Research on modal analysis method of Multiple Units car body in the preparation conditions

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Abstract: The first-order vertical bending frequency is an important parameter to evaluate the vibration performance of car body in the preparation conditions. Increasing its frequency value can avoid the resonance between the car body and the bogie frame, reduce the vibration, and improve the running stability and riding comfort. Due to the low first-order vertical bending frequency, combined with layout optimization of equipment under vehicle and structure optimization, the optimal equipment layout scheme and the thickness of the key parts were obtained based on the refined modeling. The first-order vertical bending frequency of car body was increased from 7.63Hz to 8.35Hz, which was increased by 9.4%.

1. Introduction

The body vibration of high-speed train is related to train operation safety and comfortableness [1]. The car body modal is an important factor to be controlled in car body design because the lower first-order vertical bending modal will seriously threaten the safety of vehicle operation [2]. As for the evaluation standard “The bench test method for dynamic performance of locomotive and rolling stock”, it is clearly stipulated that the first-order vertical bending frequency should avoid the nodding and sinking frequency of the bogie, or frequency value should not be lower than 10Hz [3]. Increasing the first-order vertical bending frequency can avoid the resonance between the car body and the bogie frame, reduce the vibration of the car body, and improve the running stability and riding comfort [4]. Refined modeling can accurately simulate the car body modal, and the same time the reasonable arrangement of equipment location under vehicle and structure optimization can effectively improve the first-order vertical bending frequency.

In order to obtain the exact value of the first-order vertical bending modal, Xie S [5] created five modal analysis models of car body in the preparation conditions according to the principle of "stiffness equivalence". The comparison of simulation analysis and test results showed that the refined modeling can accurately simulate the car body modal. Zhou H [6] designed a reasonable arrangement scheme of equipment under the vehicle on the basis of ensuring a small axle load deviation, which effectively improved the first-order vertical bending frequency. Based on the sensitivity analysis method, Zhang J [7] selected the appropriate design variables, and selected mutual effects of increasing the low frequency and decreasing the mass as objective functions and constraints to realize the optimization design. Based on the relative sensitivity analysis method, Jia S [8] determined the area that had a greater impact on the first-order vertical bending frequency, and the corresponding plate thicknesses were selected as the design variables to complete the optimization design of car body structure. However, the refined
modeling, layout optimization of equipment under vehicle and structure optimization were not considered at the same time for the modal study of car body in the preparation conditions.

In this paper, taking the car body of Multiple Units in the preparation conditions as the research object, the modal was simulated accurately based on the detailed weight distribution and equipment center of gravity of car body in the preparation conditions considering all of the above optimization methods. Furthermore, the first-order vertical bending frequency was improved effectively on the basis of ensuring the small increase of the car body mass, which combined with layout optimization of equipment under vehicle and structure optimization.

2. Refined modeling and modal analysis

The car body mainly consists of side wall, underframe, roof, end wall and other welding parts, which are made of aluminum-alloy hollow extrusion bases. All equipment with the mass greater than 80kg like the seats, luggage cabinets, doors and other interior equipment as well as the air conditioner on the car body were simulated by mass elements. According to the actual equipment center of gravity, the flexible elements were used to mount the equipment according to the actual installation position. The equipment under vehicle were also simulated by mass elements based on their gravity. And the equipment mounting position used three-direction spring elements to simulate the actual connection stiffness. Ultimately, the center of gravity of the simulation model was close to the actual center of gravity of the design car body in the preparation conditions.

Figure 1 showed the finite element model of the car body, which was divided into 1299842 elements, 1059370 nodes, and the total mass of car body in the preparation conditions was 35.18t. The first-order vertical bending frequency of car body was 7.63Hz, and the vibration modal was shown in Figure 2.

Fig. 1. Finite element model      Fig. 2. First order vertical bending vibration modal

3. Layout optimization of equipment under vehicle

The mass of car body (without equipment under vehicle) \(m_0\) is 29.87t, the longitudinal coordinate of car body center of gravity \(L_0\) is 333 (mm), and the longitudinal length of equipment cabin under vehicle \(L_1\) is 12380mm. The equipment was suspended on the underframe chute, and the initial equipment layout and specific parameters of equipment were shown in Table 1.

