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Body optimization approach of sedan structure for improving small overlap impact rating

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Abstract：Small overlap accidents of sedans are frequent and often result in severe occupant injuries. Small overlap scenarios exert loads which by-pass the current vehicle loadpath architecture and generate extreme body in white (BIW) deformations which can in some cases destroy the door opening panel (DOP) in the A pillar area. These accidents are now of serious concern to the automotive community, as such vehicles are now subjected to an impact evaluation rating protocol initiated by IIHS and this since 2012. The paper proposes for the first time an optimization process using a response surface methodology, to improve the small overlap rating, by considering engine-room energy management, suspension safety design and passenger compartment enhancement, with the objective of minimizing BIW intrusions. The research has initially created a baseline scenario by building a small overlap computer scenario which was correlated against real IIHS small overlap crash test data. Longitudinal and shotguns section sizes to meet critical buckling forces as well as ‘A’ pillar gauges were considered in the study, which lead to the redesign of the engine bay re-design to decrease the impact force transferred to the passenger compartment. The optimal results indicated that the intrusion was decreased by an average of 58.64 %, with a minimum percentage of 44.98 % around footrest area, leading to an IIHS ratiung improvement from poor to good. The proposed crashworthiness design approach is effective in vehicle structure optimization for better small overlap impact performance.

Key words：sedan body optimization; small overlap impact; finite element model; crashworthiness analysis

1. Introduction

Small overlap impact is frequent and often results in remarkably injury risk to the occupant [1,2]. A vehicle small overlap impact evaluation rating protocol was issued by the IIHS to improve the crashworthiness of vehicles and reduce passenger injuries in 2012 [3]. For an automobile manufacturer, the small overlap impact test is a significant challenge. In this test, the crash forces bypass the vehicle’s longitudinal frame rails and then concentrate the force in the front wheel, suspension, firewall and A-pillar [4,5]. This usually results in severe intrusions in the passenger compartment.

Over the past decades, the study on energy absorption characteristics and body optimization has been one of the hottest technical matters in the field of vehicle safety, and a series of essential research achievements have been made. For instance, Chen and Wierzbicki [6] investigated the energy-absorption characteristics of hollow multi-cell columns. They found that the gain in specific energy absorption of the double cell and the triple cell is about 15% compared to the single cell, and the triple cell is no better than double-cell in terms of specific energy absorption. Kecman [7] studied the bending collapse behaviors of rectangular and square section tubes and derived a set of formulae relating the hinge moment and associated angle of rotation. Mamalis et al. [8] theoretically analyzed the inextensional and extensible collapse mechanisms. They found that polygonal tubes could achieve a better energy-absorbing efficiency. Wu et al. [9] compared the crashworthiness characteristics of multi-cell thin-wall structures. In their study, single-cell square tube, four-cell square tube, and five-cell tube simulation models were set up. Their simulation results showed that the energy-absorption of multi-cell tubes could increase with the number of cells. Moreover, Zhang and Lan [10] studied the energy absorption characteristic of unicellular, square-hole multi-cell and honeycomb multi-cell tubes. Their results showed that the square-hole multi-cell absorbed maximum energy and had a stable deformation mode. Furthermore, Fang et al. [11] investigated the effect of cell number and oblique loads crashing behaviors. They found that the increase in cell number can be beneficial to the energy absorption but detrimental due to the increase in peak force. However, the plastic hinge generated appeared easily in hollow thin-walled structures under impact. Yu et al. [12] studied bending performance of thin-walled beam enforcement structure. In their study, the thin-walled square tube, the cover-plated reinforcement thin-walled square tube, aluminum-foam filled square tube and equal strength cover plated reinforcement square tube models were carried out, and they found that the deformation form of the equal strength cover plated reinforcement square tube is more stable than other enforcement structure, and the energy absorption of the equal-strength square tube column is the largest in the oblique impact.

