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Optimal Control Method for Side Impact Safety of Vehicle Frame Structure

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Abstract

In order to verify the safety of side impact of vehicle frame structure and reduce the number of accidents of members injured due to side impact. On the basis of the basic theory of collision simulation and the theory of anti-side collision optimization design, the finite element model of side impact of the whole vehicle is established with the aid of the CAD model of a foreign mass production vehicle. With the finite element analysis software hyper mesh, the boundary and constraint conditions are set according to the requirements of GB20071-2006, so that the deformation speed and invasion amount of B-pillar in the important parts of side wall are normal. Through increasing the thickness of sheet metal, using ultra-high strength steel, adding structural strengthening parts, adopting different cross-section shape of anti-collision beam and optimizing the position of anti-collision beam, the safety of the original vehicle is verified by simulation test. The results show that different improvement methods can optimize the side impact performance of the vehicle, but the safety of the vehicle can be improved. The most effective way is to increase the reinforced parts reasonably without affecting the function of the original parts in the relevant position. The optimization method, the conclusion and the measures to improve the side impact performance provide a certain reference for the design and research of vehicle side impact safety performance optimization control.

Keywords: Automobile frame structure; Side impact; Safety optimization control;

1. Introduction

With the rapid development of economy, cars play an increasingly important role in people's daily life. However, with the rapid increase of car ownership, the number of various traffic accidents related to automobiles has risen sharply, which not only causes huge property losses, but also seriously threatens people's life safety. In all kinds of traffic accidents, the frequency of frontal and side impact is the highest. When the side impact occurs, the deformation speed and intrusion volume of the door are too large, which may cause serious injury to the passengers. Therefore, side impact is the highest form of accidents, with side impact accounting for 30% of all accidents. In automobile accidents, head and neck injuries account for 58% and trunk injuries at most injuries accounted for 32% and abdominal injuries accounted for 21% [1]. Therefore, the research on side impact safety has become an important research content in the field of vehicle passive safety, and has attracted the attention of automobile manufacturers and automobile research institutions all over the world.

In the research of automobile side impact, door and B-pillar are the key objects and important indexes to evaluate the side impact resistance of automobile. For the door and B-pillar, the displacement and speed changes of various components under the action of external force are mainly investigated, because these two indicators are directly related to the injury degree of passengers. In order to effectively reduce the injury of passengers in side impact, it is necessary to make the impact force evenly transmitted to all parts of the body, so as to ensure that the displacement and speed of the vehicle are reduced to the minimum. Automobile body is a large and complex system, and there are mutual constraints between the performance of the body. In the process of research and development of automobile body, we must comprehensively consider the interaction of multi-function and multi-disciplinary. Therefore, the impact of lightweight, NVH, dynamic performance and other disciplines should be considered in the study of vehicle side impact safety. To solve the problem that single discipline or single objective optimization method is difficult to achieve the comprehensive design of vehicle performance, multidisciplinary optimization technology was born. Multidisciplinary design optimization (MDO) technology fully considers the influence of performance, realizes multidisciplinary integration and optimization, and obtains the overall optimal design of the system.

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Based on this, this paper uses different methods to optimize and improve the crashworthiness of the side structure in side impact from different aspects, and verifies the effectiveness of these optimization methods in theory. Aiming at the problem of mutual restriction between body side impact safety and body lightweight, uniform test design is used to construct the test, and the stepwise regression surrogate model is constructed according to the test data. Finally, the multi-disciplinary collaborative optimization research is carried out by using the sequential quadratic programming method and the adaptive weighting method. Finally, the optimization results meeting the design objectives are obtained. The security is verified by computer simulation.

2. Establishment and Experiment of Side Impact Finite Element Model

It is difficult to understand the dynamic response of human body in the real accident process because of the short time and great deformation of vehicle collision. However, in order to study the safety of vehicle side impact, the major automobile companies and research institutions must present the real accident process. In the early stage, they obtained the relevant data through the real vehicle experiment, while the research on human injury was obtained by placing the corpse or dummy on the test vehicle. With the rapid development of computer technology and the gradual improvement of mathematical model, computer simulation method has been widely used in the field of vehicle collision safety, and its analysis accuracy and feasibility are higher and higher. It can not only help researchers to obtain accurate information, improve the crash safety of the vehicle, but also reduce the development cost, so that the market competitiveness of the car is stronger. It has an important guiding significance for the research of automobile passive safety.

2.1. Key technologies of finite element analysis

The method of computer simulation is to replace the real vehicle collision with computer simulation through the establishment of vehicle collision model [2]. At the cost of accurate calculation efficiency, in order to ensure the calculation accuracy and reduce the calculation scale, reasonable finite element model simplification and mesh generation become the key of finite element analysis [3]. Relevant research shows that: The accuracy and calculation efficiency of simulation analysis results are related to the element type and contact algorithm, to a large extent, it also depends on the simplification degree of finite element model, the accuracy of mesh generation and the setting of material parameters. The following focuses on the simplification of finite element model and mesh generation.