### Table 1 Initial layout and specific parameters of equipment under vehicle

| Equipment location | Location 1 | Location 2 | Location 3 | Location 4 | Location 5 | unit |
|-------------------|------------|------------|------------|------------|------------|------|
| Equipment name    | Sewage tank| Air compressor | Vehicle power and brake controller | Exhaust and brake air cylinder unit | Traction converter | /    |
| Equipment number \((i)\) | 1          | 2          | 3          | 4          | 5          | /    |
| Longitudinal      | 5037       | 3652       | 1831       | 138        | -3882      | mm   |
According to the equilibrium equation of force, there are:
\[ G + \sum F_i = F_{N1} + F_{N2} \]  
(1)

According to the moment balance equation, there are:
\[ GL_0 + \sum F_i x_i + (F_{N1} - F_{N2}) \times L_0/2 = 0 \]  
(2)

Let:
\[ \delta = F_{N1} - F_{N2} \]  
(3)

\( \delta \) is the vehicle axle load deviation. When the vehicle axle load deviation is small, FN1 and FN2 are close to each other, at this time \( \delta \to 0 \).

Let:
\[ f(x) = GL_0 + \sum F_i x_i \]  
(4)

Let:
\[ g(x) = |f(x)| = |\sum F_i x_i + C| \]  
(5)

Where C is a constant and \( C = GL_0 \).

So far, the problem of minimizing the load-bearing deviation between the front bogie and the rear bogie is transformed into the problem of minimizing the absolute value of torque of vehicle body weight and equipment weight under vehicle to y-axis.

### 3.2. Establishment of the mathematical model of optimization problem

In order to facilitate calculation, the problem of minimum absolute value of function could be further transformed into the problem of minimum square value of function in mathematics.

Let:
\[ h(x) = g^2(x) = (\sum F_i x_i + C)^2 \]  
(6)

Hypothesis: \( F = [F_1, F_2, F_3, F_4, F_5]^T, x = [x_1, x_2, x_3, x_4, x_5]^T \).

Then:
\[ h(x) = (F^T x + C)^2 = x^T FF x + 2CF^T x + C^2 \]  
(7)

Let:
\[ H = 2FF^T = 2 \begin{bmatrix} F_1 F_1 & \cdots & F_1 F_5 \\ \vdots & \ddots & \vdots \\ F_5 F_1 & \cdots & F_5 F_5 \end{bmatrix} \]  
(8)

Let:
\[ f = 2CF^T = 2C [F_1, F_2, F_3, F_4, F_5] \]  
(9)

Substituted formula (6) with (7) and (8), the following formula is obtained:
\[ h(x) = \frac{1}{2} x^T H x + f x + C^2 \]  
(10)
Because C is a constant, the minimum problem of \( g(x) \) is transformed into the minimum problem of \( l(x) = \frac{1}{2} x^T H x + f^T x \), which is a quadratic programming problem. The specific optimization model is as followed:

\[
l(x) = \frac{1}{2} x^T H x + f^T x
\]

\[
\text{S.T.} \quad \begin{cases} 
- \left( x_1 - L_{x01} - \frac{l_{x1}}{2} \right) \leq \frac{l_{11}}{2} \\
x_5 - L_{x05} + \frac{l_{x5}}{2} \leq \frac{l_{11}}{2} \\
(x_2 - x_1) + (L_{x01} - L_{x02}) \geq \frac{l_{x1}}{2} + \frac{l_{x2}}{2} + d \\
(x_3 - x_2) + (L_{x02} - L_{x03}) \geq \frac{l_{x2}}{2} + \frac{l_{x3}}{2} + d \\
(x_4 - x_3) + (L_{x03} - L_{x04}) \geq \frac{l_{x3}}{2} + \frac{l_{x4}}{2} + d \\
(x_5 - x_4) + (L_{x04} - L_{x05}) \geq \frac{l_{x4}}{2} + \frac{l_{x5}}{2} + d
\end{cases}
\]

Where, equations (12) and (13) are constraints. Equation (12) indicates that the end equipment shall not exceed the boundary of equipment cabin under vehicle; Equation (13) indicates that interference between the equipment is avoided.

\( x_i \) is the longitudinal coordinate of the equipment, \( L_4 \) is the longitudinal length of equipment cabin under vehicle, \( L_{xoi} \) is the longitudinal centroid offset of the equipment, \( L_{xi} \) is the longitudinal width of the equipment, \( d \) stands for equipment maintenance interval and \( d = 400 \text{mm} \).

3.3. Optimized equipment layout scheme

For the five equipment under vehicle mentioned above, the quadratic programming algorithm was used to optimize the problem considering all the equipment layout schemes.

