Multi-objective structural optimization of the aluminum alloy subway car body based on an approximate proxy model

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Abstract
To improve the comprehensive structural performance and optimize the structural quality of the aluminum alloy subway car body, a multi-objective structural optimization method based on a response surface approximate proxy model was proposed. On the basis of the sensitivity calculation of the objective response to the thickness changes of profiles in different regions of the car body, 21 groups of plate variables with the highest optimization potential were selected for optimization. The optimization experiments were performed on the design variables, and sufficient sample points were selected for the target analysis. Based on the calculated data, second-order response surface models for the bending modal frequency, bending stiffness, and complete mass of the subway car body were constructed considering the optimized profile thickness. After verifying the accuracy of the model, it was used to replace the finite element model for optimization analysis. To achieve feasible optimization results, two schemes with the profile thickness varied in steps of 0.1 and 0.5 mm in the design space were used. After the completion of the calculation, experimental evaluation of the two optimal schemes indicated that the structural performance was consistent with the numerical results.

Keywords
Subway car body, structural performance, proxy model, multi-objective optimization, optimal schemes

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Introduction
The main body of an aluminum alloy subway car is a barrel-shaped, thin-walled structure composed of the floor, side walls, roof, and end walls. When the train is running, the car body is generally subjected to severe working conditions and must have sufficient structural performance to meet the specified standards. The car body structure is an infinite vibration system with many degrees of freedom, and low-frequency time-varying excitation primarily exists on the running line. The car body resonates when the external excitation frequency is close to the natural frequency of the system. The resonance not only generates noise but also causes early fatigue damage of the structural components as well as destroys the protective layer and tightness of the car body surface. Therefore, improving the natural modal frequency of the car body and avoiding external low-frequency vibration are of practical significance. In addition, certain requirements for the stiffness, especially the bending stiffness of the car body, need to be established considering the irregularity of the train line and the large passenger flow during the peak period. For the consideration of lightweight design, the car

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body stiffness should not be improved at the cost of a substantial increase in the structural quality, otherwise the operating cost will increase, causing various adverse effects.

Extensive research has been conducted on the optimization of the overall performance objectives of the various structures. Schmidt et al.\textsuperscript{1} selected a new material system for topology optimization and proposed lightweight design with the complete mass as the objective function, which provided optimization results. Engin et al.\textsuperscript{2} set the minimum deflection as the objective function and the rectangular cross-section size of the beam element as the design variable and obtained smaller deflection after optimization. In view of the low first-order vertical bending frequency of rail vehicles, You et al.\textsuperscript{3} significantly increased the first-order bending mode frequency of the whole car body by optimizing the thickness of the main section and the elastic hanging parameters of the equipment under the car. Aiming at the aluminum alloy car body of a high-speed train, Gao et al.\textsuperscript{4} of Dalian Jiaotong University analyzed the sensitivity of stress and displacement of key parts on the basis of finite element analysis of the static strength and the modal characteristics, which guided the lightweight research; consequently, the weight of the car body structure was reduced by 6.64%. Yu et al.\textsuperscript{5} of Changchun University of Technology performed a static analysis of three working conditions of the truck frame and then optimized the truck sizes in Optistruct. Yu et al.\textsuperscript{6} studied the effect of the change in the thickness of the structure of the high-speed train car body on the modal characteristics of the car body; they accurately selected the plate parts, which contributed considerably to the bending and torsion modal frequencies of the car body. Thus, they provided a practical basis for the dynamic optimization of the car body structure. Wang et al.\textsuperscript{7} established the finite element model of the whole car body of a 160 km/h electric locomotive and analyzed its static strength performance. By adopting the static strength performance that met the relevant standards as the constraint and the minimum total mass of the car body structure as the optimization objective, they established the final optimization scheme that achieved the lightweight effect. In general, many studies have optimized the small and simple parts of the car body structure considering multiple objectives. However, large and complex overall structures have been optimized by considering only a single performance index under various constraints and with limited number of objectives. This shortcoming must be overcome by providing an optimal theoretical solution with a relatively simple operation logic.