2.1.1. Simplification of finite element model

In order to describe the vehicle structure and the connection between its components, the key of computer aided design (CAD) modeling is to ensure the accuracy of the model. However, the purpose of computer aided engineering (CAE) analysis is different from that of CAD. Many processing and assembly details (such as holes, steps, stiffeners, small chamfering, etc.) in CAD often lead to unit quality degradation when meshing in CAE. The reduction or even disqualification will affect the accuracy of simulation calculation, as well as the calculation time and efficiency. Therefore, when establishing the finite element model, the details of the model built by CAD should be reasonably simplified according to the actual calculation scale and accuracy requirements.

For side impact, the thin-walled metal parts at the impact part are the main deformation energy absorbing parts, such as B-pillar, threshold beam, roof beam and door, etc. When the geometric model of the above parts is established, its geometric characteristics must be accurately reflected and not too simplified. The structural stiffness of the parts, such as engine and gearbox, which are not deformed or not deformed in the process of collision, is much larger than that of thin-walled metal parts, so they can be treated as rigid bodies. For some non main body load-bearing parts, although the absorption of collision energy is very small, it has a great impact on the efficiency and accuracy of simulation calculation. Reasonable structure simplification can be carried out to reduce the workload of modeling and improve the work efficiency. So in this paper, according to the structural characteristics of the sample car and the actual calculation needs, the following simplification work is carried out in the finite element modeling.

1. Remove the hole, shoulder, concave and flanging on the surface of the component, and round and smooth it.
2. Omit some parts with small mass and less energy absorption, such as armrest, brake pedal bracket, instrument panel support, etc.
3. For those parts that conflict with the installation and use requirements of other components, the section shape of the components should be reasonably simplified.
4. Some parts (such as batteries) which have little influence on the simulation results are omitted, but the mass of these parts is large, which directly affects the position of vehicle centroid, so they can be reasonably distributed in the corresponding positions in the form of mass points.

2.1.2. Finite element mesh generation

The core idea of finite element analysis is block approximation. The mesh density or the degree of structural discretization have great influence on the calculation error. The increase of grid density will lead to the increase of calculation scale, which will increase the simulation cost and reduce the computational efficiency. When the element length is reduced by K times uniformly, the element density and calculation time will increase by K2 times and

K3 times respectively, which means that the calculation time increases nonlinearly with the element density of 1.5 power. Therefore, the selection of grid density has an important impact on the computational efficiency and accuracy. For the parts with large deformation, the grid density should be increased to better simulate the structural deformation, while for the parts with small deformation, the grid density should be reduced, the calculation time should be shortened and the calculation efficiency should be improved.

On the basis of balancing the calculation accuracy and efficiency and setting the grid density reasonably, we should ensure the grid quality. The better the mesh quality, the faster the calculation speed and the higher the calculation accuracy. Therefore, the following problems should be paid attention to in the process of meshing.

1. The degenerate shell element and body element should be avoided as far as possible. Compared with other common element types, they are too rigid and have poor calculation accuracy.
2. Due to the shell element type, attention should be paid to the minimum element size when meshing, that is, the minimum side length of shell element should be more than five times of its element thickness.
3. The side length of the element directly affects the calculation speed and accuracy. On the basis of ensuring the calculation accuracy, the calculation efficiency should be considered. In the case of certain material properties, the side length and area of the element should be increased as much as possible to avoid the occurrence of too large internal angle of the element.
4. The selection of element size follows the basic principle of gradual change in order to avoid the phenomenon that the size difference between adjacent elements is too large, and prevent the element from distortion, which will affect the calculation accuracy and simulation results.
5. On the premise of the same element type, the stiffness of small-size element is smaller than that of large-size element, and it is more prone to deformation. According to the deformation of car body in real vehicle crash test, the corresponding size element should be selected for different parts of car body to ensure the accuracy of simulation of actual car body deformation.
6. Check the quality of the unit, delete the repeated and cross elements, modify the warpage and deformation elements.

2.2. Establishment and verification of finite element model of moving deformable barrier

Vehicle frontal collision is a collision between the test vehicle and the rigid wall, while the side impact is to impact the stationary vehicle with the variable moving barrier. Therefore, the movable deformable barrier model must be established in the study of side impact simulation, which is the biggest difference between the simulation research of vehicle side impact and frontal collision. In the current side impact regulations, whether it is fmvss214 or ECER95 in Europe and America, or occupant protection in vehicle side impact (GB20071-2006) in China, the movable deformable barrier is the basic tool for side impact test research.

2.2.1. Establishment of finite element model of moving deformable barrier

According to GB20071-2006, the process of finite element modeling of MDB is: Firstly, the CAD model is transformed into the preprocessing software hyper through IGES graphics data exchange format. In mesh, mesh is used to mesh and define the materials and properties of each component, and the connection relationship between the components. Finally, the load, constraint or boundary conditions are applied to the model. According to the side impact standard of our country, the moving deformable barrier needs to impact the test vehicle at the speed of 50 km/h. The movable deformable barrier model is mainly composed of the mobile vehicle and the deformable impactor, and its overall dimensions are shown in Figure 1.