The optimization results showed that: in a total of 120 possible layout schemes, 20 schemes satisfied the constraints of the optimization problem. The optimal arrangement scheme was obtained by comparing the calculated values of the first-order vertical bending frequency of each scheme: From the front end to the rear end, the equipment under vehicle was arranged in the order of 4, 2, 1, 5 and 3, and the longitudinal coordinates of the equipment were 5535, 3746, 702, -2028 and -5286 (mm) respectively. After the layout optimization of equipment under vehicle, the modal of car body in the preparation conditions was recalculated, and the calculated value of the first-order vertical bending frequency was 7.90Hz.

4. Optimization of car body structure

The structural parameters are the main factors to determine the modal parameters. The structural optimization was carried out to further improve the first-order vertical bending frequency based on ensuring a small mass increase of the car body.

4.1. Selection of design variables

A total of 36 plate thickness values of the roof, end wall, floor, side wall and underframe were selected according to the structural characteristics of the car body. Then, the sensitivity of the first-order vertical bending modal frequency and mass of the car body to the plate thickness change were calculated, as shown in Figure 4 and Figure 5. The ratio of first-order vertical bending frequency sensitivity \( F \) to mass sensitivity \( M \) was expressed by relative sensitivity \( S \) in order to better reflect the influence of the thickness of main car body profiles on car body mass and first-order vertical bending frequency, i.e., \( S = F/M \). The results of relative sensitivity calculation were shown in Figure 6.
According to the analysis results of relative sensitivity, 25 variables including roof inner and outer skin \((t_1, t_2, t_6)\), underframe side beam \((t_{29}, t_{30}, t_{31}, t_{32}, t_{33}, t_{34}, t_{35}, t_{36})\), end wall inner and outer skin and rib \((t_7, t_8, t_9, t_{10}, t_{11}, t_{12})\) and side wall rib \((t_{22}, t_{23})\) were selected as the optimization design variables, and the initial values were shown in Table 2.

4.2. Optimization calculation

The specific optimization model was as followed:

\[
\begin{align*}
\text{find } & \quad X = (x_1, x_2, \ldots, x_{25})^T \\
\text{max } & \quad f_1(X) \\
S.T. & \quad m(X) \leq m_U \\
& \quad X_L \leq X \leq X_U 
\end{align*}
\]

(14)

Where \(x\) is the column vector of design variables, \(x_i\) is the \(i\)th design variable, \(f_1(X)\) is the first-order vertical bending frequency of car body in the preparation conditions, \(m(X)\) is the total mass of the car body, \(m_U\) is the upper limit of body mass, and \(m_U = 36.24\), \(X_L, X_U\) are the upper and lower limits of design variables.

The optimization results of design variables were shown in Table 2. The mass of optimized car body was 35.64t and the first vertical bending frequency was 8.35Hz.

| number | Initial value | Optimized value | number | Initial value | Optimized value |
|--------|---------------|-----------------|--------|---------------|-----------------|
| \(t_1\) | 3.00          | 3.50            | \(t_{33}\) | 4.00          | 4.50            |
| \(t_2\) | 3.00          | 2.50            | \(t_{34}\) | 6.00          | 6.00            |
| \(t_6\) | 3.00          | 3.00            | \(t_{35}\) | 5.00          | 6.00            |
| \(t_{15}\) | 2.50         | 3.00            | \(t_{36}\) | 4.00          | 4.50            |
| \(t_{16}\) | 2.50         | 3.00            | \(t_7\)   | 8.00          | 6.50            |
| \(t_{19}\) | 3.00         | 3.50            | \(t_8\)   | 8.00          | 6.50            |
5. Conclusion

(1) The refined modeling was carried out to simulate the body modal more accurately according to the detailed distribution of car body mass and the actual equipment center of gravity. The first-order vertical bending frequency of car body in the preparation conditions was 7.63Hz.

(2) By means of quadratic programming method, 20 groups of equipment layout schemes were obtained. Finally, the optimal layout scheme was selected, and the first-order vertical bending frequency of car body was calculated to be 7.90Hz, which was increased by 3.5%.

(3) A total of 25 plate thicknesses were selected as the design variables by means of the sensitivity analysis method, and the mathematical model was established to optimize the problem. After optimization, the first-order vertical bending frequency value was calculated to be 8.35Hz, which was further increased by 5.9%.

(4) The weight of the car body was increased from 35.18t to 35.64t based on layout optimization of equipment under vehicle and structure optimization, which was increased by 1.3%. At the same time, the first-order vertical bending frequency of car body in the preparation conditions was increased form 7.63Hz to 8.35Hz, which was increased by 9.4%. Although it did not meet the requirement that the first-order vertical bending frequency reached 10Hz, the purpose of optimization was achieved. And combined with the overall structure, the modal frequency will be further improved in the future.

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