To rapidly determine the optimal solution of multiple objectives of the research object, many studies focus not only on the research and application of multi-objective methods,\textsuperscript{9,10} but also on the improvement of the optimization process efficiency. Because of the complexity of the design space, the manual selection and verification method based on the experience of personnel is time-consuming and laborious, and the accuracy and the optimization effect of the method are insufficient. In recent decades, the response surface method (RSM),\textsuperscript{11–13} hierarchical kriging (HK) method,\textsuperscript{14–17} and radial basis function (RBF) method\textsuperscript{18–21} have been proposed. These methods construct a proxy model between design variables and responses\textsuperscript{22} for predicting the response value and avoiding the verification of data one by one.\textsuperscript{23} In the case of weak non-linearity and large number of data samples, the proposed response surface method has the characteristics of faster fitting and higher accuracy than the HK and RBF methods.\textsuperscript{24}

In this study, multiple optimization objectives are considered to address the limitation of single-objective optimization for the whole body of an aluminum alloy subway car. For this purpose, an optimization method based on an approximate proxy model of the response surface is proposed to construct a numerical model\textsuperscript{25,26} of the relationship of the design variables with the car body structure mass, first-order vertical bending mode, and bending stiffness. Using this fast and simple multi-objective optimization calculation, a set of optimal solutions can be obtained, and, subsequently, the feasible design scheme can be determined.

**Optimization method based on an approximate proxy model of the response surface**

In the multi-objective optimization of the car body structure, to establish the response surface model,\textsuperscript{27} the corresponding physical transfer relationship between the structural response and the design variables is replaced with a numerical model, and then a reasonable value of the independent variable is applied to determine the optimal solution of the objectives. Design of experiments (DOE) provides raw data for the establishment of an approximate proxy model required for the analysis of the response surface. Figure 1 shows the flow chart of the optimization method based on the response surface approximate proxy model.

**Simulation analysis of car body structure performance**

The car body was made of a large, hollow, closed aluminum alloy extruded profile with the total mass being 22.07 t and the aluminum structure mass being 6.8 t. The car body structure was discretized, and the final finite element model of the whole car body reflecting the force transfer relationship of the structure was
established on the basis of appropriate geometric model simplification and cleaning. The local mesh model is shown in Figure 2.

Based on the structural performance of the car body required to meet the relevant standards, the strength, bending stiffness, and inherent frequency modes of the car body were numerically calculated. The results show that under all load analysis conditions, the static strength and fatigue strength of the car body structure could meet the standard requirements, and the safety margin was large. Under the over-staff condition, the vertical deflection of the middle point of the side beam of the car body underframe was 9.22 mm (Figure 3(a)), which was less than 1% (15.7 mm) of the fixed distance of the vehicle, and the bending stiffness met the design requirements. For the natural mode of the car body in the complete mass state, the first-order bending mode frequency was 9.73 Hz (Figure 3(b)), which was lower than the recommended value (10 Hz) of railway vehicles in China.

Based on the calculation results of the structural performance, the first-order bending mode frequency of the whole car body was taken as the optimization object. Considering their importance in improving the comprehensive performance of the car body, the bending stiffness and structural quality of the car body were also listed as the optimization objects.

**Multi-objective optimization design of car body**

**Selection of design variables**

The car body structure was subdivided, and a total of 30 sets of plate thickness of the roof, floor, side walls, underframe side beam, inner and outer end walls, and other parts were taken as the initial design variables (Table 1).

The relative sensitivity of 30 design variables under multiple objectives were calculated, and the results are shown in Figure 4.