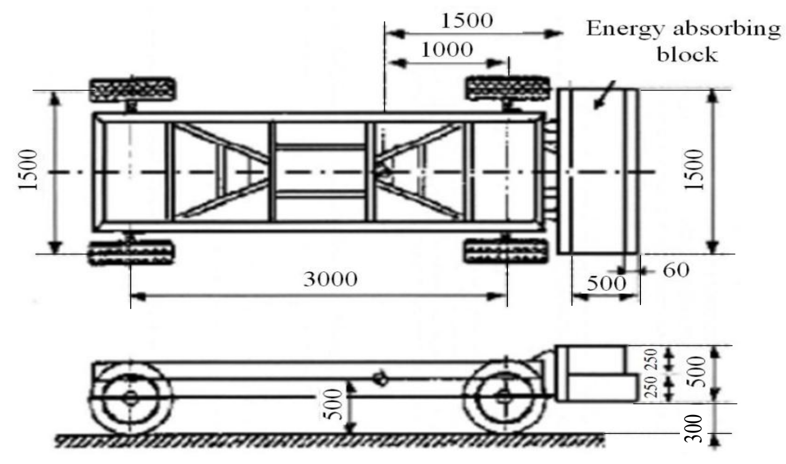

Figure 1. Overall structural dimensions of moving deformable barrier (unit: mm)

As shown in Figure 1, the standard requirements for the finite element model of movable variable barrier are shown in Figure 1, and the structural dimensions of the front end of the model are shown in Figure 2. The total mass of MDB was 950±20 kg, and the center of mass was within ±10 mm of the longitudinal symmetry plane. The finite element model of the movable deformable barrier is mainly composed of the moving frame model, the front energy absorbing block model, the middle glue and the front and rear connecting panels. The moving frame is simulated by shell element, because it will not deform in the whole collision process, so the rigid material (MAT) is adopted _ Rigid); The energy absorbing block in front of the frame is simulated by solid element, and the most commonly used honeycomb aluminum material (MAT) is used _ The glues in the middle are simulated by beam element, and the nonlinear plastic spring discrete material (MAT) is used _ NONLINEAR _ PLASTIC _ DISCRETE _ BEAM).

The trolley model is based on the frame, and some parts (such as wheels, rear connecting plate, key mass, etc.) pass through the CONSTRAINED _ RIGID _ BODIES. The definition of bodies establishes its connection with the frame. The energy absorbing block in the last part is connected with the rear connecting plate through tie contact. Based on the above modeling ideas, the final finite element model is shown in Figure 3 and the finite element model parameters are shown in Table 1.

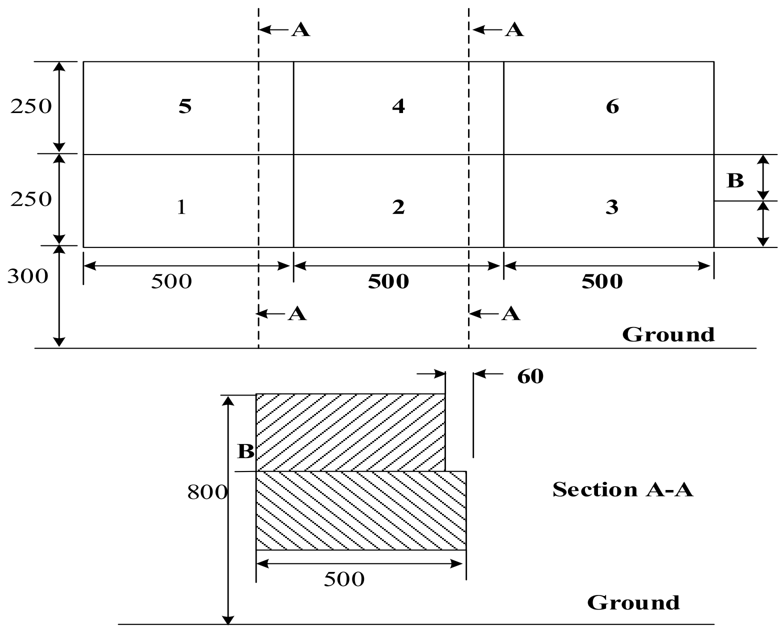

Figure 2. Structure size of energy absorbing block for moving deformable barrier (unit: mm)

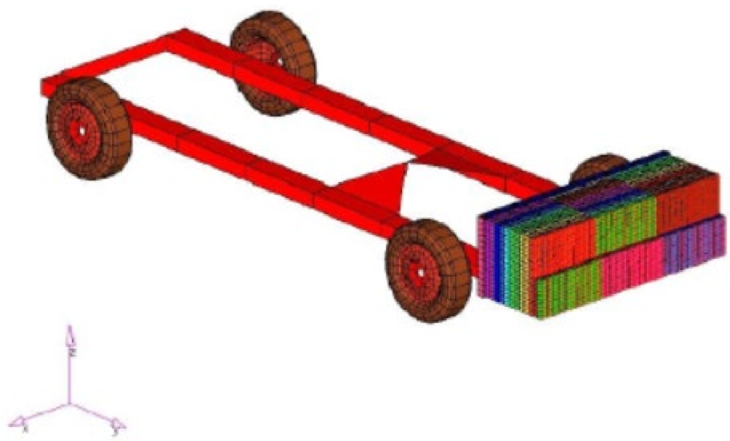

Figure 3. Finite element model of moving deformable barrier

Table 1. Model parameters

Mass of MDB (kg)	952.16
Location of MDB centroid (mm)	X = 0.45, y = -1859, 77 z = 511.43
Geometric parameters of mobile vehicle (mm)	Wheelbase 3000, wheelbase 1500, ground clearance 300
Initial speed of MDB (m/s)	9.722
Total number of units	239765
Number of deformable elements	18039
Number of rigid body elements	181375
Time step (ms)	0.76

