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Aluminum Alloy Anti-collision Beam Assembly Multi-objective Optimization

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Abstract

Based on the strict CFVSS215 Canadian low-speed collision regulations, the 8km/h pendulum collision test conditions conducted a crash test on the anti-collision beam assembly of a medium-sized car, and found that the anti-collision beam assembly still has a large space for lightweight. Taking the thickness of each plate of the collision beam assembly as the design variable, using the Latin hypercube sampling experimental method to select 49 groups of sample points, using the relationship between the design variables and the response value to construct the radial basis function model, with the minimum mass of the collision beam assembly, the minimum maximum collision force as the objective function, and the maximum deformation as the constraint, the model is solved by the sequence quadratic programming method in the isight software, and the mass of the final collision beam assembly is reduced by 8.4%, and the maximum collision force is reduced by 17.2%.Finally,the energy absorption increased by 1.8%, indicating that this method can improve the optimization efficiency and provide a reference for the design of the collision avoidance beam assembly.

Keywords

Anti-collision Beam Assembly; Collision Regulations; Multi-objective Optimization.

1. Introduction

In the process of car collision, the anti-collision beam assembly is the first line of defense to protect the safety of the vehicle and the safety of the personnel in the car, so the research of the collision beam assembly has become a hot spot in the research of automobile safety performance. Studies have shown [1] that the accident rate of low-speed collisions is much higher than the probability of high-speed collisions, so it is of positive significance to study the low-speed collision performance of anti-collision beam assembly.

Park[2] et al. have designed beams and energy-absorbing boxes, and numerically simulated the anti-collision beam assembly according to the experimental working conditions of FMVSS581 in finite element software, and optimized their performance indicators. Zhengwei Zhang [3] was use of the requirements of the European ECE-R42 low-speed collision regulations,in addition the finite element simulation model of the bumper was established, and the simulation analysis was carried out, and the variation law of each performance index was obtained. Wenshun Huang [4] based on the three experimental working conditions of the two low-speed collision regulations, and established a finite element model of low-speed collision and explained that the collision avoidance beam assembly meets the collision requirements. Optimization of the thickness of each part of the anti-collision beam assembly can also greatly improve the crash performance and safety performance of the collision beam assembly [5,6,7].

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In this paper, the finite element modeling of the anti-collision beam assembly of a medium-sized car is carried out, and the collision beam assembly is simulated and analyzed. According to the latin hypercube sampling experimental method, the sample point is selected, and the radial basis function model (RBF) proxy model is established by using the relationship between the design variables and the response of the sample point, and the optimization algorithm is used to solve, and the final anti-collision beam assembly achieves the lightweight effect under the premise of improving the performance index.

2. Establishment of Finite Element Model and Collision Theory

2.1 Establishment of Finite Element Models

The anti-collision beam of a medium-sized car is always a polyline-shaped anti-collision beam, and a low-speed collision model is established according to the size parameters of the collision beam assembly and the pendulum size parameters in the regulations, of which the material of the energy-absorbing box is AA6063-T4, the material of the beam is AA6082-T6, and the material parameters of each component are shown in Table 1, at this time the total mass of the collision-proof beam assembly is 3.77kg. The geometric model of the collider and anti-collision beam assembly is shown in Fig 1 and 2. Since the aluminum alloy material is non-strain rate sensitive material, the effect of strain rate effects is not considered in the low-speed collision model.

The anti-collision beam assembly of the front end of the car is simplified in accordance with the crash requirements of the car collision regulations. The bottom end of the energy-absorbing box in the collision avoidance beam assembly is constrained by six degrees of freedom, giving the pendulum vehicle an overall weight of 1240kg, and the pendulum collides the collision beam assembly at the speed of the regulations. In the Abaqus finite element software, the anti-collision beam assembly is meshed, and the weld spot connection is used between the energy absorbing box and the beam, and since the thickness of the energy absorption box and beam is much smaller than its length and width, the element type taken is S4R. The pendulum is a non-deformed body during the collision, so it is set to a rigid body. The contact condition uses the common contact pattern that comes with the software, using the penalty function to define the friction, which is set to 0.2. Since the unit type of this model adopts a reduced integration unit, which improves the calculation efficiency, there will be hourglass energy generated during the calculation process, and the proportion of hourglass energy to the total energy should be controlled below 5% to ensure the credibility of the model [8]. The established finite element model is shown in Fig 3.