Based on the optimization efficiency maximization criterion, 21 design variables with the highest optimization potential, namely variables 1–20 and 26, were selected from the 30 variables for performing optimization. These 21 variables were mainly located at the car body...
bottom frame, side wall, and connection between the roof and the side wall. However, the sensitivity data of the panel at the end of the car body indicated that it had a small or even negative effect on the bending mode and flexural stiffness; therefore, it was excluded as an optimization variable.

**Optimization model determination**

The optimization model consists of three elements: design variables, constraint conditions, and objective functions. The multi-objective optimization problem can be mathematically expressed as follows:

**Table 1.** Main design variables of car body. Units: mm.

| Parts       | Number | Description of the structure          | Thickness | Parts       | Number | Description of the structure          | Thickness |
|-------------|--------|---------------------------------------|-----------|-------------|--------|---------------------------------------|-----------|
| Chassis     | 1      | Floor                                 | 2.8       | Roof        | 16     | Connecting plate 1 with the side wall | 3.0       |
|             | 2      | Reinforcement plate of the floor      | 2.5       | Reinforcement plate of the side beam | 17     | Connecting plate 2 with the side wall | 3.0       |
|             | 3      | Reinforcement plate of the side beam  | 4.0       | Reinforcement plate of the side beam | 18     | Connecting plate 3 with the side wall | 3.0       |
|             | 4      | Connecting plate between the floor and the side beam | 3.0       | Connecting plate ribs with the side walls | 19     | Connecting plate ribs with the side walls | 2.5       |
|             | 5      | Outer plate of the side beam          | 4.5       | Dome of the panel | 20     | Dome of the panel                     | 2.8       |
|             | 6      | Inside plate of the side beam         | 7.0       | Side plate 1 of the flat top | 21     | Side plate 1 of the flat top          | 2.2       |
| Side walls  | 7      | Upper panel ribs                      | 2.5       | Side plate 2 of the flat top | 22     | Side plate 2 of the flat top          | 3.0       |
|             | 8      | Lower panel ribs                      | 2.5       | Middle panel of the flat top | 23     | Middle panel of the flat top          | 2.4       |
|             | 9      | Upper panel 1                         | 2.8       | Side plate rib plate of the flat top | 24     | Side plate rib plate of the flat top  | 2.2       |
|             | 10     | Upper panel 2                         | 6.0       | Middle panel ribs of the flat top | 25     | Middle panel ribs of the flat top     | 2.2       |
|             | 11     | Upper panel 3                         | 5.0       | Junction of the vertical plate between the flat top and the dome | 26     | Junction of the vertical plate between the flat top and the dome | 8.0       |
|             | 12     | Upper panel 4                         | 3.5       | Door beam panels | 27     | Door beam panels                      | 3.5       |
|             | 13     | Middle panel                          | 2.8       | Outer panel of the threshold | 28     | Outer panel of the threshold          | 4.0       |
|             | 14     | Lower panel                           | 2.8       | Inside panel of the threshold | 29     | Inside panel of the threshold         | 5.0       |
|             | 15     | Connection of the plate with the bottom frame | 6.0       | Joint plate flanging the side wall | 30     | Joint plate flanging the side wall    | 4.0       |

**Figure 3.** Finite element calculation results: (a) vertical displacement of the bottom frame side beam and (b) first-order vertical bending mode of the car body.
\[
\min \ y = [f_1(x), f_2(x), \ldots, f_n(x)] \\
\text{s.t. } g_i(x) = 0 \\
i = 1, 2, \ldots, m; \ x_1, x_2, x_3, \ldots, x_j \geq 0 \\
\]

where \( n \) represents the number of objective functions, \( i \) represents the number of constraint functions, and \( j \) represents the number of design variables.