2.2.2. Verification of finite element model of moving deformable barrier

Because the stiffness of the front part of the impact vehicle is mainly simulated by the energy absorbing block, the performance of the energy absorbing block needs to be tested dynamically. The verification of moving deformable barrier in ECE R95 regulations and China's side impact regulations is to examine the overall force displacement characteristics of the energy absorption block structure, the force displacement characteristics of each energy absorption block, and the energy absorption characteristics of each energy absorption block. The force displacement characteristics are required to be within the band range specified by the corresponding upper and lower limits, and the energy absorbed by each energy absorbing block meets a certain range [4-6].

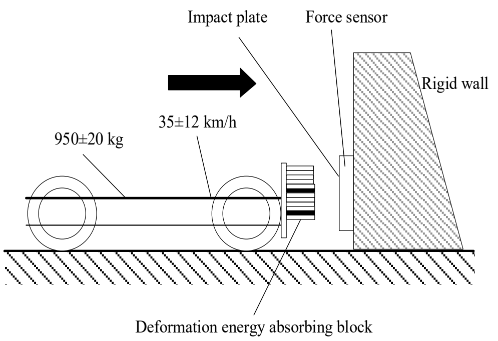

Figure 4. Schematic diagram of mobile barrier verification test

As shown in Figure 4, the schematic diagram of mobile obstacle avoidance verification test is shown. When the rigid wall is vertically impacted at the speed of 35 km/h, the force displacement curve and energy absorption situation of the energy absorption block installed at the front end of the moving barrier are investigated. Among them, the rigid wall is a rigid reinforced concrete structure, the width of one side contacting with the MDB is not less than 3 m, the height is not less than 1.5 m, and the total weight is not less than 70 t. The collision contact plane should be perpendicular to the runway axis, and the surface should be covered with load sensors to measure the total load of each part of the moving deformable barrier at the moment of collision. In the last part of the runway, in front of the rigid barrier, there should be at least 5 m horizontal and smooth pavement [7-9].

There is a force measuring wall in front of the rigid wall to measure the impact force during impact. Through the simulation test of the barrier module, the contact between the rigid wall, the ground and the barrier are defined in the inertial space [10-13]. The displacement curve and velocity curve of the barrier in the process of collision are output from the position of the mass center of the barrier, so as to obtain the motion characteristics of the moving barrier in the process of full-frontal impact [14, 15]. The force time history curves of six impact energy absorbing blocks are output from the model, which is converted into the curve of force variation with displacement, so as to test the stiffness characteristics of the front deformation energy absorption block.

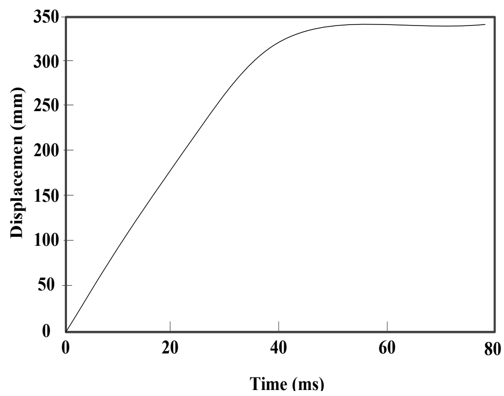

Figure 5. Displacement curve of MDB centroid

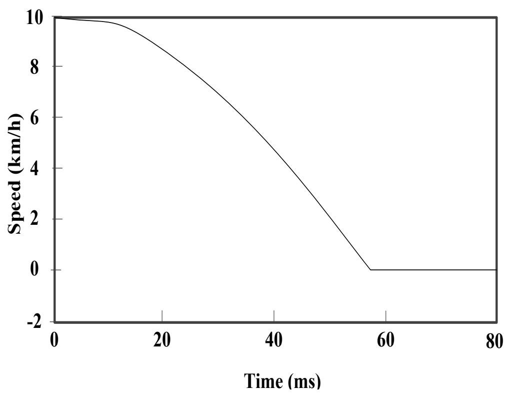

Figure 6. Speed time curve of MDB

As shown in the figure, Figure 5 shows the displacement diagram of the moving deformable barrier mass center with time, and Figure 6 shows the change diagram of the moving deformation barrier speed with time. From the displacement curve of the mass center of the moving deformation barrier (Figure 5), it can be seen that the maximum deformation of the moving barrier energy absorbing block is 341 mm, which meets the requirements of GB20071-2006 regulations of 330±20 mm. Figure 6 shows that the time for the moving barrier to decelerate from the initial speed of 35 km/h to 0 is 56.5 ms, and then rebound, and the speed reaches -1.2 km/h.

See Figures 7-10, in which the dotted line is the upper and lower limit values of the force displacement curve, and the part surrounded by it is the stiffness characteristic range of the impactor meeting the requirements of regulations. It can be seen that the stiffness characteristics of the six deformation blocks are within the limits specified by the regulations, among which the stiffness characteristics of the first and third blocks are the same, the fifth and sixth blocks are the same, and the barrier model is symmetrical left and right. In addition, Table 2 shows that the energy absorbed by each energy absorption block meets the requirements of regulations. Through the verification test, it can be concluded that the moving deformable barrier finite element model meets the test requirements and can be used as the impactor of the side impact simulation model.

