Calculation Method of Working Rotational Angle of Automobile Transmission System and Its Applications

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Abstract. The working state and operating parameters of the automobile transmission system play a key role in the vehicle noise and vibration performance. Based on the basic calculation method of relative rotational angle, this paper proposes two methods for calculating the working rotational angle and torsional stiffness of the transmission system, which can effectively obtain the key information of the transmission system under the vehicle operating state. The working rotational angle, whose initial angle should be corrected by the average angle in the neutral gear coasting condition, can reflect the actual working state of the torsional vibration damper effectively. And the accuracy of the linear torsional stiffness obtained will be above 90%. Both simulation and experimental analysis results show that these two proposed application methods have high calculation accuracies and engineering feasibility.

Keywords. Noise and vibration, transmission system, torsional stiffness, working rotational angle.

1. Introduction

The vibration and noise (NVH) performance is one of the most important performances in the automobile [1, 2]. And the higher the vehicle level is, the larger the proportion of NVH performance has. Moreover, the transmission system plays a decisive role in the NVH performance of the automobile [3].

Automobile transmission system generally includes engine, clutch, torsional vibration damper, transmission, drive shaft and so on. While the four-wheel drive vehicle also includes propeller shaft, power divider, rear differential mechanism, etc. As the most important damping component in the transmission system, the torsional vibration damper has the key design parameters such as in the inertia moment of the active and the passive components, the value and the form of the torsional stiffness, the range of the working rotational angle, that directly affect the automobile NVH performance [4-6]. A suitable torsional vibration damper can effectively isolate the torsional vibration transmitted from the engine to the transmission, improving the transmission gear rattle, vehicle shaking and booming in the ignition and acceleration [7, 8]. The drive shaft and the propeller shaft also play important roles in the NVH performance of the transmission system. Their moment of inertias and torsional stiffness become the key parameters in the NVH problems such as automobile booming, judder [9, 10].

Based on the basic calculation method of the relative rotational angle, this paper proposes the application method of the working rotational angle of automobile torsional vibration damper and the application method of the torsional stiffness of rotational shaft system, which can effectively obtain the key working information of the transmission system under the whole vehicle state.
simulation and experimental analysis results show that these two proposed application methods have high calculation accuracy.

2. Calculation Method of Relative Rotational Angle and Its Applications

2.1. Calculation Method of Relative Rotational Angle
The basic principle of calculation method of the relative rotational angle is that calculating the high-precision relative rotational speed difference between the input and the output shaft of the measured shaft system or rotating component, then obtaining the relative rotational angle signal by integrating the speed difference. The specific process of this basic method is shown in figure 1, and the calculation steps are shown as follows.

1. Obtain the rotational speed pulse signals of the input and the output shafts of the measured shaft system or the rotating component. And calculate the high-precision rotational speeds according to the equation (1).

\[ n_{\text{in}}(t) = \frac{360}{\Delta T_{\text{in}}(t)} \quad \text{or} \quad n_{\text{out}}(t) = \frac{360}{\Delta T_{\text{out}}(t)} \]  

where \( \Delta T_{\text{in}}(t) \) and \( \Delta T_{\text{out}}(t) \) are the time intervals between two adjacent pulses passing the trigger voltage \( V_0 \) at the rising or falling edge in the input and the output pulse signals, respectively. The higher the signal sampling frequency \( f_s \) is, the more accurate the time intervals are. And \( z_{\text{in}} \) and \( z_{\text{out}} \) are the teeth number of the measured gears or the grid number of code discs at the input shaft and the output shaft, respectively. The units of \( n_{\text{in}}(t) \) and \( n_{\text{out}}(t) \) are deg/s.

2. Calculate the relative speed difference \( \Delta n(t) \) between the input and output shafts by equation (2).

\[ \Delta n(t) = n_{\text{in}}(t) - n_{\text{out}}(t) \]  

3. Obtain the relative rotational angle \( \theta_1(t) \) by discrete integration of the relative speed difference according to equation (3) whose unit is degree.

\[ \theta_1(t) = \sum_{j=0}^{t} \frac{\Delta n(j)}{f_s} \]  

4. Obtain the relative rotational angle \( \theta_2(t) \) without high-frequency interference by performing the low-pass filter on the signal \( \theta_1(t) \). The cut-off frequency \( f_c \) of the low-pass filter is generally set to 5 Hz.

2.2. Application Method
Combining many specific signal acquisition steps with the above method in different application scenarios, it can be further developed into a working rotational angle calculation method or a torsional stiffness calculation method for the rotational component.