The optimization objective is the first-order vertical bending mode frequency (\( \text{bend\_fre} \)), the bending stiffness (\( \text{bend\_stiff} \)), and the car body mass (\( \text{mass} \)) in the maximum mass state. The corresponding optimization mathematical model is

\[
\min \ y = [-\text{bend\_fre}, -\text{bend\_stiff}, \text{mass}] \\
\text{s.t. } x_L < x_i < x_H \quad i = 1, 2, 3, \ldots, n \\
\]

where \( x_L \) represents the lower limit of the thickness of the design variable, and \( x_H \) represents the upper limit of the thickness of the design variable. According to practical experience, the thickness of the 21 optimization variables can vary from \( x_i + 1.5 \text{ mm} \) to \( x_i - 1.5 \text{ mm} \) during the optimization process.

**Latin hypercube experimental sampling**

Sampling data are the basis for the establishment of the proxy model. The sample points should not be redundant and should accurately represent the design space.31,32 The Latin hypercube method is a stratified random sampling method and has high sampling efficiency when sampling bias is not involved in the design space. If \( S \) is the number of sample points and \( N \) is the number of variables, the following condition holds:

\[
S \geq \frac{(N + 1)(N + 2)}{2} \\
\]

By using the Latin hypercube method, 253 groups of sample points were automatically extracted in the dimension space composed of 21 variables to calculate their corresponding first-order vertical bending modes, flexural stiffness, and vehicle body mass. The results are shown in Table 2.

**Establishment and comparison of approximate proxy model**

Because it is impossible to directly judge which method has high precision, an approximate proxy effects of first-order vertical bending frequency, bending stiffness, and car body mass under the RSM, HK method, and RBF method33,34 are investigated on the basis of the sampling analysis data. For the RSM, the least squares regression (LSR) method and the moving least squares method (MLSM) are selected to verify the precision.

The coefficient of determination \( R^2 \), which quantifies the accuracy of each prediction model, is calculated as follows:

\[
R^2 = 1 - \frac{\sum_{i=1}^{n} [y_i - \bar{y}]^2}{\sum_{i=1}^{n} [y_i - \bar{y}]^2} \\
\]

Figure 4. Sensitivity results: (a) sensitivity of first-order vertical frequency relative to mass and (b) sensitivity of flexural stiffness relative to mass.
where $y_i$ represents the actual response value of the $i$th sample point, $\bar{y}_i$ represents the fitting prediction value at the $i$th point, and $\bar{y}$ represents the average value of the actual response. The closer the value of $R^2$ is to 1, the better is the quality of the fitting. In practice, a value greater than 0.8 is considered to represent a high fitting precision of the model. The $R^2$ values of the response objective approximate proxy models established using the four methods are shown in Table 3.

As Table 3 shows, the optimized values of the three targets were obtained by the LSR method, with $R^2$ being the closest to 1; this was followed by the HK method and the radial basis method. The MLSM had the worst fitting accuracy, although it was greater than 0.8. To sum up, the LSR method can be used to establish the response approximate model for the three optimization objectives. To avoid low-order inadequate fitting and high-order overfitting, the recommended order of fitting is 2. Furthermore, the second-order response surface approximate proxy model can be generated as follows:

$$f(x) = \beta_0 + \sum_{i=1}^{n} \beta_i x_i + \sum_{i=1}^{n} \beta_{ij} x_i x_j$$

(5)

where $f(x)$ is the approximate objective, $\beta$ is the undetermined constant, and $x$ is the optimization variable.

### Precision verification of approximate proxy model

In terms of the fitting quality of sample points, the second-order response surface approximate proxy model fitted by the LSR method showed the best effect. To further verify the accuracy of the proxy model for the global design, 20 groups of new sample points were randomly selected and their calculated values were compared with the second-order response surface model fitting values, as shown in Figure 5.
Based on validation, the errors of the second-order response surface proxy models remained within a small range. Considering the nonlinear numerical calculation for the modal, the LSR method was used to fit the model. Although the error was relatively larger with LSR compared with other methods, the overall accuracy was stable and reliable; therefore, subsequent optimization was based on this proxy model.

**Multi-objective optimization calculation**

For the second-order response surface approximate proxy model, a global response search method (GRSM) was used to optimize the calculation. The GRSM was optimized with some random design points initially; then, the response surface model was updated in real time with some new design points that were generated in the global space during the process of calculation. From local to the whole, the GRSM optimization range expanded step by step.