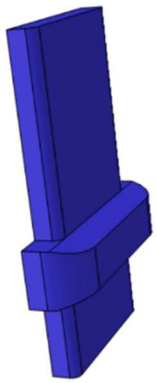

Fig 1. Pendulum geometry model

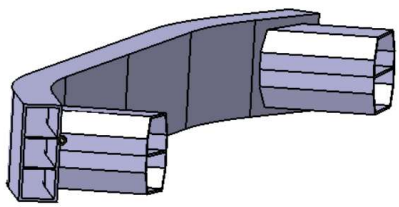

Fig 2. anti-collision beam assembly geometry model

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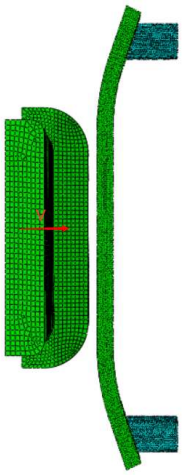

Fig 3. Collision finite element model

Table 1. Material parameters of each component

Material parameter	Obsorber	Beam
Density/kg·mm^{-3}	2.7×10^{-6}	2.7×10^{-6}
Elasticity modulus /GPa	70	70
Poisson's ratio	0.28	0.28
Yield strength /MPa	198	395

2.2 CollisionTheory

Implicit solution methods are sometimes used to iterate in finite element software, but convergence is often not guaranteed for highly nonlinear problems such as collisions. In the Abaqus/Explicit software, the problem of collision is solved by explicit methods, and the equations are:

$$M\ddot{x}_t + C\dot{x}_t + Kx_t = P_t \tag{1}$$

The center difference method is used to solve, the time point is discretized, and the time step Δt is used, and the t moment structural velocity $\dot{x}_t$ and acceleration $\ddot{x}_t$ are approximated.

$$\dot{x}_t = \frac{1}{2\Delta t}(x_{t+\Delta t} - x_{t-\Delta t}) \tag{2}$$

$$\ddot{x}_t = \frac{1}{\Delta t^2}(x_{t+\Delta t} + 2x_t - x_{t-\Delta t}) \tag{3}$$

Bringing $\dot{x}_t$, $\ddot{x}_t$ into the equations of motion yields:

$$\left(\frac{M}{\Delta t^2} + \frac{C}{2\Delta t}\right)x_{t+\Delta t} = P_t - \left(K - \frac{2M}{\Delta t^2}\right)x_t - \left(\frac{M}{\Delta t^2} - \frac{C}{2\Delta t}\right)x_{t-\Delta t} \tag{4}$$

Wherein: M represents the mass matrix, C represents the damping matrix, K represents the stiffness matrix, and P_t represents the external force matrix.

From the above equation, it can be seen that according to the x_t, $x_{t-\Delta t}$ can be found $x_{t+\Delta t}$.

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3. Initial Anti-collision Beam assembly Simulation Analysis

Draw a deformation cloud map and energy change curve of the collision avoidance beam assembly. Fig 4 shows the deformation response of the anti-collision beam assembly at 0.048s under working conditions, and the deformation of the collision beam assembly at this time is the largest, corresponding to the maximum deformation occurring in the middle of the beam, and the maximum deformation is 65.25mm.

As can be seen from Fig 5, the total energy during the collision process is basically maintained at 3062.4J, indicating that the energy conservation of the whole process is in line with the law of energy conservation. In the final stage of the simulation, the hourglass energy can be obtained 15.46J, the proportion of hourglass energy is 0.5%, and the hourglass energy that meets the requirements can be controlled below 5%, indicating that the model is credible.

