Dynamic Design of the Test Bench of Electric Vehicle Motor

Huibin Li1,*, Peiyun Xu2, Cheng Cao3, Dongmei Hu4, Xiaojun Yan5 and Zhihuan Song6

1School of Mechanical and Vehicular Engineering, Beijing Institute of Technology, Beijing, P.R. China
2School of Management and Economics, Beijing Institute of Technology, Beijing, P.R. China
3,4,5Acoustic center, SAIC Volkswagen Automotive Company, Co., Ltd., No.1199 Yuan Gong Road, Anting, Shanghai, P.R. China
6Huayu Automotive Electric Drive System Co., Ltd, No.88, Jinwen Road, Pudong New Area, Shanghai, P.R. China

*Corresponding author email: huibinli@163.com

Abstract. In order to avoid the resonance of the bench and the laboratory floor, it is necessary to carry out vibration analysis and dynamic design on the motor test bench. Firstly, the lightweight target and the constraint conditions related to the rigid body modal frequency and elastic modal frequency of the platform are set, and the structure of the platform is optimized. Then, the three-dimensional model of the whole bench is built and the finite element is meshed to carry out modal analysis. The optimized modal analysis results show that the natural frequencies of the designed bench will not be excited by various vibration excitation forces at the set speed of motor test, the vibration isolation rate of the lower bracket reaches more than 52% (except that the vibration isolation rate of the lower bracket is 36.30% at 1000 rpm), which effectively eliminates the resonance problem between the bench and foundation during motor NVH test, and the normal performance of motor and NVH test can be effectively ensured.

Keywords: Electric vehicle motor; Dynamic design; Finite element method; NVH.

1. Introduction
In order to solve the background noise caused by the test bench’s vibration, and to avoid the bench and the laboratory floor resonance, and to improve the accuracy of the motor bench test, it is necessary to carry out vibration analysis and dynamic design on the motor test bench [1-7].

As shown in Figure 1, in addition to the loading motor, the tested motor, the various kinds of sensors and bracket, the new designed test bench also includes the L-shaped plate for supporting motors, the base plate for fixing the relative location of two motors, the isolator to be used for isolating the vibration between the test bench and the laboratory platform, and the mounting pad for being simulated to support the motor and to isolate motor’s vibration, etc.
2. Principle of Dynamic Design of Bench

The objective function and constraint conditions are set for the test bench for low vibration and noise motor, and the genetic algorithm was used to optimize the initial motor bench structure.

\[
\begin{align*}
\text{min} & \quad M \quad \text{subject to} \\
\quad & f_{r,i} \leq 12\text{Hz} \\
\quad & f_{e,k} - \frac{n}{60} \geq \Delta_1 \\
\quad & f_{e,k} - \frac{2n}{60} \geq \Delta_1 \\
\quad & f_{e,k} - \frac{2pn}{60} \geq \Delta_2 \\
\quad & f_{e,k} - \frac{zn}{60} \geq \Delta_3 \\
\quad & \eta \geq 40\%
\end{align*}
\]

Among them,

- \(M\) is the total mass of the motor bench;
- \(f_{r,i}\) is the rigid body mode of the motor bench, \(i\) is the rigid body mode order, take \(i = 1, 2, 3, 4, 5, 6, 7, 8\);
- \(f_{e,k}\) is the first eight order elastic mode of the motor bench, and \(k\) is taken as \(1, 2, 3, 4, 5, 6, 7, 8\);
- \(\Delta_1\) is the frequency interval set to avoid the motor bench being resonated by the excitation frequency from misaligned connection between the two pairs of the motor rotor;
- \(\Delta_2\) is the frequency interval set to avoid the motor bench being resonated by the electromagnetic excitation frequency of the motor;
- \(\Delta_3\) is the frequency interval set to avoid the motor bench being resonated by the excited frequencies from the motor stator tooth slot cycle electromagnetic field force;
- \(n\) is the motor rotating speed, used in the towing experiment, that is, \(n = 1000\text{rpm}, 2000\text{rpm}, 3000\text{rpm}, 4000\text{rpm}, 5000\text{rpm}, 6000\text{rpm}, 7000\text{rpm}, 8000\text{rpm}, 9000\text{rpm}\) and \(10000\text{rpm}\).
- \(p\) is the number of motor poles;
- \(z\) is the number of stator slot;
- \(\eta\) is the vibration isolation efficiency of the vibration isolation system supported by the platform. The theoretical calculation process is as follows.

In order to ensure that the bench system has a good vibration isolation ability for the bottom foundation, it is necessary to calculate and optimize the parameters of vibration isolation equipment such as air spring. Set the excitation force generated by the motor as following:

\[
F = F_0 \cos \omega t
\]
Among them, $F_0$ is the exciting force amplitude of the motor; $\omega$ is the exciting force circle frequency. If $x$ is used as the displacement response of the motor bench, the vibration equation of the motor bench can be expressed as

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos\omega t$$

(3)

Then the solution of equation (3) is expressed as

$$x = X \cos(\omega t - \psi)$$

(4)

Among them, $\psi$ is the phase difference; $X$ is the vibration displacement amplitude.

By substituting formula (4) into formula (3), we can get

$$X = X_0 \beta = \frac{F_0}{k} \frac{1}{\sqrt{(1-\gamma^2)^2 + (2\zeta \gamma)^2}}$$

(5)

Among them, $k$ is the stiffness of the support system; $\beta$ is the dynamic factor of the platform support system; $\gamma$ is the frequency ratio of the platform support system; $\zeta$ is the damping ratio of the bench supporting system.

