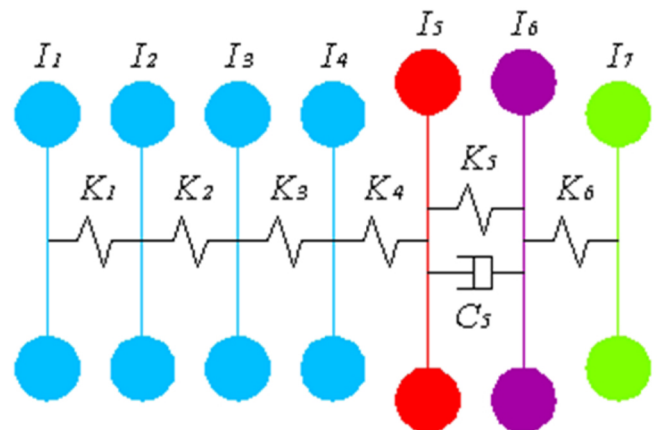


Figure 1. Torsional vibration model of transmission system under idling condition

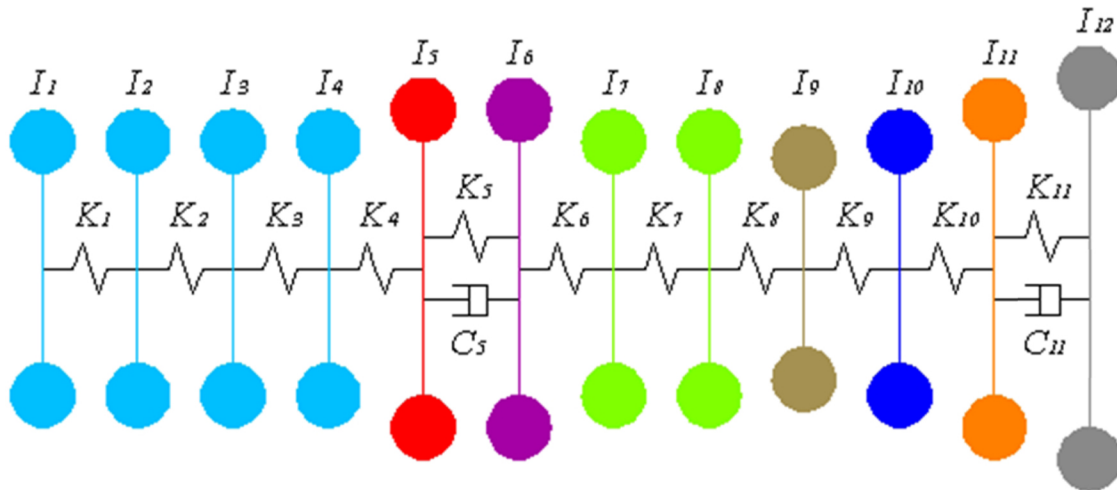


Figure 2. Torsional vibration model of transmission system under driving condition

In Figure 1 and Figure 2, I_i ($i=1\sim 12$) is the rotational inertia of each inertial element in the transmission system ($\text{kg} \cdot \text{m}^2$), K_j ($j=1\sim 11$) is the torsional stiffness of each elastic element ($\text{kN} \cdot \text{m/rad}$), C_k ($k=5, 11$) is torsional damping of each damping element ($\text{kN} \cdot \text{m} \cdot \text{s/rad}$). The specific parameters are shown as below.

I_1 - rotational inertia of the crank-connecting rod of engine first cylinder;

I_2 - rotational inertia of the crank-connecting rod of engine second cylinder;

I_3 - rotational inertia of the crank-connecting rod of engine third cylinder;

I_4 - rotational inertia of the crank-connecting rod of engine fourth cylinder;

I_5 - rotational inertia of active part of dual mass flywheel;

I_6 - rotational inertia of driven part of dual mass flywheel;

I_7 - rotational inertia of the gearbox input shaft and constant mesh gear;

I_8 - rotational inertia of the gearbox output shaft and constant mesh gear;

I_9 - rotational inertia of transmission shaft;

I_{10} - rotational inertia of main reducing gear and differential mechanism;

I_{11} - rotational inertia of half shaft and driving wheel;