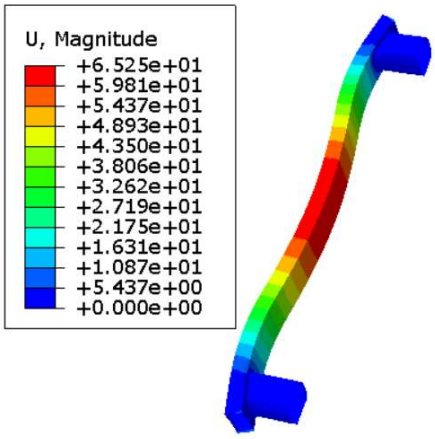

Fig 4. Maximum deformation diagram of the anti-collision beam assembly

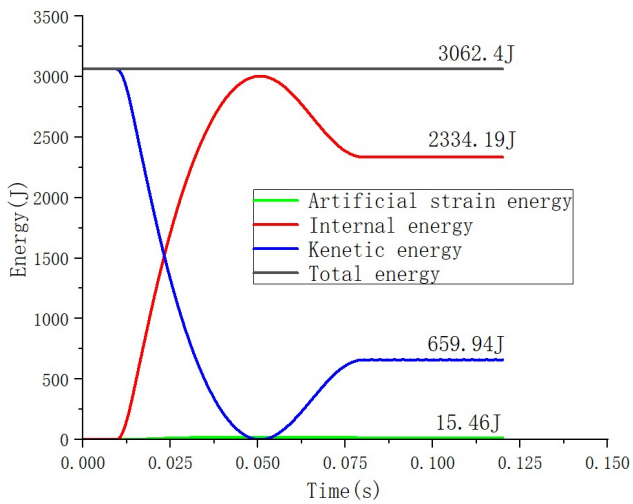

Fig 5. Energy change curve of collision avoidance beam assembly

The distance from the beam of the collision beam assembly to the front tank device for the collision avoidance beam assembly of this car is 80mm, and the maximum intrusion of the collision beam assembly under such regulatory conditions is 65.25mm. It can be seen that there is a large lightweight design margin in the anti-collision beam assembly, and its performance is improved by improving the thickness of each component of the anti-collision beam assembly.

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4. Establishment of RBF Approximate Model

4.1 Design Variables

The design takes the thickness of each plate of the anti-collision beam assembly as the design variable, and its design variables are shown in the Fig 6, of which x1 to x8 are the inner ribs of the energy absorption box of the anti-collision beam assembly, the outer plate of the energy absorption box, the upper plate of the beam, the lower plate of the beam, the back plate of the beam, the front plate of the beam, the upper rib of the beam, and the lower rib of the beam. The range of values for the thickness of each part is shown in the table 2.

Table 2. Range of values for design variables

Design valuable	Initial value/mm	Lower limit/mm	Upper limit/mm
x_1	2.2	2.0	2.6
x_2	2.2	1.8	2.6
x_3	2.8	2.2	3.4
x_4	2.8	2.2	3.4
x_5	3.0	2.4	3.6
x_6	3.0	2.4	3.6
x_7	1.4	1	3
x_8	1.4	1.2	1.55

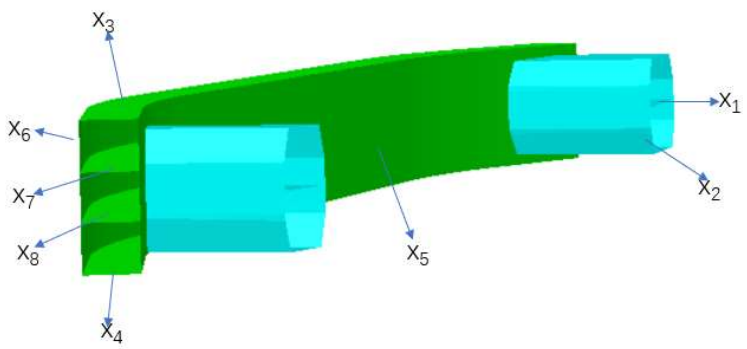

Fig 6. Schematic diagram of the anti-collision beam assembly assembly

4.2 Experimental Design

When fitting the approximate model, the optimal Latin hypercube experimental method is used to randomly sample each variable, which can greatly save computational costs. A total of 49 sets of sample points were used in this fitted approximation model, and they were simulated, and the sample points and calculation results were shown in the following table 3. In the text, F represents the maximum collision force, and M represents the mass of the anti-collision beam assembly. On the other hand, U represents the maximum deformation of the anti-collision beam assembly.

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Table 3. Optimal Latin hypercube sample points Design valuable unit:mm

Number	x_1	x_2	x_3	x_4	x_5	x_6	x_7	x_8	F/KN	M/kg	U/mm
1	2.0	1.796	2.285	2.331	2.939	2.816	1.286	1.277	65.128	3.3	74.470
2	2.012	1.939	3.040	2.377	2.669	3.110	2.796	1.528	80.514	3.62	66.751
3	2.024	2.371	2.971	2.880	2.669	3.477	1.653	1.483	79.007	3.71	66.950
4	2.037	1.849	3.268	3.154	3.208	3.135	1.041	1.506	74.454	3.68	68.55
5	2.049	2.442	3.063	3.108	3.159	3.282	1.163	1.266	75.638	3.76	67.750
.	.	.	.	.	.		.	.	.	.	.
.	.	.	.	.	.	.	.	.	.	.	.
.	.	.	.	.	.	.	.	.	.	.	.
47	2.576	2.227	2.263	3.085	2.865	2.939	2.306	1.346	81.002	3.61	65.918
48	2.588	2.191	2.743	2.903	2.621	2.841	2.592	1.437	80.533	3.58	66.363
49	2.6	2.550	2.651	2.949	3.012	2.450	1.898	1.534	79.195	3.57	67.085