2.3. Establishment and verification of vehicle finite element model

2.3.1. Establishment of vehicle finite element model

In this paper, a mass production car abroad is taken as the research object. The whole vehicle model is built on the basis of CAD three-dimensional model, and the finite element pre-processing software hyper is used Mesh is divided in mesh [16-19]. In order to control the size of the model under the premise of ensuring the accuracy of the model and saving the calculation time, the smaller mesh is used in the main deformation parts of side impact, and the larger mesh is used in the secondary parts. The BIW, body panel and seat frame of the finite element model of the whole vehicle are simulated by shell element mesh, and the piecewise linear plastic material is used; The interior panel material is also defined as piecewise linear plastic material when the front door of the driver's side is added with the interior trim panel; The suspension system, door hinge and tire are defined according to their motion relations, mainly including: Ball Joint _ Spherical, joint _ Revolve), column hinge (joint) _ Cylindrical) and spring damping unit; The connection relationship between components is also referred to the real body manufacturing process, and the welding position is controlled _ Spotweld spot welding unit, bolt connection means constrained _ NODAL _ RIGID _ Body rigid connection; Powertrain part is simplified as rigid body, and other parts of vehicle harness, pipeline and sealing strip which have little impact on side impact are replaced or omitted by mass unit.

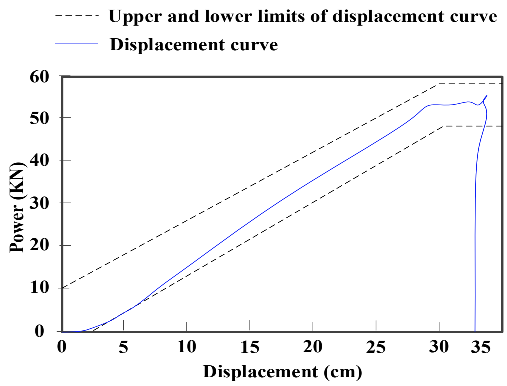

Figure 7. Force displacement curves of the first and third blocks

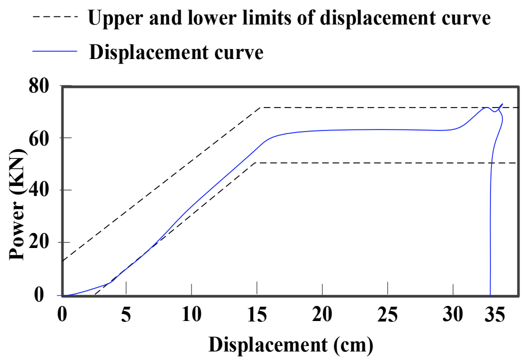

Figure 8. Force displacement curve of the second block

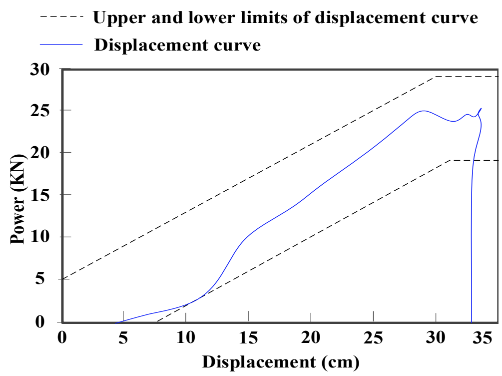

Figure 9. Force displacement curve of the fourth block

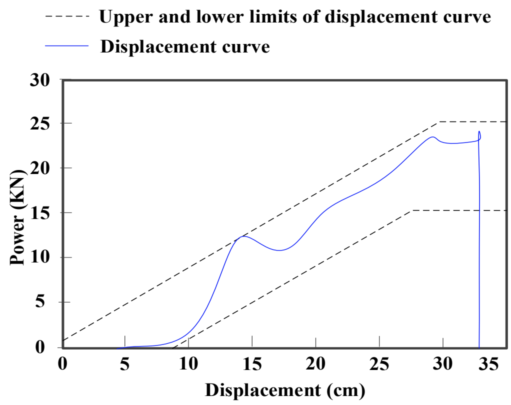

Figure 10. Force displacement curves of the 5th and 6th blocks

2.3.2. Validation of finite element model for vehicle side impact

Qualitative and quantitative evaluation methods are usually used to evaluate the authenticity and accuracy of simulation results [20-22]. The qualitative evaluation method is mainly to check and compare the deformation shape of the impact area in the test and simulation, the impact characteristics of the main components, and the movement of the vehicle and moving deformable barrier after the collision. Quantitative evaluation focuses on the comparison of deformation, acceleration and impact load of different parts of vehicle and moving deformable barrier.

In this paper, according to the requirements of GB20071-2006, the side impact simulation model is established, as shown in Figure 12. The moving deformable barrier (MDB) vertically impacts the side of the stationary vehicle at the speed of 50 km/h. Its vertical line passes through the R point of the front seat on the impact side of the vehicle. The simulation time is set at 140 Ms. According to the energy change of the side impact simulation process, confirm whether the parameters of the side impact simulation model are reasonable. The overall energy change of the system is shown in Figure 13.