I_{12} - equivalent rotational inertia of body translation quality;

K_1 - torsional stiffness of crank axle of engine first cylinder;

K_2 - torsional stiffness of crank axle of engine second cylinder;

K_3 - torsional stiffness of crank axle of engine third cylinder;

K_4 - torsional stiffness of crank axle of engine fourth cylinder;

K_5 - torsional stiffness of the torsional damper of double mass flywheel;

K_6 - torsional stiffness of the gearbox input shaft;

K_7 - torsional stiffness of the gearbox intermediate shaft;

K_8 - torsional stiffness of the gearbox output shaft;

K_9 - torsional stiffness of transmission shaft;

K_{10} - torsional stiffness of half shaft;

K_{11} - torsional stiffness of the driving wheel;

C_5 - torsional damping of the torsional damper of double mass flywheel;

C_{11} - torsional damping of the driving wheel.

According to Figure 1, oscillatory differential equations of transmission system equivalent model under idle speed were built as equation (1).

$$\begin{cases} J_1 \ddot{\theta}_1 + K_1(\theta_1 - \theta_2) = T_1 \\ J_2 \ddot{\theta}_2 - K_1(\theta_1 - \theta_2) + K_2(\theta_2 - \theta_3) = T_2 \\ J_3 \ddot{\theta}_3 - K_2(\theta_2 - \theta_3) + K_3(\theta_3 - \theta_4) = T_3 \\ J_4 \ddot{\theta}_4 - K_3(\theta_3 - \theta_4) + K_4(\theta_4 - \theta_5) = T_4 \\ J_5 \ddot{\theta}_5 - K_4(\theta_4 - \theta_5) + K_5(\theta_5 - \theta_6) + C_5(\dot{\theta}_5 - \dot{\theta}_6) = 0 \\ J_6 \ddot{\theta}_6 - K_5(\theta_5 - \theta_6) + K_6(\theta_6 - \theta_7) - C_5(\dot{\theta}_5 - \dot{\theta}_6) = 0 \\ J_7 \ddot{\theta}_7 - K_6(\theta_6 - \theta_7) = 0 \end{cases} \quad (1)$$

According to Figure 2, oscillatory differential equations of transmission system equivalent model under driving condition were built as equation (2).

Table 1. Natural frequency of transmission system under idling condition (Hz)

Gear selection	1st order	2nd order	3rd order	4th order	5th order	6th order
Natural frequency	30.4	370.4	500.3	1297	1967	2407

Table 2. Natural frequencies of transmission system under driving condition (Hz)

Gear selection	1st order	2nd order	3rd order	4th order	5th order	6th order
1st gear	2.41	18.68	36.89	166.89	363.57	1547
2nd gear	4.33	18.68	36.41	170.72	362.35	857.6
3rd gear	6.38	18.67	35.62	178.03	358.82	575.5
4th gear	7.92	18.65	34.87	186.75	350.77	462.3
5th gear	8.66	18.64	34.46	192.59	341.42	426.58

$$\begin{cases}
 J_1 \ddot{\theta}_1 + K_1(\theta_1 - \theta_2) = T_1 \\
 J_2 \ddot{\theta}_2 - K_1(\theta_1 - \theta_2) + K_2(\theta_2 - \theta_3) = T_2 \\
 J_3 \ddot{\theta}_3 - K_2(\theta_2 - \theta_3) + K_3(\theta_3 - \theta_4) = T_3 \\
 J_4 \ddot{\theta}_4 - K_3(\theta_3 - \theta_4) + K_4(\theta_4 - \theta_5) = T_4 \\
 J_5 \ddot{\theta}_5 - K_4(\theta_4 - \theta_5) + K_5(\theta_5 - \theta_6) + C_5(\dot{\theta}_5 - \dot{\theta}_6) = 0 \\
 J_6 \ddot{\theta}_6 - K_5(\theta_5 - \theta_6) + K_6(\theta_6 - \theta_7) - C_5(\dot{\theta}_5 - \dot{\theta}_6) = 0 \\
 J_7 \ddot{\theta}_7 - K_6(\theta_6 - \theta_7) + K_7(\theta_7 - \theta_8) = 0 \\
 J_8 \ddot{\theta}_8 - K_7(\theta_7 - \theta_8) + K_8(\theta_8 - \theta_9) = 0 \\
 J_9 \ddot{\theta}_9 - K_8(\theta_8 - \theta_9) + K_9(\theta_9 - \theta_{10}) = 0 \\
 J_{10} \ddot{\theta}_{10} - K_9(\theta_9 - \theta_{10}) + K_{10}(\theta_{10} - \theta_{11}) = 0 \\
 J_{11} \ddot{\theta}_{11} - K_{10}(\theta_{10} - \theta_{11}) + K_{11}(\theta_{11} - \theta_{12}) + C_{11}(\dot{\theta}_{11} - \dot{\theta}_{12}) = 0 \\
 J_{12} \ddot{\theta}_{12} - K_{11}(\theta_{11} - \theta_{12}) - C_{11}(\dot{\theta}_{11} - \dot{\theta}_{12}) = T_f + T_w
 \end{cases} \quad (2)$$