2.2.1. Calculation Method of Working Rotational Angle of Automobile Torsional Vibration Damper.
The key point for this application scenario is the initial angle correction of the relative angle. Therefore, the original signal acquisition needs to be carried out according to these following steps

a) Accelerate the vehicle in these specific conditions which is that at a fixed throttle opening position \( O \), a fixed gear position \( M_n \), the clutch engagement and a low engine speed \( r_1 \).

b) Release the throttle to allow the vehicle to coast and slow down when the engine speed reaches to a predetermined speed \( r_2 \).
c) Shift the transmission into the neutral position quickly to allow the vehicle to coast for a certain distance when the engine speeds nearby down to speed $r_1$.

In this application, the working relative angle $\theta_1(t)$ between the input and the output shafts of the torsional vibration damper will be calculated according to Section 2.1 firstly. And then the real relative rotational angle $\theta(t)$ needs to be corrected by equation (4).

$$\theta(t) = \theta_1(t) - k$$

where $k$ is the average value of the relative rotational angle in the neutral gear coasting condition which is not loaded approximately.

2.2. Calculation Method of Torsional Stiffness of Rotating System. The key point of this application is that it is necessary to simultaneously obtain the working rotational angle and torque of the measured rotating shaft in the shaft loading and deformation condition. The speed signal and torque signal need to be collected by these following steps.

a) Accelerate the vehicle in these specific conditions which is that at a fixed throttle opening position $O$, a fixed gear position $n_M$, the clutch engagement and a low vehicle speed $v_1$.

b) Release the throttle instantaneously to allow the vehicle to coast and slow down when the vehicle speed reaches to a predetermined speed $v_2$.

c) Repeat steps a) and b) multiple times.

The calculation method of the shaft torsional stiffness is shown in figure 2, in which it needs to be added these following steps based on the basic method of the relative rotational angle in 0.

a) When collecting the speed pulse signal of the rotating shaft system, it is necessary to collect the torque $M'(t)$ of the measured shaft or the related shaft with the same sampling frequency $f_s$ simultaneously. And the shaft torque $M'(t)$ needs to be converted to the torque $M(t)$ of calculated shaft by equation (5).

$$M_i(t) = M'(t)/i$$
Where \( i \) is the transmission ratio between the measured and the calculated shafts. If the measured shaft and the calculated shaft are the same, this transmission ratio \( i \) will be set at 1.

b) Obtain the shaft torque \( M_2(t) \) without high-frequency interference by performing the low-pass filter on the torque signal \( M_1(t) \). The cut-off frequency \( f_c \) of the low-pass filter should be the same of the relative rotational angle \( \theta_2(t) \).

c) Draw the graph in which the relative rotational angle \( \theta_2(t) \) is the horizontal ordinate and the shaft torque \( M_2(t) \) is the longitudinal ordinate.

\[
M(t) = k\theta(t) + c
\] (6)

Since the torsional shaft is designed into the linear stiffness working area, the curve will show an approximately linear phenomenon [11, 12]. After eliminating the data around the zero torque and other disturbances, the curve can be fitted by one dimensional formula by equation (6) using the linear regression method. The slope parameter \( k \) is the torsional stiffness of the calculated shaft. It should be noted that the torsional stiffness obtained by the proposed method is the total equivalent torsional stiffness of between the input end and the output end where the speed pulse signals are measured.

3. Simulation Analysis

A one-dimensional simulation model of torsional vibration of an automobile transmission system was established. This model was used to simulate and analyze the torsional vibration characteristics of the transmission system under the 3rd gear wide-open throttle condition, as shown in figure 3. The torsional stiﬀnesses of both the dual-mass flywheel (DMF) and the front left driveshaft were considered. And the key parameters are shown in table 1 and table 2, respectively. The simulation step length is 1×10^{-4} second in this case.

**Figure 3.** The simulation model of torsional vibration of an automobile transmission system.

**Table 1.** The key simulation parameters of the dual-mass flywheel.

| Parameter                              | Value       | Parameter                              | Value       |
|----------------------------------------|-------------|----------------------------------------|-------------|
| Torsional stiffness of the 1st stage    | 2.4 Nm/deg  | Torsional stiffness of the 2nd stage    | 5.6 Nm/deg  |
| Moment of primary inertia              | 0.08 kg.m.m | Moment of secondary inertia            | 0.05 kg.m.m |
| Angle of torsional stiffness inflection point | 50 deg      | Initial angle                          | -33 deg     |
| External damping                       | 20 Nm       |                                        |             |

**Table 2.** The key simulation parameters of the front left driveshaft.

| Parameter      | Value       | Parameter      | Value       |
|----------------|-------------|----------------|-------------|
| Torsional stiffness | 106 Nm/deg | Moment of inertia | 0.00253 kg.m.m |
3.1. Working Angle of the Dual-Mass Flywheel

The relative rotational angle of the dual-mass flywheel was calculated by the proposed method, as shown in figure 4(a). The original rotational speed signals which were for processing were extracted from the primary inertia and the secondary inertia in the simulation model. Since the rotational speed signal had no phase information, the initial angle of the calculated working rotational angle was needed to be corrected accordingly.