It can be seen from Figure 13 that the energy composition of the model system is relatively reasonable, the total energy is conserved, the slip interface energy and hourglass energy keep very small positive values, and do not exceed 5% of the total energy. Therefore, it can be explained that in the modeling process, the standard of finite element mesh, the connection of models and the setting of various solution cards are reasonable. Figure 14 shows the simulation and experimental comparison of the acceleration time curve at the lower end of the B-pillar on the impacted side. It can be seen from the figure that the change trend of the acceleration curve is basically consistent, and the peak value is in good agreement with the occurrence time. The error may be caused by the omission of some body accessories, material parameters and solder joint model, which are different from the actual situation, but the overall error, especially at the first and second peak, is less than 5%. It can be seen that the side wall stiffness of the finite element model is basically consistent with that of the real vehicle, and this model can be used in the following side impact simulation analysis instead of the real vehicle.

Table 2. Energy absorbed by each energy absorption block (kJ)

Project	The first piece	The second piece	The third piece	The fourth piece	The fifth piece	The sixth piece	Population
Simulation	9.55	14.26	9.55	4.27	3.93	3.93	45.49
Regulatory requirements	9.5±2	15±2	9.5±2	4±1	3.5±1	3.5±1	45±3
Result	Qualified	Qualified	Qualified	Qualified	Qualified	Qualified	Qualified

Figure 11. Finite element model of vehicle

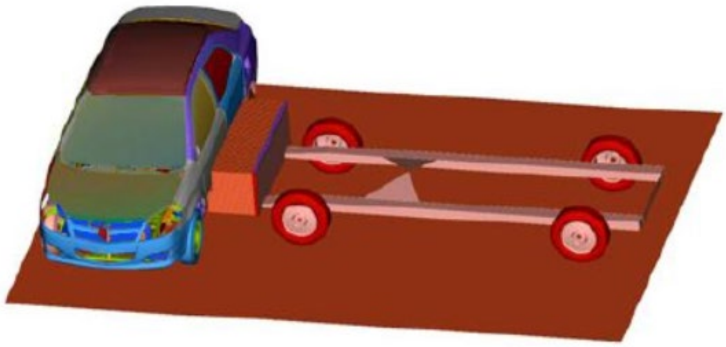

Figure 12. Vehicle side impact model

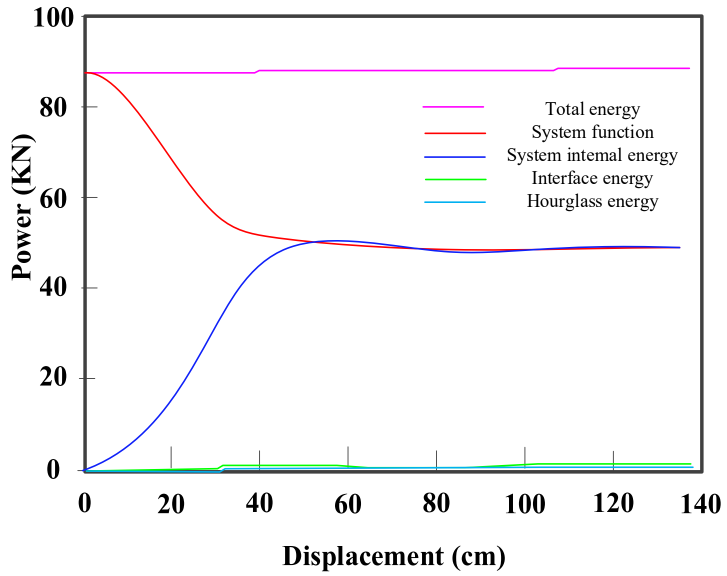

Figure 13. Overall energy curve of the system

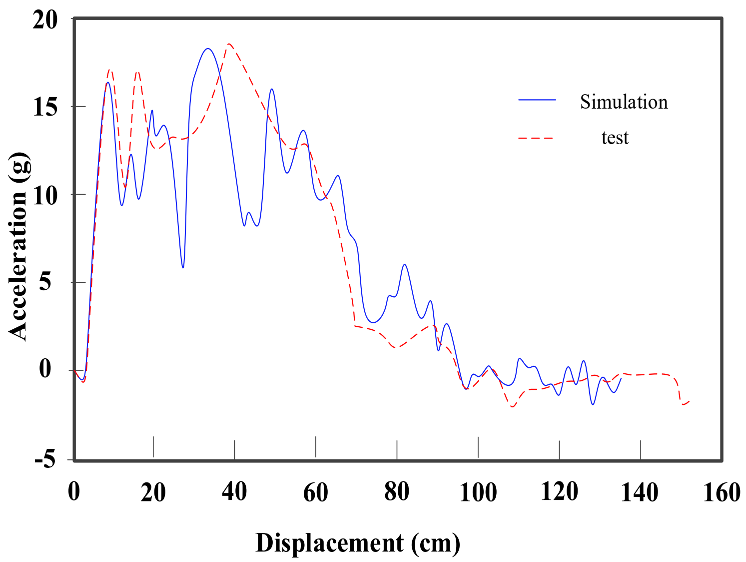

Figure 14. Acceleration at the lower end of B-pillar on the impacted side

3. Safety Optimization Design of Vehicle Side Impact

The optimization design of side impact safety performance can not completely rely on the real vehicle test method, the main means is to use CAE analysis and multidisciplinary optimization theory to optimize the body design. The main content of body multidisciplinary optimization research is to explore effective multidisciplinary collaborative optimization design strategy, realize the concurrent design of multi-disciplinary subsystems of the body, and finally obtain the global optimal solution of the body system.