The amplitude of vibration source force can be obtained from equation (5)

$$F_0 = kX\sqrt{(1-\gamma^2)^2 + (2\zeta \gamma)^2}$$

(6)

The load transmitted by the motor vibration source to the bottom foundation is expressed as

$$F_k = kx = kX\cos(\omega t - \psi)$$

(7)

And

$$F_c = c\dot{x} = -c\omega X\sin(\omega t - \psi)$$

(8)

Among them, $F_k$ is the spring force; $F_c$ is the damping force; $c$ is the damping coefficient of the support system.

The resultant force amplitude of the two forces is calculated as following

$$F_T = X\sqrt{k^2 + c^2\omega^2} = kX\sqrt{1 + (2\zeta \gamma)^2}$$

(9)

Therefore, the force transmissibility is defined as

$$T = \frac{F_T}{F_0} = \sqrt{\frac{1+(2\zeta \gamma)^2}{(1-\gamma^2)^2 + (2\zeta \gamma)^2}}$$

(10)

The vibration isolation efficiency of the motor bottom isolation system is expressed as

$$\eta = 1 - T = 1 - \sqrt{\frac{1+(2\zeta \gamma)^2}{(1-\gamma^2)^2 + (2\zeta \gamma)^2}}$$

(11)

Through the above theoretical calculation and optimization analysis, the vibration isolation pad with stiffness of 110N/mm and the air spring with stiffness of 80N/mm are selected for the final vibration isolation system.

3. Modal Simulation of Bench with Finite Element Method

The structure model of motor test bench is established, and the finite element mesh is drawn by HyperMesh software, as shown in Figure 2.

In order to study the excitation of the natural frequency of the test bench in the motor operating speed range (0-10000 rpm), the modal analysis is carried out according to the relationship between the motor frequency and speed.

$$n = \frac{6f}{2p}$$

(12)

Among them, $n$ is the motor speed; $f$ is the motor frequency; $p$ is the number of the motor pole. In this paper, $p=4$. 
The frequency range of the motor which should be focused on is 0-1333.3Hz. Therefore, the first 19 modes of the finite element model of the platform are extracted. The specific modal frequency and mode description are shown in Table 1. Figure 3 is the first order modal mode.

Table 1. Simulation results of modal parameters of the motor bench

| Order | Frequencies/Hz | Motor speed/rpm | Mode                                                                 |
|-------|----------------|-----------------|---------------------------------------------------------------------|
| 1     | 24.1           | 180             | The whole bench moves horizontally from left to right                |
| 2     | 24.6           | 185             | The whole bench moves forward and backward                         |
| 3     | 36.0           | 270             | The platform swings around the z-axis                              |
| 4     | 85.4           | 640             | The platform swings around the x-axis                              |
| 5     | 91.5           | 685             | The whole platform jumps vertically up and down                     |
| 6     | 108.0          | 810             | The bench swings around the y-axis                                 |
| 7     | 198.3          | 1490            | Reverse bending vibration of two L-shaped supports                  |
| 8     | 286.3          | 2150            | The torsional vibration of the two L-shaped supports in the same direction; Torsional vibration of bottom plate |
| 9     | 342.1          | 2565            | Bending vibration of base plate                                    |
| 10    | 357.5          | 2680            | Overall torsion of bench                                          |
| 11    | 489.6          | 3670            | Bending vibration of two L-shaped supports                         |
| 12    | 496.1          | 3720            | Rotor "breathing"                                                 |
| 13    | 517.7          | 3880            | The motor group moves forward and backward                         |
| 14    | 547.4          | 4105            | Torsional vibration of two L-shaped supports                       |
| 15    | 560.9          | 4205            | Motor group swing around z-axis                                    |
| 16    | 572.2          | 4290            | Reverse bending vibration of two L-shaped supports                  |
| 17    | 694.6          | 5210            | The two motors swing slightly around the y-axis reversely           |
| 18    | 1026.6         | 7700            | Overall slight torsional vibration                                 |
| 19    | 1322.7         | 9920            | Slight reverse bending vibration of two L-shaped supports           |

Figure 2. Mesh grid of motor bench

Figure 3. The first mode of motor bench
Figure 4. Vibration isolation rate curve of motor test bench under various working conditions
When the motor is tested in the speed range of 0-10000 rpm, the above natural frequencies will not be excited by various vibration excitation forces, which meets the constraint conditions described in equation (1), and ensures the normal operation of motor performance and NVH test. As shown in Figure 4, the air spring between the base plate of motor NVH test bench and the foundation is designed and optimized, so that the vibration isolation system can effectively attenuate the vibration energy under the excitation of electromagnetic vibration of motor and periodic electromagnetic force of stator cogging. The vibration isolation rate of the lower bracket reaches more than 52% (except that the vibration isolation rate of the lower bracket is 36.30% at 1000 rpm), which effectively eliminates the resonance problem between the bench and the foundation during the motor NVH test, and ensures the normal operation of the motor NVH test.

4. Summary
In this paper, the motor test bench with a low vibration and noise level is designed. The main conclusions are as follows:
(1) The lightweight target and the constraint conditions related to the rigid body modal frequency and elastic modal frequency of the platform are set, and the structure of the platform is optimized.
(2) The optimized modal analysis results show that the natural frequency of the designed bench will not be excited by various vibration excitation forces at the set speed of motor test, the vibration isolation rate of the lower bracket reaches more than 52% (except that the vibration isolation rate of the lower bracket is 36.30% at 1000 rpm), which effectively eliminates the resonance problem between the bench and the foundation during the motor NVH test, and the normal performance of motor and NVH test can be effectively ensured.

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