The previous work mentioned above focuses on the thin-walled structure performance under axial loading or lateral impact. However, in small overlap impact, the vehicle structure suffers more complicated loading. Besides, previous research found that vehicle crashworthiness optimization in small overlap impact was beneficial for other frontal crash configurations[1]. Thus, the energy-absorption performance of the vehicle enforcement structures to small overlap impact loading attracted researchers' attention. Some previous physical tests and computer simulation studies indicate that engine room structure, vehicle front wheel and passenger compartment place a significant role in improving small overlap impact rating [13-18]. Zhang et al. [19] investigated the energy-absorption performance of a B class vehicle in small overlap impact. They found that the vehicle could reach good rating when the left front rail absorbs 31.47% collision energy, the left shotgun absorbs 9.79% collision energy, and the subframe absorbs 17.48% collision energy. Munjurulimana et al. [20] studied the effect of adding energy-absorbing members in the sidewalls of longitudinal on improving small overlap impact rating. They found that the energy-absorbing structures which are made of metal plastic hybrids, metal and plastic can achieve a maximum 150mm decrease in the forward movement of the base of the A-pillar. Mueller B C et al. [21] investigated the effect of reinforcement of the passenger compartment, the use of energy-absorbing fender structures, and the addition of engagement structures. In their study, the vehicle with the most reduced passenger compartment intrusion is designed by extending shotgun and passenger compartment enhancement. Nguyen et al. [22-24] optimized the vehicle by developing two reinforced components such as longitudinal reinforcement and rocker panel reinforcement. In their study, energy absorption of the optimal vehicle during the impact increased by 163%, the intrusion of the passenger compartment was significantly reduced, and the overall rating of frontal structures was upgraded from 'poor' to 'good'. Elliot et al. [25] investigated a passenger compartment structure that combines energy absorption and high rigidity structure. They proposed a new front door hinge pillar dual box structural to reduce the deformation of the hinge pillar and decreased. In their study, the amount of intrusion can be reduced by up to 30%. Kim [26] proposed a body lift ring structure that was defined by the front side member, dash panel, and A-pillar. Through analyzing, they found that the new structure could convert collision energy into the deformation of the front side member and the energy of lifting A-pillar which could benefit the crashworthiness. Brar [27] established an optimization to improve the crashworthiness of the vehicle body by optimizing the passenger compartment. In his study, he proposed an internal structure of the door and found that the passenger compartment enhancements structure could improve crashworthiness. Chen et al. [28] enhanced the vehicle structure by filling structural foam in the A-pillars and the side panels, adding a roof crossbeam, and reinforcing the rear wall of the passenger compartment. Their results indicated that energy absorption was more homogenous.

Deterministic optimization has been made, but most studies on crashworthiness design focus on deterministic optimization, in which the design variables and parameters involved are assumed to be confident, resulting in less meaningful in the optimization results [29]. To overcome this drawback, researchers introduce the response surface methodology to propose the optimal values of the vehicle structure variables to improve vehicle crashworthiness [21-24,30-31]. Kurtaran et al. performed crashworthiness design optimization using successive response surface methodology [32]. Hou et al. used response surface methodology to minimize the crash peak force by seeking for optimal design of multi-cell cross-sectional thin-walled columns [33]. Zhang et al. used the response surface methodology with quadratic functions to optimize the vehicle side interior panels [34]. Toksoy and Güden used the response surface methodology in the optimization of the energy absorption of Al crash boxes [35]. Lu et al. presented a methodology for response surface methodology which was applied to crashworthiness optimization of frontal impact, considering structural crashworthiness [36]. The above researches show that the response surface methodology is playing an important role in vehicle optimization design procedures.

According to the conclusions from the previous study above, it is true that the multi-cell structure has better crashworthiness than the single-cell structure subjected to axial loading and lateral loading. Thus, a multi-cell structure likely could play an essential role in improving vehicle crashworthiness. Meanwhile, it is true that the vehicle wheel plays a vital role in the transfer impact force to the passenger compartment. Thus, suspension safety design could contribute to improving vehicle crashworthiness. Furthermore, it is also true that the vehicle crashworthiness could be optimal by adding energy-absorption members in the vehicle engine room, making suspension safety design and enhancing passenger compartment. However, to our best knowledge, the method of structure optimization by controlling energy-absorption of engine room structures, making suspension safety design and enhancing passenger compartment has not been widely investigated and reported.