The ideal optimization objectives were to maximize the first-order vertical bending mode frequency of the vehicle body, minimize the vertical deflection of the midpoint of the side beam of the underframe, and minimize the car body mass in the maximum mass state. The GRSM algorithm iteratively searches for the ideal objectives. To meet the requirements of manufacturability and ensure the search depth and breadth of the algorithm, the design variables were discretized. The optimization steps were set to 0.1 mm (depth) and 0.5 mm (breadth), and the number of iterations was set to 50,000. After the calculation, the optimal scheme was selected from the optimized data set as shown in Table 4.

The plate thickness designs with step lengths of 0.1 and 0.5 mm were denoted as scheme 1 and scheme 2, respectively. The calculation results of the approximate proxy model under the two schemes were extracted, and the car body structure under the two schemes was updated for the finite element analysis. The calculated values of the approximate proxy model were compared with those of the finite element numerical solution, as shown in Table 5.

After the calculation of the updated structure, the dimension design with scheme 1 finally obtained the first-order vertical bending mode frequency of the maximum car body mass as 10.14 Hz, which is 0.41 Hz higher than that of the initial structure (9.73 Hz), providing a 4.2% increase. The midpoint deflection of the side beam of the body bottom frame was optimized to 8.39 mm, which was 0.83 mm lower than the initial structure (9.22 mm), providing a 9.0% reduction; this further improved the bending stiffness of the body. The body full mass under scheme 1 was 22.06 t, which was 10 kg less than that of the original structure (22.07 t).
After the calculation of the updated structure, the dimension design with scheme 2 finally obtained the first-order vertical bending mode frequency of the maximum car body mass as 10.15 Hz, which was 0.42 Hz higher than that of the initial structure, providing a 4.3% increase. The midpoint deflection of the side beam of the car body underframe was 8.25 mm, which is 0.97 mm lower than that of the initial structure, providing a 10.5% decrease. The vehicle body mass under scheme 2 was 22.03 t, which is 40 kg less than that of the original structure.

Under both schemes, the car body’s first-order bending modal frequencies were above 10 Hz, which helped the modal performance evaluation qualify the requirement. Furthermore, the schemes reduced the body mass under the maximum mass state, thereby significantly improving the flexural rigidity of the bodywork. However, by comparison, the comprehensive accuracy of scheme 2 was better than that of scheme 1 and the requirements for production and manufacturing with scheme 2 were lower; therefore, scheme 2 was determined as the best optimization scheme.

Finally, the static strength and fatigue strength of the car body structure under the two schemes were analyzed and calculated under all working conditions, and it was confirmed that the checking results of structure performance were all qualified.

## Conclusion
The main results of the study are as follows:

1. Based on the dynamic and static structural performance analyses of the subway car body, the optimization objects were determined, namely the first-order vertical bending mode frequency, bending stiffness, and structural quality of the car body. According to the sensitivity analysis, 21 design variables with high optimization potential for the car body were determined.

2. Based on the experimental sample point calculation data, the second-order response surface approximation model of the three optimization objectives were established by using the least squares ratio method. The accuracy verification showed that the error of the approximate proxy model was small and met the operational requirements.

3. Multi-objective optimization calculation of the car body was performed with the approximate proxy model and the global response search algorithm. Using the final optimization scheme, the first-order vertical bending mode frequency of the whole car body reached the recommended value of railway vehicles. At the same time, the midpoint deflection of the side beam of the car body underframe was reduced, further improving the bending stiffness, and the mass of the car body was also reduced. Compared with the original structure, the thickness of the optimized plate was optimized in a step size of 0.5 mm, which enabled better manufacturability.

4. The static strength and fatigue strength analysis on the basis of multi-objective optimization showed that the performance of the optimized car body satisfied the design requirements.

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