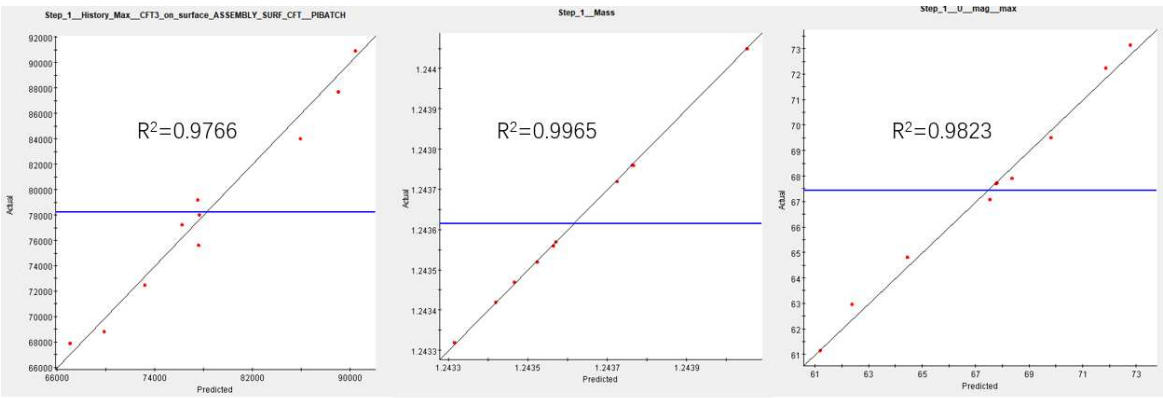
4.3 Establishment of an RBF Approximate Model

In order to improve the subsequent optimization efficiency, the relationship between the independent variables and the response is fitted, and radial basis function model (RFB) is constructed instead of the simulation model. The test for the model can be tested by the coefficient of determination R2. The closer R2 is to 1, the more accurate the model is fitted, and the expression for R2 is:

$$R^2 = \frac{\sum_{i=1}^{n}(\hat{y}_i - \bar{y})^2}{\sum_{i=1}^{n}(y_i - \bar{y})^2} \tag{5}$$

where n represents the data point used to test the accuracy of the approximate model; $\hat{y}_i$ represents the predicted value of the ith response proxy model; y_i represents the simulated value of the ith response; and $\bar{y}$ represents the mean.

From 49 sets of simulation experiments, 10 groups of experiments were randomly selected to test the approximate model, as shown in the Fig 7, the detection coefficient R2 of each response value is close to 1, indicating that the approximate model constructed is accurate enough.



(a) Force (b) Mass (c) Displacement

Fig 7. Approximate model accuracy test

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5. Optimization Design of the Anti-collision Beam Assembly

5.1 Mathematical Model

Considering the influence of the thickness of each plate of the anti-collision beam assembly on its crash performance, the anti-collision beam assembly is optimized and designed. The mathematical model can be described as:

$$\begin{aligned} \text{find } & x = (x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8)^T \\ \min \quad & \{M(x), F(x)\} \\ \text{s.t.} \quad & U(x) \leq [U] \end{aligned} \tag{6}$$

where $[U]$ represents the maximum deformation allowed for the collision avoidance beam, and the range of values for the independent variable x is the same as above.

In the isight software, the RBF approximate model is integrated with the sequence quadratic programming optimization algorithm to establish a multi-objective optimization analysis model for anti-collision beam assembly collision, and the integrated model is shown in Fig 8:

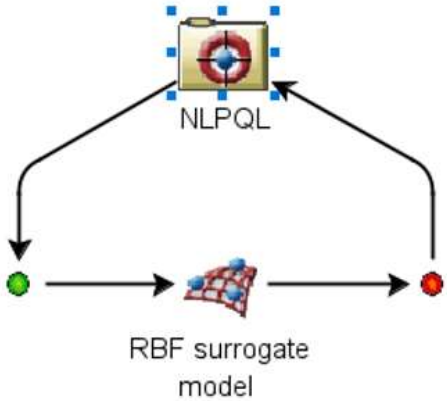

Fig 8. Integration optimization process

5.2 Optimization Results Analysis

The sequence quadratic programming method is used to iterate on the approximate model, and the optimal combination of optimized variables is obtained after 105 iterations of optimization, as shown in Table 4:

Table 4. Optimal combination of design variables after optimization

x_1	x_2	x_3	x_4	x_5	x_6	x_7	x_8	F/KN	M/kg	U/mm
2.6	1.76	3.36	2.24	2.91	2.4	1	1.26	59.66	3.26	76

The finite elements of the anti-collision beam assembly are remodeled using the optimized independent variables, and they are simulated and analyzed, and then compared with the results of the optimization of the approximate model. It is not difficult to see from the table 5 that the error between the predicted value of the model and the simulated value is relatively small, which further verifies the accuracy of the established approximate model. At the same time, the performance indicators before and after the optimization are compared, as shown in Fig 9-11, the energy absorption energy of the anti-collision beam assembly after optimization is increased to 2376J, an increase of 1.8%, the maximum crash force of the optimized collision beam assembly is reduced to 61.4KN,

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which is reduced by 17.2%, and the maximum deformation of the optimized collision beam is increased to 74.19mm, but it still meets the design requirements. The quality of the optimized anti-collision beam assembly is reduced to 3.45kg, which is reduced by 8.4%, which has a good lightweight effect.

Table 5. Comparison of model predicted and simulated values

Comparative item	F/KN	M/kg	U/mm
Value about model	59.66	3.26	76
Value about simulation	61.4	3.45	74.19
Error	2.8%	5.5%	2.4%

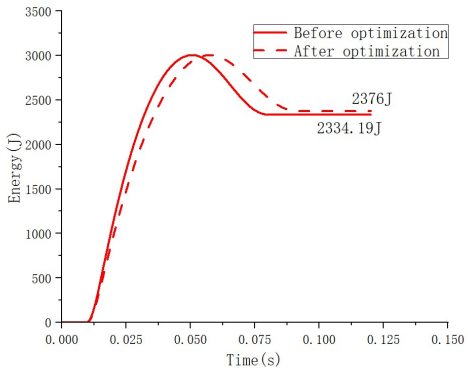

Fig 9. Comparison of energy absorption before and after optimization

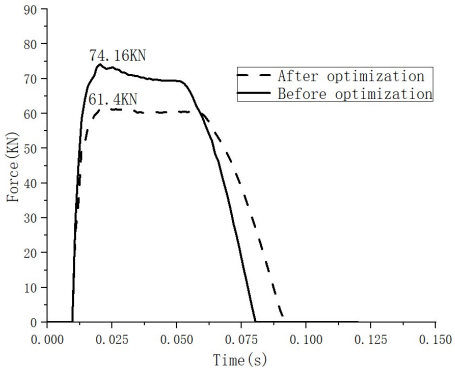

Fig 10. Comparison of maximum crash forces before and after optimization

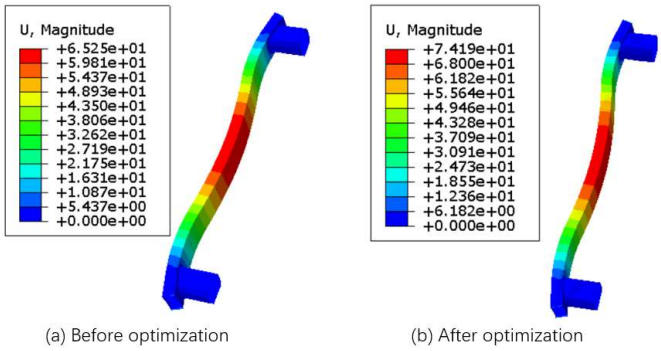

Fig 11. Comparison of the maximum deformation amount before and after optimization

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6. Conclusion

In this paper, a finite element model of a vehicle collision avoidance beam assembly is established, and it is simulated and analyzed according to the experimental conditions of the Canadian FMVSS215 crash regulations, and it is found that there is a large lightweight design margin under such conditions. The radial base neural network (RBF) proxy model is used instead of the simulation model, and the sequence quadratic programming method is optimized for the proxy model, and the final anti-collision beam assembly is reduced by 8.4%, the maximum collision force is reduced by 17.2%, and the energy absorption is increased by 1.8%, indicating that this method can improve the optimization efficiency and provide a reference for the design of the collision avoidance beam assembly.

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