Principle analysis of torque ripple test bench for electric machine for electric vehicle application

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Abstract. The torque ripple is mainly caused by the interaction between the stator magnetomotive force of the motor and the magnetomotive force harmonic of the rotor. With the gradual improvement of the comfort and stability requirements of the passenger car, the vibration and noise of the drive motor for the electric vehicle and ripple impact and other indicators affecting the comfort of the vehicle are receiving more and more attention. On the basis of ensuring the output torque capability of the motor, further improving the quality of the motor is one of the important purposes of the project research. At present, the research on the motor torque ripple measurement method is not perfect, and the motor test equipment cannot eliminate the interference of the load fluctuation on the motor torque test and cannot be accurately measured. In this paper, the load dynamometer and elastic coupling by simulating the input and output torque are studied, angular velocity and angular acceleration on the basis of previous research on torque ripple test at home and abroad. The influence of load dynamometer and elastic coupling on the test structure is studied and the test system of no-load dynamometer and elastic coupling is determined.

1. Submitting the manuscript
Researchers at home and abroad have made many explorations on torque ripple measurement, among which the typical ones are:

1.1. Direct measurement
The traditional test method uses a torque sensor to measure the shaft torque, and the measured torque ripple is proportional to the torsion angle. This method may affect the measurement bandwidth because of the resonance of the whole structure including the sensor. The power train system can be simplified to a single degree of freedom system for analysis. This method is simple to operate, but has too few factors to be considered. It can only be used at low speed, and the test range is narrow.

1.2. Balance measurement
The improved balance measurement method based on the direct method has the characteristics of making full use of the resolution of torque sensor and high measurement accuracy. The motor is connected to the load as a whole, and the transmission shaft of the torque sensor is used to transfer the torque of the motor to the inertia flywheel, so that the average torque output of the motor is balanced with the load, and the torque fluctuation part is transferred to the sensor, and the torque fluctuation component can be measured through the torque sensor. Compared with the direct method, the balance method overcomes the torque fluctuation in the transmission process, but it still uses the load and
cannot avoid the load fluctuation. In addition, angular acceleration is also used to solve the problem of narrow range of velocity measurement in traditional methods. However, on the one hand, the change of parameters is still limited, and the value of parameters cannot be arbitrarily adjusted. On the other hand, inertia torque and bearing torque cannot be eliminated by adjusting the measurement alone. Using the current signal to establish the observation model to calculate the torque fluctuation, which is heavily dependent on the internal specific parameters of the motor, and is not suitable for general motor testing.

2. Test bench design
The purpose of this paper is to establish a general motor test method. For this reason, a method of inertia loading was proposed. The torque output of the motor at different positions was measured by rotating the rotor at a number of electrical angles in an electric cycle to draw a graph, which can represent the torque fluctuation of the motor in an electric cycle.

In order to analyze the torque fluctuation in the actual dynamic running process of the motor, large inertia flywheel was selected as the load to simulate the dynamic running process of the motor, and finally the torque pulsation test method of the motor was formed.

![Figure 1: schematic diagram of motor test bench structure](image)

The test bench is shown in figure 1, which is mainly composed of the motor under test, acceleration sensor, torque sensor, rotating disc, elastic coupling, flywheel and other parts. Motor torque pulse test of cogging torque test and large inertia load test can be achieved.

3. Test system analysis
3.1. Test philosophy
According to the design of the above test bench system, the mechanical equation of the entire shaft system is:

\[
T_e = (J_1 + J_2) \frac{d\omega_r}{dt} + (D_1 + D_2)\omega_r
\]

where:
- \(T_e\) — Electromagnetic torque
- \(J_1\) — Electric machine moment of inertia
- \(\omega_r\) — System angular velocity
- \(D_1\) — Transfer the damping coefficient of the system
- \(J_2\) — The moment of inertia of the flywheel
- \(D_2\) — Flywheel damping coefficient

3.2. Dynamic loading process
Usually, the inertia of the flywheel moment of inertia of \(J_2\) is greater than the moment of inertia of the motor \(J_1\), cause the flywheel system of mechanical time constant is larger, so if there are fluctuations in torque, the angular velocity of the flywheel that \(\omega_2\) changed little, the type (2) shows that volatile components of \(T_e\) is fully reflected in the \(T_t\), so by measuring the output voltage signal of the torque sensor can be measured the fluctuation of motor torque.

Simplifying system, get the equivalent model shown in figure 2:
Figure 2: equivalent model diagram

The transfer function of the system can be obtained as:

\[
G(s) = \frac{F(s)}{E(s)} = \frac{s^2 J_m B_s + s K J}{s^2 J_m + s^2 (J_m + J_f) + s (J_f K + J_m K + B_s B_f + B_m K)}
\]

In the rotor mechanical system, the damping term is determined by the structure of the system, and temperature, stress amplitude and frequency will all have an impact on it, so it is difficult to determine the value, and usually not in the same order of magnitude with other parameters, so the damping term can be ignored, and the transfer function can be simplified.

\[
G(s) \approx \frac{1}{s^2 J_m + J_f + J_m}
\]

3.3. Output characteristic analysis

Let the input of the system be the unit sinusoidal signal, combined with the transfer function, after transformation, the final output equation is:

\[
c(t) = \frac{J_m}{o_0^2} \left[ o_0 \sin(o_0 t) - o_0 \sin(o_0 t') \right]
\]