3.1. Design objectives

Before multidisciplinary optimization design, it is necessary to define the safety design objectives of the vehicle. Through the test method of competitive product analysis, the safety analysis of competitive vehicles with high side impact safety in the market is carried out to obtain the design objectives. Refer to Table 3 for the safety performance of side impact and the design objectives of this model.

The B-pillar inner plate intrusion and B-pillar impact acceleration are selected as the optimization objectives and constraints, while other performance objectives are used as the verification conditions for the feasibility of the optimal design scheme.

Table 3. Side impact performance and design target of reference vehicle

Key position	Reference car	Target value
Front door inner panel (chest) invasion (mm)	147	≤ 140
B-pillar inner plate (chest) invasion (mm)	156	≤ 140
Invasion velocity of lower part of B-pillar (m·s-1)	7.4	≤ 7
B-pillar (head) acceleration (g)	142	≤ 135

3.2. Multidisciplinary collaborative optimization based on adaptive weighting

The essence of body multidisciplinary collaborative optimization problem is to find a set of vectors composed of design variables in the feasible region, so that the surrogate objective function values of various performance of the body system can be optimized as much as possible. In the past, the research on multi-disciplinary problems is often carried out with single objective optimization. However, it is difficult to find the overall optimal solution of the system by using the single objective optimization method because of the conflicts among the objectives. Aiming at the optimal solution problem of multi-objective collaborative optimization, a multidisciplinary collaborative optimization method based on adaptive weighting is proposed. The mathematical model is described as follows:

$$\min ... \mathrm{F}(\mathrm{t}) = \underset{0<}{}$$

$$\begin{aligned} \min \quad & F(t) = \sum_{i=1}^{n} w_i F_i(t) \qquad 0 \le w_i \le 1 \cup \sum_{i=1}^{n} w_i \\ s.t. \quad & g_i(t) < 0 \qquad i = 1, 2, \cdots, p \\ & g_j(t) = 0 \qquad j = 1, 2, \cdots, q \end{aligned} \tag{1}$$

Among them, $F(t)$, $F_i(t)$, $g_i(t)$ and $g_j(t)$ are the weighted total objective function, the objective function of each performance, the inequality constraint function and the equality constraint function respectively; w_i is the weighting coefficient; n, p, q are the number of corresponding functions; t is a vector composed of design variables. In the process of optimization design, the weighting coefficient w_i is selected by adaptive adjustment, and the total objective function $F(t)$ is constructed. After the multi-objective is transformed into a single objective, multidisciplinary collaborative optimization is carried out according to the design requirements of the designer as constraints.

After the optimization, the design objective is taken as the judgment condition to judge whether the $F_i(t)$ value obtained by optimization meets the requirements of the design objective (fihop T). If it is, the optimization result will be output. Otherwise, the weighted coefficient w_i will be adaptively transformed according to the design requirements, and multidisciplinary optimization design will be carried out again. Finally, the optimal scheme of multi-objective optimization satisfying the design objectives is obtained, and the weighting coefficient w_i obtained from this is called adaptive weighting coefficient. The adaptive weighting method can adaptively increase or decrease the weight ratio of the response function according to the optimization results and optimization objectives. Finally, it can automatically obtain the global optimum and significantly reduce the optimization times. In the optimization design of the body structure, the side impact safety of the body and the lightweight performance of the body are optimized together. Under the condition that the body mass is not increased, the B-pillar intrusion and B-pillar acceleration are the minimum, so as to improve the safety of the side impact. According to the adaptive weighting method, the mathematical expression of multidisciplinary optimization is as follows:

$$\begin{aligned} \min \quad & F(t) = w_1 d_b + w_2 a_b \\ s.t. \quad & m < m_0 \\ & d_b \le 120mm \qquad a_b \le 120mm \\ & 0.8mm \le t \le 2.5mm \end{aligned} \tag{2}$$

Among them, w_1 and w_2 are adaptive weighting coefficients; m_0 is the total vehicle mass before optimization (1674.3 kg).

According to the design variables and optimization objectives, the sequential quadratic programming (SQP) method based on Lagrange Hessian matrix is used to solve the multi-objective optimization problem by using MATLAB software. The Hessian matrix is calculated by quasi Newton method. After several iterations, the optimization results are shown in Table 4.

Table 4. Optimization results

Name	w_1	w_2	F	d_b/mm	a_b/g	m/kg	t1	t2	t3	t4	t5	t6
Initial value	0.6	0.4	147.93	146.40	150.22	1674.3	1.6	2．0	2.8	1.8	1.8	1.6000
Optimization value	0.6	0.4	120.57	113.48	131.20	1673.9	2．0	1.5	1.6	2．0	2．0	0.8873

Considering the actual production and manufacturing process, the optimal values of each design variable are determined as follows: t1 = 2.0 mm, t2 = 1.5 mm, T3 = 1.6 mm, t4 = 2.0 mm, T5 = 2.0 mm, t6 = 0.8 mm. Evaluation of optimization results in order to verify whether the optimization results have practical significance, according to the optimization value of design variables, the optimized simulation model is established for simulation test. The simulation results are compared with those before optimization.