When the transmission system achieved dynamic balance in this case, the relative angle between the input shaft and the output shaft was -33 deg. This calculated working angle with the initial angle correction is completely coincident with the simulated signal, as shown in figure 4(b).

3.2. Torsional Stiffnesses of the Transmission System

While calculating the working rotational angle of the dual-mass flywheel, the component torque was extracted for low-pass filtering. The curve of working rotational angle-torque was shown in figure 5(a). The curve showed that the dual-mass flywheel matched the two-stage torsional stiffness. And the calculated torsional stiffness of the 1st stage and the 2nd stage were respectively 2.42 Nm/deg and 5.68 Nm/deg, which were basically the same as the torsional stiffnesses set in the simulation model. Their accuracies were 99.2% and 98.6%, respectively.

Because the torsional stiffness of the 1st stage was too small, the working torque was designed below 80 Nm. The dual-mass flywheel would pass through the 1st stage working area quickly under the large-throttle condition, resulting that there were too little data points that could be used to fit the torsional stiffness of the 1st stage and the angle of inflection point became not clear.

![Figure 4](image1.png)

**Figure 4.** The working rotational angle of the dual-mass flywheel: (a) The calculated original relative rotational angle; (b) The comparison between the calculated signal and the simulated signal.

![Figure 5](image2.png)

**Figure 5.** The torsional stiffness curves calculated by the proposed method: (a) The torsional stiffness curve of the dual-mass flywheel; (b) The torsional stiffness curve of the front left driveshaft.

Similarly, the torsional stiffness curve of the front left driveshaft calculated by the proposed method was shown in figure 5(b). The calculated torsional stiffness was 106.7 Nm/deg, which was almost the same as that set in the simulation model, with an accuracy of 99.3%.
4. Application Analysis

4.1. Working Angle of the Dual-Mass Flywheel

The dual-mass flywheel, which can effectively attenuate the torsional vibration transmitted from the engine to the transmission, is an important component of the vehicle transmission system. During the development of the new automobile, it is necessary to obtain the working rotational angle signal of the dual-mass flywheel under some typical working conditions to assist the adjustment of the vehicle problem.

According to the layout characteristics of the transmission system in tested MT car, the magnetoelectric speed sensors were installed at the drive ring gear of the engine and the input gear of the 2nd gear which was fixed on the transmission’s input shaft, respectively. They could indirectly collect the high-precision speed signals at the input and the output shafts of the dual-mass flywheel, as shown in figure 6 with the sampling frequency of 102400 Hz.

![Figure 6](image6.png)

**Figure 6.** The speed pulse signals of the dual-mass flywheel: (a) The speed pulse signal of the input shaft; (b) The speed pulse signal of the output shaft.

The teeth number of the tested drive ring gear and the 2nd input gear were 114 and 36, respectively. Set the trigger voltage of the rising edge at 2V, the high-precision input speed and output speed were calculated. These two speed signals shown in figure 7, including wide-open throttle acceleration in the 4th gear, coasting in gear, shifting process and coasting in neutral gear.

![Figure 7](image7.png)

**Figure 7.** The input speed and the output speed of the dual-mass flywheel: (a) The input speed; (b) The output speed; (c) The relative speed difference.
The working rotational angle of the dual-mass flywheel before correction was obtained by performing a discrete integration process on the relative speed difference. Then the low-pass filter with a cut off frequency at 5 Hz was applied. It could be calculated that the average relative rotational angle was about -19.5 degree during coasting in neutral gear. The actual working angle after the correction of the initial angle was shown in figure 8(b).

The rotational angle fluctuation in the acceleration process was obviously larger than other working conditions, which was caused by the large speed fluctuation between the input and output ends of the dual-mass flywheel. In the process of coasting in gear, the engine did not ignite to achieve more effective fuel consumption, making it became a pure load dragged by the transmission. Therefore, their working angle had no fluctuation basically and run at a small negative angle range. But in the process of coasting in neutral gear, the engine resumed ignition to ensure that it run at the idle speed, which caused the fluctuation of working rotational angle to increase slightly.