In this paper, engine-room energy management, suspension safety design, and passenger compartment enhancement are established as an optimization approach to improve small overlap impact rating. Following the introduction, a small overlap impact simulation model is developed and validated in section 2. Then, the simulation results are described in section 3. The results are used to rate the small overlap impact rating of the sedan. In order to improve small overlap impact rating, a body optimization approach including engine-room energy management, suspension safety design and passenger compartment enhancement is established in section 4. The results of the optimization are presented quantitatively in section 5.

2 Small overlap modeling

2.1. Small overlap impact test evaluation rating protocol

The vehicle collides with a rigid barrier at 64km/h with 25% overlap based on IIHS small overlap impact evaluation rating protocol. Besides, measures of passenger compartment intrusion are used to evaluate the structure crashworthiness performance. According to [37], sixteen points are used for measuring vehicle intrusion. The purpose of this study is to develop a crashworthiness design method, therefore only lower hinge pillar, footrest, left toepan, brake pedal, parking brake pedal and rocker panel were used to measure intrusion. Figure. 1 shows some rating guidelines for assessing the car safety rating in small overlap impact [37].

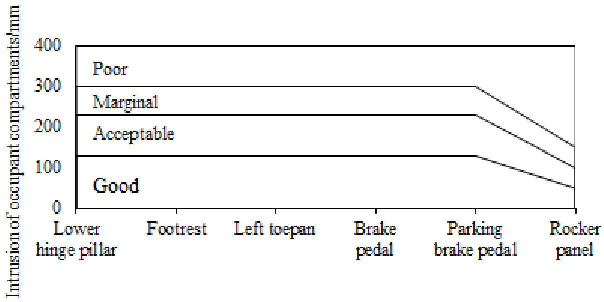


Figure.1 Guidelines for rating of passenger compartment

The IIHS divides all measurement points into two measurement areas, the upper compartments and the lower compartments [37]. The upper compartments include the steering column, upper hinge pillar max, upper dash, lower instrument panel; the lower compartments include the lower hinge pillar max, the footrest, the left toepan, the brake pedal, the parking brake, the rocker panel lateral average. The intrusion amounts of the upper and lower parts of the passenger compartment are evaluated separately, and the worse rating of them is taken as the final rating of the crashworthiness of the structure.

2.2 Description and validation of small overlap impact FE Model

A vehicle finite element model which had been verified in several ways was used in this study for simulation based on the IIHS small overlap research program [38-40]. In this study, the rigid barrier was a flat barrier with a 150 mm radius and 1533 mm high[41]. The rigid barrier was arranged on the driver's side, and the vehicle width could be divided to set the position of the rigid barrier at 25%. This paper also validated the new small overlap impact model by comparing the small overlap impact test and simulation results as shown in Figure 2 [42]. Besides, energy absorption of the sedan in small overlap impact simulation has been validated as shown in Figure 3 [17].

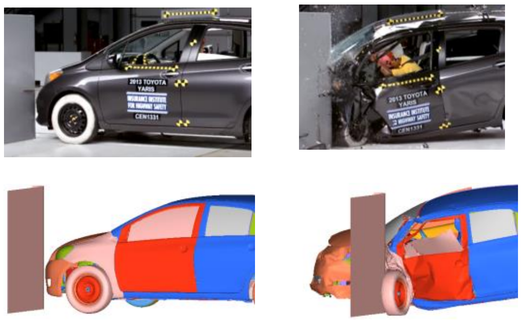


Figure.2 IIHS test and simulation results

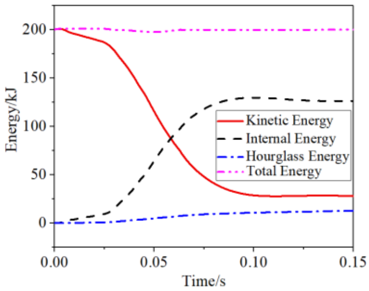


Figure.3 Vehicle energy analysis of the FE model

3 Original model simulation results

3.1 Kinematic analysis of the original model

Figure.4 shows the results of the sedan in the small overlap impact test simulation [17]. Base on the original simulation results, the body was damaged severely during the impact. The wheel moved rearward and squeezed the hinge pillar. Then the A-pillar appeared to bend. The body began to rotate around the rigid barrier, and the A-pillar rebounded slightly with the rotation of the body after the deformation of the passenger compartment reaching its maximum.