In [equation \(1\)](#) and (2), T_1 , T_2 , T_3 and T_4 are the torques that the 1st, the 2nd, the 3rd and the 4th cylinder on crankshaft; T_f and T_w are equivalent torques of the rolling resistance and air resistance under driving condition. Dynamics [equations \(1\)](#) and (2) can be expressed as matrix [equation \(3\)](#).

$$[\mathbf{I}]\{\ddot{\theta}\} + [\mathbf{C}]\{\dot{\theta}\} + [\mathbf{K}]\{\theta\} = \{\mathbf{T}\} \quad (3)$$

RESONANCE ANALYSIS OF TRANSMISSION SYSTEM

It will produce resonance when the modal frequency of transmission torsional vibration system and excitation frequency come closer, and cause abnormal vibration and noise, which was transmitted through powertrain mount, shaft support, suspension to vehicle interior [9][10][11].

The rationality of the distribution of transmission inherent frequency could be evaluated by the inherent characteristics of transmission system, which provide reference for the torsional stiffness value of torsional damper. In order to make sure stiffness of torsional damper match with transmission

system, it's necessary to calculate the inherent frequency and vibration mode shape of transmission torsional vibration model.

When analyzing the inherent characteristics of transmission torsional vibration model, the damping matrix $[\mathbf{C}]$ and torque vector $\{\mathbf{T}\}$ could be ignored. Matrix [equation \(3\)](#) can be transferred to Matrix [equation \(4\)](#).

$$[\mathbf{I}]\{\ddot{\theta}\} + [\mathbf{K}]\{\theta\} = \{\mathbf{0}\} \quad (4)$$

The modal frequency and mode shape of transmission system during idling condition and driving condition could be got by solving [equation \(4\)](#). [Table 1](#) shows the first six order natural frequencies of transmission system during idling condition; [Table 2](#) shows the first six order natural frequencies of transmission system at 1st-5th gear position.

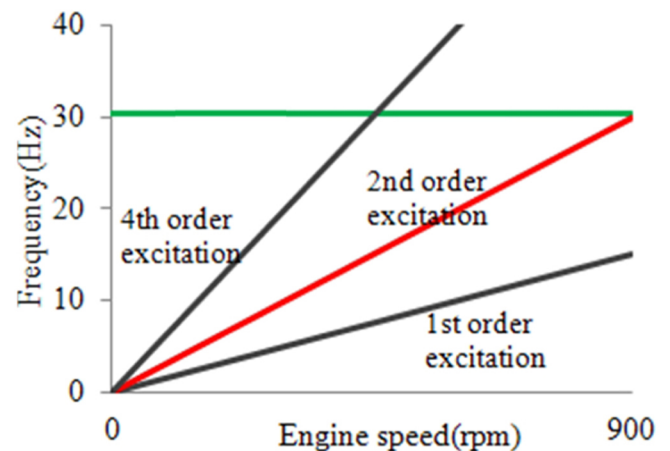


Figure 3. The mode tracking spectrum of transmission system during idling condition

[Figure 3](#) shows the mode tracking spectrum of transmission system during idling condition. Horizontal line represents the 1st order natural frequency of transmission system, and oblique lines represent each order harmonic excitation of engine. The engine's idling speed is 800rpm (up to 900rpm when air conditioner turned on). The 1st natural