As can be seen from the above equation, when the sine wave signal of a certain frequency is input to the system, the steady-state output contains two different frequency signals. When the input frequency is equal to the resonant angular frequency, the system will oscillate. Since the measuring device is simplified to a single degree of freedom system, the motor torque is determined by the measured phase current and the known motor parameters according to the torque equation. Select the appropriate voltage vector, so that the motor torque to reach the measurement range. Frequency ranges from zero to 1600Hz to cover the required measurement bandwidth. The transmission characteristics of shaft torque \(T_m\) and motor torque \(T_e\) are shown in figure 3, and the frequency response is divided into two parts. On the left of the resonant frequency \(f_{res}\), the transfer function is close to the ideal case, while on the right, the error becomes obvious. Therefore, it is impossible to characterize the torque ripple at high speed with sufficient amplitude. In this case, the resonant frequency is a key factor. It indicates that the frequency band of the signal directly tested by the torque sensor has a certain range, that is to say, the speed tested by the sensor is limited to a certain extent, only about one-tenth of the resonance frequency can be tested, and the motor cannot be tested in the full speed range.
In order to solve the measurement problem of torque ripple in the full speed range of the motor, this paper plans to use an acceleration sensor to measure the angular acceleration of the shaft in the high-frequency region. Its structure diagram as shown in figure 4: the rotational inertia of the acceleration sensor is \( J_{m,s} \), and rotation acceleration is \( \alpha \), transfer function of the system are as follows:

\[
\frac{\alpha_m}{T_s} = \frac{s^2 J_1 + s b_1 + c_0}{s^2 J_{m,s}^2 + s b_1 (J_{m,s} + J_1) + c_0 (J_{m,s} + J_1)}
\]

(5)

4. Virtual prototype simulation analysis

The three-dimensional models of the motor shaft and flywheel were established, and the corresponding rotary inertia was obtained by assigning material properties, as shown in the following table. The corresponding parameters of torque sensor and elastic coupling are obtained through the product manual of selected products, as shown in table 1 below:

| Parameter                           | Settings |
|-------------------------------------|----------|
| Moment of inertia of shaft          | 0.035    |
| Torsional stiffness of torque sensor| 180      |
| Damping coefficient of torque sensor| 0.01     |
| Torsional stiffness of an elastic coupling | 13      |
| Damping coefficient of elastic coupling | 0.1      |
| The moment of inertia of the flywheel | 2.1      |

By establishing the system simulation analysis model of motor, torque sensor, elastic coupling and flywheel, the elastic damping components are used to replace torque sensor and elastic coupling.

As shown in figure 5, there is no elastic coupling in the test system, and only the torque sensor is regarded as a flexible part. Assuming the motor shaft end torque is \( F=50+4\sin (2\pi \times 50 \times t) \), the torque ripple input is simulated to analyze the torque angular velocity angular acceleration waveform at the torque sensor. The corresponding waveforms at the input waveform and the torque sensor are compared, as shown in figure 7, 8 and 9. It can be seen that the input and output of the system follow well, which conforms to the condition that torque sensor is used for measurement in the low frequency region and acceleration sensor is used for measurement in the high frequency region mentioned above.
The test system shown in figure 6 has couplings. Assuming the motor shaft end torque is $F=50+4\sin(2\pi*50*t)$, the torque ripple input is simulated and the torque, angular velocity and angular acceleration waveforms at the torque sensor are simulated and analyzed. The corresponding waveform comparison between the input waveform and the torque sensor is shown in figure 10, 11, 12. For couplings with different stiffness, the comparison of output torque and input torque is shown in figure 13.

Figure 7: comparison of input and output torque

Figure 8: comparison of input and output angular velocities

Figure 9: comparison of input and output angular accelerations

Figure 10: shows the comparison of input and output torques of the coupling system

Figure 11: shows the comparison of input and output angular velocities of the coupling

Figure 12: shows the comparison of input and output angular accelerations of the coupling
Figure 13: comparison of output torque and input torque of couplings with different stiffness

It can be seen from the comparison between figure 7, 8, 9 and figure 10, 11 and 12 that the torque follower of the system decreases slightly with the introduction of the coupling. As can be seen from the analysis in figure 13, the coupling stiffness has a great influence on the system transfer. In order to ensure the system follow, the coupling with a large stiffness should be selected. The introduction of the elastic coupling can compensate for the rotor eccentricity and other problems in the assembly process. The elastic coupling is used to connect the flywheel and the torque sensor. At the same time, due to the use of elastic coupling, the two ends of the coupling produce different angular displacement, the measuring shaft of the torque sensor produces deformation, and the corresponding voltage signal is output. At the same time, the elastic coupling can make up the installation error of the shaft system and ensure that the system will not be damaged by the installation error and vibration impact.

5. Conclusion and prospect

Through simulation and comparative analysis, the following conclusions are drawn in this paper:

(1) under the condition of including load dynamometer, the load of various frequencies and amplitudes will make the measurement result have 5% error with the initial input. Therefore, the test system should avoid the use of load dynamometer to improve the measurement accuracy.

(2) with the introduction of coupling, the torque follower of the system decreases slightly. However, the introduction of elastic coupling can compensate for rotor eccentricity and other problems in the assembly process. In addition, it compensates the rotor eccentricity and other problems existing in the assembly process, and makes up the installation error of the shaft system, so as to ensure that the system will not be damaged by installation deviation and vibration impact.

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