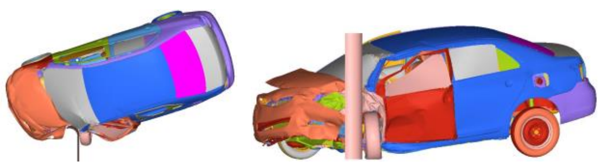


Figure.4 Top and right views for the intrusion of the original model.

3.2 Intrusion analysis of the original model

The severe deformation of the passenger compartment was caused by the fact that the majority of the loading was outside longitudinal structures. The longitudinal failed to effectively reduce the impact of the rigid barrier, resulting in significant collision energy being transmitted to the passenger compartment and severe intrusion to the passenger compartment. According to the IIHS rating rule, the initial structural rating is based on a comparison of the measured intrusion with the rating guidelines, as shown in Figure.5[17,37].

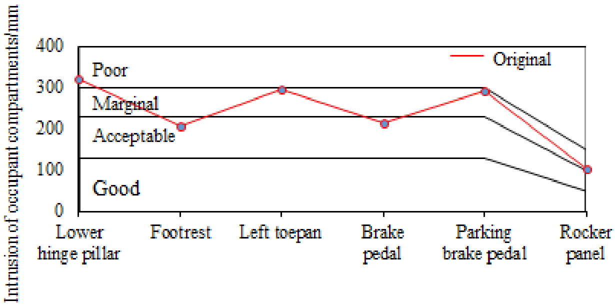


Figure.5 Passenger compartment rating of original model

The result in Figure.5 demonstrates that this model has severe problems among the small overlap impact. The lower hinge pillar points fell in the "poor" zone, and the footrest and the brake pedal intrusion measurement point fell in the "acceptable" zone, the left toepan and parking brake pedal measurement point fell in the "marginal" zone.

4 Optimal vehicle structure design model

In the full and offset frontal impacts both the shotgun and longitudinal are the important crash energy absorption members. However, the crash forces bypass the vehicle's longitudinal frame rails and there are not enough components or space to absorb the impact energy in a small overlap impact [4-5]. Consequently, substantial intrusion concentrates in the impact zone. Previous study found that when the longitudinal reinforcement absorbed 18% energy, the shotgun absorbed 11% energy, suspension safety design and passenger compartment enhancement were met, the structural crashworthiness rating of the vehicle could reach a good level. In this paper, a method, including engine-room energy management, suspension safety design and passenger compartment enhancement, was proposed to optimize vehicle body.

4.1 The engine room energy management

The engine room structure is the main energy absorption component in the case of small overlap impact. Moreover, it directly affects the intrusion of the passenger compartment. In order to achieve the energy absorption target, the energy absorbed by the body is reasonably distributed to the engine-room structure based on the principle of energy management in the collision process. Zhang et al. [19] found that the speed of more than 80% of the vehicles under the small overlap impact condition is 20-30km/h at the end of the collision. Therefore, the terminal velocity of the vehicle can be set as 25km/h in the preliminary design. Thus, the total energy absorbed by the sedan in the impact can be calculated according to energy conservation law. Thus, the energy absorbed by the longitudinal reinforcement was designed as 23kJ, and the energy absorbed by the shotgun was designed as 14kJ.

4.1.1 Longitudinal reinforcement designs

As shown in Figure 6, the length L of the longitudinal reinforcement is designed to be 260 mm based on the longitudinal length of the sedan. According to the average axial structural force calculation formula [43], the compression coefficient of the longitudinal reinforcement is initially set to 0.7. Therefore, the target average axial structural force of the longitudinal reinforcement is 138.889kN. The average axial structural force calculation formula is shown as follow.

$$\frac{F}{M_0} = 52.42\left(\frac{b}{t}\right)^{\frac{1}{3}} \tag{1}$$

where, F is the average axial structural force, b is average of the length and width of the rectangular section, t is thin wall structure thickness, M_0 is plastic limit bending moment per unit length.