It can be seen from Table 5 that the performance of the optimized structure exceeds the design target, and the side impact safety is significantly improved without increasing the vehicle mass.

Table 5. Comparison of performance indexes before and after optimization

Name	Before improvement	After improvement
Front door (chest) intrusion (mm)	146.98	122.46
B-pillar (chest) invasion (mm)	142.00	122.05
B-pillar (head) acceleration (g)	150.20	132.8
B-pillar (bottom) invasion velocity (m·s-1)	10.70	6.3
Vehicle mass (kg)	1674.30	1673.8

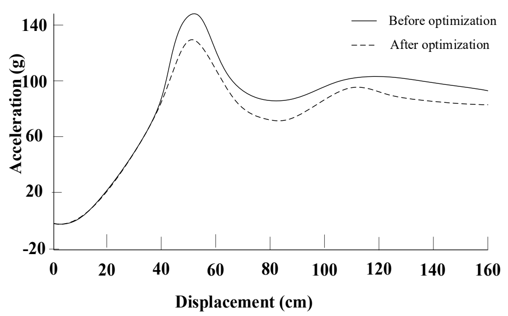

Figure 15. Acceleration comparison of B-pillar (head position) before and after optimization

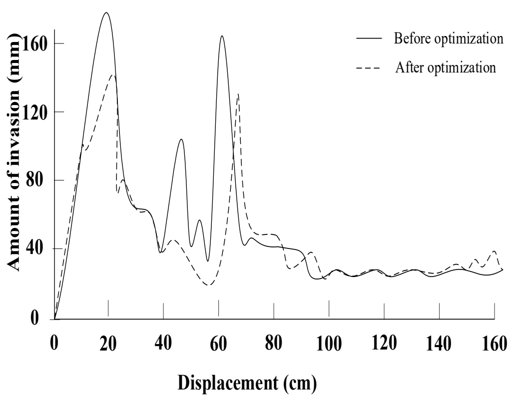

Figure 16. Comparison of B-pillar (chest position) invasion before and after optimization

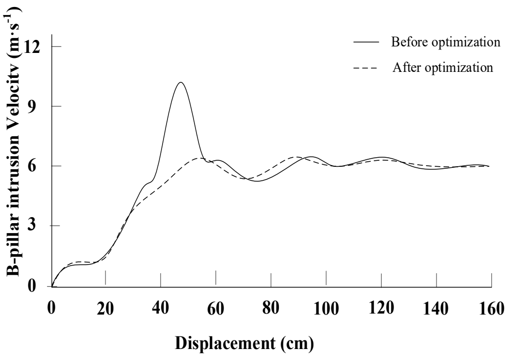

Figure 17. Comparison of invasion velocity of B pillar before and after optimization

At the same time, from the comparison diagram of acceleration, intrusion volume and intrusion speed before and after optimization in Figures 15-17, it can be seen that the acceleration value, intrusion volume and intrusion speed of the optimized structure are reduced by 20 g, 130 mm and 5.5 $m.s^{-1}$ respectively, which is significantly lower than that of the original model. Through the above analysis, it can be seen that the optimized structure is significantly better than the original vehicle in terms of side impact safety, which also shows that the optimization scheme proposed in this paper is feasible. The reason for this phenomenon is that the design method in this paper optimizes the position of the anti-collision beam by increasing the plate thickness, using ultra-high strength steel, increasing structural stiffeners, and using anti-collision beams with different section shapes. The safety of the original vehicle is verified by simulation test.

4. Conclusions

Because the defect of simulation analysis is that it can produce error, even result in error, so the correct result depends on the correct simulation model. The model is reliable after strict verification, and the analysis and optimization design on this model are effective. According to China's side impact regulations and standards, combined with the verification method of simulation models, the moving deformable barrier and vehicle models are verified respectively. The verification results show that the finite element model is reliable and effective. On this basis, the vehicle side impact simulation model is established and verified. The verified vehicle side impact simulation model can be used for further analysis and research. For front impact or rear end collision, the car body has larger buffer space design, but the side only provides a small collision buffer space. Therefore, it is very difficult to enhance the impact resistance of the side of the car and improve the anti-collision technology. In order to improve the side impact resistance of the vehicle body, the key point is to transfer the collision energy to other protective beams, columns, floors, roofs and other parts of the car body. These parts disperse and absorb the impact force, which greatly limits the degree of possible injury, even to the minimum. Through the research on the impact characteristics of vehicle side impact, several improvement schemes are proposed to improve the crashworthiness of vehicle side impact structure. On this basis, the effectiveness of the scheme is analyzed and evaluated. The results show that the acceleration, intrusion volume and intrusion velocity of the optimized structure are reduced by 20 g, 130 mm and 5.5 m.s^{-1} respectively, which is significantly lower than that of the original model. Through the evaluation of the optimization results, the feasibility of the optimization scheme is verified, and the side impact safety is significantly improved.

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