Research on lightweight design of power battery cabin in electric vehicle

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Abstract: In this paper, according to the design requirements of the battery cabin and the structure of the frame, the finite element model is established. The static analysis and constrained modal analysis are carried out for the battery cabin under two extreme conditions of bumpy sharp turn and bumpy emergency braking, and the first six modal frequencies and modes are obtained. The optimized battery cabin achieves the goal of 2\% lightweight optimization design, and other performances are satisfied. After the topography optimization, the endurance and rigidity of the battery cabin are improved to varying degrees. The elastic limit of the material is much higher than the maximum stress. The weight of the battery cabin is reduced by 2.5\% through topology design.

1. preface

As we all know, in all the most important structures of electric vehicles, only the power battery cabin is one of the most important parts of the whole vehicle as the protection and load-bearing device of the power battery of electric vehicles. The strength and stiffness of battery compartment structure, space design and connection technology all have an important impact on vehicle performance \cite{1}. Therefore, under the premise of achieving all the basic conditions, the lightweight optimization design of the power battery cabin structure has become the top priority.

This paper is based on the design code of the battery cabin and the size of the frame, the three-dimensional model of battery cabin was established, in this paper, a three-dimensional model of the battery cabin is established based on the building specifications of the battery cabin and the size of the frame, then, according to the requirements of the finite element model, the simplification, geometric reasoning, mesh generation and quality control of the model are carried out, finally, the finite element model of the battery bin is obtained. After that, static analysis and modal analysis are carried out under the extreme conditions of bumpy sharp turn and bumpy emergency braking, and the first six modal frequencies and vibration modes of the cell of the battery cabin are calculated, and their decomposition and evaluation are carried out. Based on the static and dynamic characteristics and modal analysis results of the battery cabin, the shape optimization of the upper cover and lower plate of the battery cabin and the topology optimization based on the optimization of longitudinal and transverse beams are optimized to reduce the weight of the battery cabin and realize the design requirement of 2\% lightweight.
2. Construction of finite element model of battery cabin
The overall external plane dimension of certain vehicle battery cabin is 1860.700mm × 740.000mm × 166.581mm. There are 8 battery packs in the lower box of the battery cabin, with 12 cells in each group. The standard voltage of the battery pack is 360V. The eight battery packs can be assembled and disassembled independently. The power battery assembly of battery cabin is 230kg, and the weight of battery cabin is 38.87kg. In order to get high-quality grid, the components which have little effect on the overall performance are eliminated and the battery cabin is simply treated[2]. The structure line of the battery cabin is simulated by using the plate and shell element, and the hexahedron element with strong convergence-confinement method and high calculation accuracy is used to simulate the battery cell. The accuracy and velocities of calculation are summarized, the average mesh size is 5mm. The upper cover and the lower cover of the battery compartment are connected by riveting simulation, and the lower cover and bracket are connected by welding simulation. Rigid element is used in riveting, ACM element is used in welding. Each node between hexahedron element and welding surface element is connected flexibly by RBE3 to transfer torque. 6082-T6 aluminum alloy is used as the material for the battery cabin structure.

According to the above steps, the FEM model of battery cabin structure is constructed in HyperMesh, as shown in Fig.1.

3. Performance analysis of battery cabin
3.1.Static analysis
When solving statics problems by FEM, the basic governing equations are used[3].

(1) Analysis of emergency braking condition on bumpy road
The simulated battery unit is connected to the vehicle chassis by setting all the motion and rotation degrees of freedom in the center of the support bolt hole connected to the chassis. The mass of power battery assembly is 230kg. The vertical acceleration caused by turbulence is 2g, and the longitudinal acceleration caused by emergency braking is 1g. The gravity of the battery shall be applied to the corresponding position in the form of an average load.

The stress and strain under emergency braking on bumpy road are shown in Fig. 2 respectively, the maximum displacement is 0.85 mm, which appears in the middle of the lower box, which is within the allowable range. The elastic limit of 281 MPa is greater than the maximum stress of 65.6 MPa, which is mainly distributed in the battery cabin bracket, so it has high safety and reliability. It can be seen that the battery cabin can meet the requirements of the anti-collision strength of the battery cabin under the condition of bumpy emergency braking.
3.2. Modal analysis of battery cabin

Since there is prestress in the connection, the first six modal frequencies of the battery cabin are calculated by using the finite element analysis program. The first six modal frequencies of the battery bin are listed in Table 1. The modal vibration modes are shown in Fig. 4 to Fig.6.

| Order | Fluctuation frequency | Mode shapes                      |
|-------|-----------------------|----------------------------------|
| 1     | 26.5                  | wagging                          |
| 2     | 27.8                  | Local mode of base plate         |
| 3     | 36.8                  | Local modes at the rear of the base plate |
| 4     | 40.2                  | Local mode of roof cover         |
| 5     | 44.6                  | First order torsion              |
| 6     | 46.3                  | Local mode of base plate         |
3.3. Results analysis and evaluation