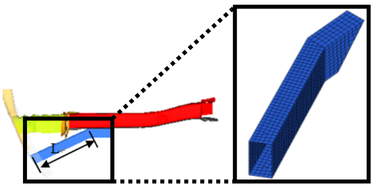


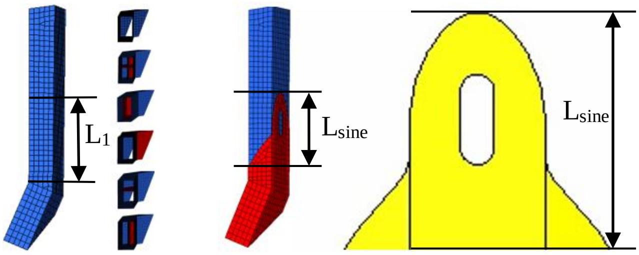
Figure. 6 Longitudinal reinforcement structure

Liu proposed a formula to calculate the average force [43]. According to engineering experience, the formula is improved to obtain the average axial force calculation formula as follows to reduce the calculation error:

$$F = 11.5324\sigma_0 b^{\frac{1}{3}} t^{\frac{5}{3}} \left[1 + \left(\frac{0.33 v_0}{bc}\right)^{\frac{1}{p}}\right] \tag{2}$$

where, σ_0 is average flow stress, v_0 is the initial velocity of the collision, c is characteristic strain rate, p is material sensitivity.

Based on the formulas above, the parameters of the longitudinal reinforcement are as follows: the b is 54.87 mm, and the thickness t is 1.6 mm. In engineering practice, the design section is 40mm long and 70mm wide, and the material is BR1500HS. The simulation result indicates that single-cell structure could undergo global bending which is an inefficient deformation mode. According to previous researches, six simulation models were carried out to get a more stable deformation mode, as shown in Figure 7[6,9-10,12,18,45-46]. The six simulation models include: original single-cell tube , four-cell tube, cover-plated reinforcement tube, equal strength cover plated reinforcement tube, two-cell in vertical tube and two-cell in horizontal tube, and the length of the six reinforcement structure in the longitudinal structure is L_1. The reinforcement structure was optimized by the analytic energy-absorption of simulation model, the optimal structure is two-cell in vertical tube.



(a) Top and front views for the reinforcement structure

(b) Shape of cover-plated of the reinforcement tube.

Figure.7 Sample of the reinforcement structure of the longitudinal reinforcement model.

4.1.2 shotgun structure redesign

In order to achieve the target of 14kJ energy absorption, the shotgun needs to have a better deformation trend in the collision. However, under the load transmitted by the shock housing top and the suspension system during the collision, the root of the shotgun is prone to bend with less energy absorption. Therefore, it is considered to extend the shotgun forward so that the extended structure can fully deform to increase energy absorption before the root of the shotgun is bent during the crash. At the same time, the induced deformation structure is arranged at the root of the shotgun. Consequently, the root could have better energy absorption.

Considering the average axial force and the engine-room structure arrangement, the length of the shotgun is 300mm, and the compression coefficient is 0.7. Therefore, the average axial structural force of the upper finger beam is 71.43kN base on the average axial structural force calculation formula. According to the average axial force calculation formula, the values of the section design parameters of the shotgun extensions are as follows: the material is SAPH440, the b is 54.87 mm, and the t is 1.6mm. In combination with the specific actual situation of the target model, the cross-sectional dimension of the front end of the shotgun is designed to be 40mm long and 70mm wide. Considering the requirements of structural energy absorption and weight reduction, this paper also sets three induction grooves at the root of the shotgun. Furthermore, a reinforcement rod is planed between the shock housings. However, redesign work is advised for the frontal light, since it interferes with the new shotgun design. The effect of the shotgun optimization is shown in Figure 8.