Combining with the design torsional stiffness curve of the dual-mass flywheel shown in figure 9, it could be seen that the working angle fluctuation become bigger around the inflection point 36 degree between the 1st and the 2nd torsional stiffness, leading the speed fluctuation of the input shaft become more obvious. The maximum working rotational angle reached about 55 degree at 2500 rpm, which was close to the maximum design limit angle 57.7 degree. Therefore, there would be a potential risk of the vehicle comfort and drivability due to the sudden change in torsional stiffness. Some control techniques, such as increasing the maximum limit angle or limiting the engine output torque, should be considered in the adjustment of the dual-mass flywheel or the engine.

4.2. Torsional Stiffnesses of the Automobile Transmission System
In another AT four-wheel drive SUV, a hydraulic torque converter was matched between the engine
and the transmission to transfer power, and a propeller shaft and its related controllers were used between the front and rear wheels to achieve the timely four-wheel drive function.

According to the proposed procedure of calculating the torsional stiffness of rotating shaft system, these signals, which were the input and output speeds of the torque converter, the input speed of the propeller shaft, the output speed of the elastic coupling behind the propeller shaft, the torque of the front left and right drive shafts, and the torque of the propeller shaft, should be collected synchronously.

4.2.1. Torsional Stiffness of the Propeller Shaft Assembly. Due to the limitation of structural arrangement, it was impossible to directly measure the torsional stiffness of the propeller shaft. The output speed sensor only could be set at the location between the elastic coupling and the torque distributor, resulting that the torsional stiffness calculated was the equivalent torsional stiffness of the assembly of both the propeller shaft and elastic coupling.

The torsional stiffness of the propeller shaft measured on the torque bench was 119 Nm/deg, as shown in figure 10(a). The torsional stiffness of the elastic coupling was about 200 Nm/deg. The assembly of these two components was a series system, and its theoretical equivalent torsional stiffness was about 74.6 Nm/deg.

The tested torsional stiffness curve of the assembly calculated by the proposed method was shown in figure 10(b). This curve exhibited approximately linear phenomena during the acceleration process. And the equivalent torsional stiffness of that assembly fitted by the linear regression was 81.1 Nm/deg with an accuracy of 91.3%.

4.2.2. Torsional Stiffness of the Single-Mass Torsional Vibration Damper. A single-mass torsional vibration damper was designed in the lock-up clutch which was integrated in the hydraulic torque converter in this tested automobile. The engine power would be transmitted to the transmission through the damper while the clutch was locked. The tested torsional stiffness curve of the torsional vibration damper obtained by the proposed method was shown in figure 11, in which the torque was converted from the torque of the front left and right drive shafts and the torque of the propeller shaft. The tested torsional stiffness was about 10.9 Nm/deg and the accuracy was 97.2% which was compared with the design value 10.6Nm/deg.

Figure 10. The bench and field test curves of torsional stiffness of the propeller shaft: (a) The bench test curve of torsional stiffness of propeller shaft; (b) The field test curve of torsional stiffness of the assembly between propeller shaft and elastic coupling.

Figure 11. The bench and field test curves of torsional stiffness of the single-mass torsional vibration damper.
In conclusion, the proposed calculation methods both of the working rotational angle of the automobile torsional vibration damper and the torsional stiffness of the rotating shaft system had good accuracies in this paper.

5. Conclusions
The basic calculation method of the working relative rotational angle of the rotating system is introduced in this paper firstly. Combining many specific signal acquisition steps with the above method in different application scenarios, it can be further developed into a working rotational angle calculation method and a torsional stiffness calculation method for the rotational component, respectively. Both the simulation and experimental analysis results show that these two types of application method proposed have high calculation accuracies.

(1) Since the relative rotational angle signal is obtained by preforming a discrete integration on the relative speed difference, its initial angle is unbelievable and needed to be corrected. In the operation of obtaining the working angle signal of the automobile torsional vibration damper, it is necessary to calculate the initial angle which is not loaded approximately in the neutral gear coasting condition. The signal information, such as the amplitude of the working angle and the fluctuation, can reflect the working state of automobile torsional vibration damper effectively.

(2) The proposed calculation method of the torsional stiffness is based on the design principle that the rotating shaft system generally works on the linear elastic deformation region. Combining the relative rotational angle and the shaft torque, the torsional stiffness can be fitted by the linear regression with an accuracy above 90%. Due to the limitation of the measured position of the rotational speed signal, the obtained torsional stiffness shall be the equivalent torsional stiffness between the rotational input and output speeds.

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