The primary excitation source of electric vehicles is the fluctuation of motor and the fluctuation caused by surface irregularity. Generally, the excitation frequency of the motor is less than 25Hz, while the ground excitation frequency is related to the road condition. The relationship between road excitation frequency, vehicle speed and road wavelength is [5]:

\[ f = \frac{V_{\text{max}}}{L_{\text{min}}} \times 3.6 \]  

In formula 1, it is the fastest speed of electric vehicle, and is the distance between the displacement of road surface roughness and time balance. Table 2 lists the distance between two particles under different uneven road conditions.

| pavement          | macadam         | Flat road | Unpaved pavement | Washboards |
|-------------------|-----------------|-----------|------------------|------------|
| distance          | 0.32-6.3        | 1.0-6.3   | 0.77-2.5         | 0.32-6.3   |

When driving outdoors on gentle road conditions, the maximum speed is set at 100km / h, and the minimum distance between two particles is 1.0 under uneven road condition. When the data is brought into equation 3-5, the following results can be obtained:
To sum up, the first order fixed frequency of the battery cabin in this paper is 26.5hz, which is slightly less than 30Hz to avoid large amplitude vibration, which may cause large amplitude vibration and fatigue disturbance. Therefore, in order to improve the anti-jamming ability of the battery cabin, it is necessary to optimize the design of the top cover and the local fluctuating substrate.

4. Optimization design of battery cabin structure

Through the research and analysis of the dynamic and static characteristics of the battery cabin in the previous section, it is found that the first-order natural frequency of the battery cabin is less than 30 Hz, so it is necessary to topography optimization. Combined with the analysis of dynamic and static characteristics before, according to the manufacturing process of the board, the cover design of the area is to cover the cover surface to the maximum extent, which must include the whole substrate, battery and base plate design, so as to facilitate the configuration of the whole battery pack. The shape objective function is optimized to select the maximum limit value of the first order fixed frequency response of the battery cabin. The mathematical model of morphology optimization can be expressed as follows:

\[
f = \frac{V_{\text{max}}}{L_{\text{min}} \times 3.6} = \frac{100}{1 \times 3.6} = 27.78Hz
\]

Among them, \( L_{\text{min}} \times 3.6 \) is the interval of element nodes moving in the normal direction in the design area; \( C \) is the flexibility of the structure, or the inverse reciprocal of the structural rigidity; \( U \) is the displacement of the node under the supported working condition; \( \) \( K(e_i) \) is the structural stiffness after the optimized element node displacement; \( D \) is the upper limit value of the given element node movement.

4.1. Performance analysis of battery cabin with topography optimization

After three iterations with Optistruct, the final constraint converges, and the optimization results of the top cover and bottom plate are obtained, as shown in Fig.7.

The finite element model of the top cover and bottom plate of the battery compartment after topography optimization is shown in Fig.8.
After the topography optimization of the battery cabin, the same stress constraint and load were applied to analyze its dynamic and static characteristics and check the effect of topography optimization. The results show that the frequency of the first six modes of the battery compartment is increased from 26.5Hz to 30.4Hz, higher than 30Hz. The stress and strain of the optimized battery
cabin under two working conditions are shown in Fig. 9 and Fig. 10, respectively. The maximum stress is 55.85 MPa and the maximum displacement is 0.71 mm under emergency braking on bumpy road. The maximum stress and maximum displacement of battery cabin are 74.59 MPa and 0.74 mm respectively, the first six modes of the optimized battery cabin are shown in Fig. 11 to Fig. 13.

Table 3 The performance indexes after topography optimization

| Performance index | Bumpy emergency braking condition | Bumpy sharp turning condition |
|-------------------|----------------------------------|-------------------------------|
|                   | Maximum stress | Maximum displacement | Maximum stress | Maximum displacement |
| Before optimization | 65.64 MPa | 0.85 mm | 86.04 MPa | 0.83 mm |
| After optimization | 55.85 MPa | 0.71 mm | 74.59 MPa | 0.74 mm |
| Relative variation | -9.79 % | -0.14 % | -11.45 % | -0.09 % |
| Rate of change | -14.9% | -16.5% | -13.3% | -10.8% |

Table 4 Fluctuation frequency before and after optimization

| Order | Fluctuation frequency before optimization | Fluctuation frequency after optimization | Variation value |
|-------|-----------------------------------------|----------------------------------------|-----------------|
| 1     | 26.5                                    | 30.4                                   | +3.9            |
| 2     | 27.8                                    | 32.3                                   | +4.5            |
| 3     | 36.8                                    | 39.7                                   | +2.9            |
| 4     | 40.2                                    | 44.7                                   | +4.5            |
| 5     | 44.6                                    | 49.2                                   | +4.6            |
| 6     | 46.3                                    | 50.4                                   | +4.1            |
Through the above analysis, we can see that the strength and stiffness of the battery cabin after morphology optimization have been improved to varying degrees. The yield strength of the material is far greater than the maximum stress, and has a high safety factor. The maximum displacement is lower than the maximum displacement of the original battery cabin. Therefore, the battery cabin with topography optimization has great potential for lightweight.