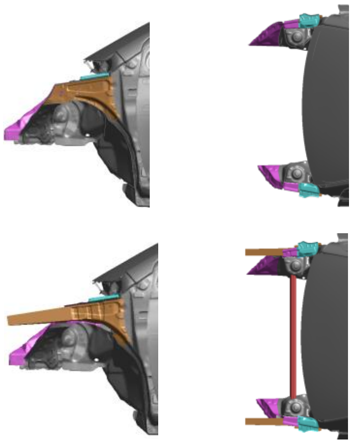


Figure. 8 Comparison of original shotgun and optimal shotgun

4.2 Safety design of the suspension

Previous researches [15,16] indicated that wheels play an essential role in the transmission of loads in small overlap impact. The wheels separated from the vehicle during the small overlap impact benefits the small overlap impact rating. As shown in Figure.5 [42], the front wheels of the 2017 KIA FORTE, the 2017 Ford Fusion, and the 2017 Volvo S90 were separated from the vehicle during the small overlap impact tests while the front wheel of 2016Acura ILX was not. When the wheel is disconnected in the collision, the shotgun can be used to absorb the collision energy, to avoid excessive collision energy being transmitted directly to the passenger compartment. Thus, it is efficient to improve vehicle crashworthiness after introducing the safety design of the suspension.

(a) 2016 Acura ILX small overlap impact test

(b) 2017 KIA Forte small overlap impact test

(c) 2017 Ford Fusion small overlap impact test

(d) 2017 Volvo S90 small overlap impact test

Figure.9 Vehicles in small overlap impact tests[42].

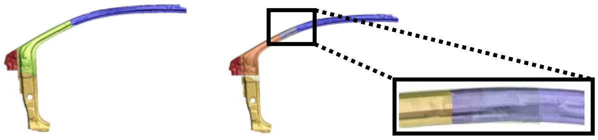
The conventional method of making suspension safety design is to choose the material and thickness for steering ball to make it fails when it subjected to the designed lateral force. Because the finite element simulation solution process has a certain oscillating property, this paper avoids the influence of the oscillation in the finite element simulation process on the model by setting the suspension system forced failure in 65ms.

4.3 Passenger compartments enhance

It is a common optimization approach that strengthening the A-pillar to help to improve the crashworthiness of vehicles. In this paper, the A-pillar was enhanced by optimizing the connection relationship and optimizing thickness. The length of the A-pillar components was extended to optimize the connection relationship of A-pillar. The response surface methodology was conducted to optimal A-pillar thickness.

4.3.1 Connection relationship optimization

The A-pillar was resulting in relatively severe damage because of the weak connection between the A-pillar components in the small overlap impact. It indicates that the connecting relationship between the A-pillar components of the sedan is in urgent need of optimization. Previous researches [17,20] indicated that optimizing the A-pillar has a significant effect on reducing the passenger compartment intrusion. The design is possible to manufacture unless the thickness of the weldment does not exceed 5 mm according to the welding process, and the number of weldments does not exceed 4 layers in order to reduce the manufacturing cost [44]. Furthermore, the shorter the lap joint, the better the weight reduction of the whole vehicle. Thus, the connection relationship optimization is possible to manufacture, and the comparison before and after the optimization of the A-pillar lap joint is shown in Figure.10.

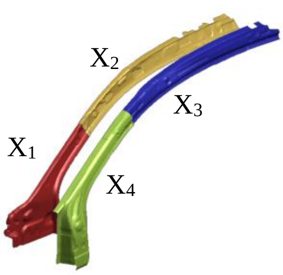


(a) Original A-pillar (b) Optimal A-pillar

Figure.10 Comparison of the original and optimal A-pillar.