4.2. topology optimization

The vertical and horizontal beams of the battery cabin is an important component of the stable structure of the battery cabin. When the battery cover and the lower cover are affected by the battery module, the stress and deformation caused by the strengthened beam are mainly concentrated at the upper end of the beam, while there is almost no deformation at the side end. In the lightweight design of the battery cabin, the lateral end of the battery cabin is optimized by topology optimization design of the strengthened beam [6]. The weight of the battery cabin is reduced by topology optimization, and the weight reduction target is 2%. Considering the structure of the battery cabin, the main purpose is to optimize the topology of the battery cabin support beam, and select the battery bin support area as the research area. The objective function of topology optimization is to select the minimum value of battery cabin mass. The constraint is that the first order natural frequency response of the battery cabin is not less than 26.5Hz.

By using Optistruct, after four iterations, the final constraint convergence is obtained. Different colors represent different cell densities. The finite element model of battery bin beam after topology optimization is shown in Fig. 14.

According to the topology optimization results, based on the first order natural frequency greater than 26.5Hz, the weight of the battery cabin is reduced from 38.87kg to 37.90kg, which is 0.97kg and 2.5% lower. Therefore, the battery cabin after topology optimization achieves the goal of lightweight.

The stress and strain of battery cabin after topology optimization are shown in Fig.15 and Fig.16 respectively. Under the condition of emergency braking on bumpy road, the maximum stress and displacement of battery cabin are 65.22MPa and 0.81mm, respectively. The maximum stress and displacement of battery cabin are 85.82MPa and 0.89mm, respectively. After optimization, the first six modal vibration modes of battery cabin are shown in Fig. 17 to Fig.19.

Fig.14. Results and shape of optimized battery bin bracket

Fig.15. Analysis results of emergency braking condition on bumpy road of battery cabin after topology optimization (stress nephogram and strain nephogram)
Fig. 16. Analysis results of battery cabin after topology optimization for sharp turning on bumpy road (stress nephogram and strain nephogram).

Fig. 17. The first and second mode shapes of the optimized battery cabin.

Fig. 18. The third and fourth order vibration modes of the optimized battery cabin.

Fig. 19. The fifth and sixth order vibration modes of the optimized battery cabin.

Table 5 Performance indicators after topology optimization

| Performance index | Bumpy emergency braking condition | Bumpy sharp turning condition |
|-------------------|-----------------------------------|------------------------------|
|                   | Maximum stress | Maximum displacement | Maximum stress | Maximum displacement |
| Before optimization | 65.64 MPa 0.85 mm | 86.04 MPa 0.83 mm |  |
| After optimization | 65.22 MPa 0.81 mm | 85.82 MPa 0.89 mm |  |
| Relative variation | -0.42% -0.04% | -0.22% +0.06% |  |
| Rate of change | -0.6% -4.7% | -0.2% 7.2% |  |
Table 6 The performance indexes after topography optimization

| Order | Fluctuation frequency before optimization | Fluctuation frequency after optimization | Variation value |
|-------|------------------------------------------|----------------------------------------|-----------------|
| 1     | 26.5                                     | 27.0                                   | +0.5            |
| 2     | 27.8                                     | 28.5                                   | +0.7            |
| 3     | 36.8                                     | 37.6                                   | +0.8            |
| 4     | 40.2                                     | 40.5                                   | +0.3            |
| 5     | 44.6                                     | 45.5                                   | +0.9            |
| 6     | 46.3                                     | 47.2                                   | +0.9            |

Through the above analysis, it can be seen that after topology optimization, the weight of battery cabin can be reduced by about 2.5% while the strength and stiffness performance remain unchanged. The lightweight optimization design goal of battery cabin is completed.

5. Conclusion
In this paper, through the shape optimization of the battery cabin, the first six order modal frequency of the battery cabin is increased from 26.5Hz to 30.4Hz, higher than 30Hz. The maximum stress and the maximum displacement of the battery cabin are 55.85 MPa and 0.71mm respectively under the condition of emergency braking on bumpy road. The maximum stress and maximum displacement of battery cabin are 74.59MPa and 0.74mm, respectively. After that, the topology optimization of the support beam is carried out. The weight of the battery cabin is reduced by 2.5% when the first natural frequency is greater than 26.5Hz. The maximum stress is 65.22 MPa and the maximum displacement is 0.81mm under emergency braking on bumpy road. The maximum stress and maximum displacement of battery cabin are 85.82 MPa and 0.89mm respectively. Compared with the original battery cabin, the weight of the optimized battery cabin has been reduced by 2.5% while maintaining its performance, and obvious results have been achieved in the optimization design.

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