4.3.2 Thickness optimization

The A-pillar is subjected to complex impact forces during the collision, the thickness and material of the A-pillar have an influence on the crashworthiness. However, due to the complicated shape of the A-pillar and the high manufacturing cost of the high-strength steel, the thickness is selected as the optimized parameter of the A-pillar. A detailed parametric investigation was carried out to determine the optimal A-pillar thickness parameters. In this paper, the response surface methodology was utilized for optimal A-pillar thickness. The object function was used to solve the following optimization problem: minimize f(x); subject to g(x)<0;Xmin<X<Xmax ; where f(x) is a function of the design objectives to be minimized (e.g., A-pillar deformation), g(x) is constrained (e.g., intrusion), and Xmin and Xmax are the minima and maximum bounds for the vector of design variables x, which has a number of design variables. In order to analyze the main intrusion structure in the small overlap impact, four major variables were designed as shown in Figure 11. Figure 12 shows the measurement points for measuring A-pillar deformation. The minimum and maximum bounds for the vector of the variables are shown in Table 1. The minimum bounds for the variables are the values of the current point decreased by 20%, the maximum bounds for the variables are the values of the current point increased by 20%. The design of experiments and small overlap impact results are shown in Table 2.



A-pillar upper inner (X_1), roof rail internal(X_2), roof rail rear(X_3), A-pillar inner upper (X_4),

Figure.11 Illustration of design variables for vehicle structure.

Figure 12. Locations for measuring A-pillar deformation.

Table 1. Parameters of the new designed frontal cabin structure

Compartments	Low	Base	High
X_1	0.808mm	1.01mm	1.212mm
X_2	0.8552mm	1.069mm	1.2828mm
X_3	1.8016mm	2.252mm	2.7024mm
X_4	1.1296mm	1.412mm	1.6944mm

In order to improve small overlap impact rating, the A-pillar intrusion should be taken into account. The functional connection between deformation and the thicknesses of the roof rail and A-pillar can be calculated by a response surface methodology function. The quadratic polynomial is assumed as formulation (3). As shown in Table 3, the optimal design values were chosen according to the formulation (3).

$$Y(x)=1.261\times10^3-7.163\times10^2X_1-2.521\times10^2X_2-3.512\times10^2X_3-3.446\times10^2X_4+2.603\times10^2X_1X_2+79.710X_1X_3+3.151X_1X_4+44.481X_2X_3-5.509X_2X_4+75.682X_3X_4+76.073X_1^2-39.087X_2^2+28.268X_3^2+38.204X_4^2 \quad (3)$$

Where Y(x) is the optimal target which is the A-pillar intrusion in this paper.

Table 2.Design of experiments and small overlap impact results

X_1	X_2	X_3	X_4	A-pillar displacement/mm
-	-	0	0	77.8624
+	-	0	0	18.8376
-	+	0	0	62.5493
+	+	0	0	48.4908
0	0	0	0	49.463
0	0	-	-	74.4572
0	0	+	-	75.3879
0	0	-	+	24.1572
0	0	+	+	63.5927
-	0	0	-	99.8422
+	0	0	-	53.5636
0	0	0	0	49.463
-	0	0	+	61.8277
+	0	0	+	16.2681
0	-	-	0	61.861

0	+	-	0	54.5608
0	-	+	0	48.217
0	0	0	0	49.463
0	+	+	0	58.0502
-	0	-	0	74.325
+	0	-	0	20.601
0	0	0	0	49.463
-	0	+	0	74.6655
+	0	+	0	49.9499
0	-	0	-	62.729
0	+	0	-	80.1168
0	-	0	+	15.1624
0	+	0	+	31.2197
0	0	0	0	49.463

Table 3. Optimized design variables

Variables	Original	New design
X_1	1.01mm	1.18mm
X_2	1.069mm	0.88mm
X_3	2.252mm	1.95mm
X_4	1.412mm	1.66mm

4.4 Optimal vehicle structure model results

This paper studied the engine-room energy management, suspension safety design and passenger compartment enhancement to improve the vehicle structure in the small overlap impact rating. The response surface methodology was applied for the optimal design solution. The optimal design variables is chosen as shown in Table 3.

4.4.1 Kinematic analysis of the optimal model

According to the optimization results, the wheel began to move back after its impact with the rigid barrier, and the wheel was separated from the body at 65ms. At this time, the crushing deformation of the shotgun has been completed, the rigid barrier hit the hinge column, then the rigid barrier squeezed the passenger compartment, and the deformation of passenger compartment was intensified. The body began to rotate around the rigid barrier when the passenger compartment deformation was close to the maximum. The top and right views for the intrusion of the optimal model during the collision is shown in Figure 13.

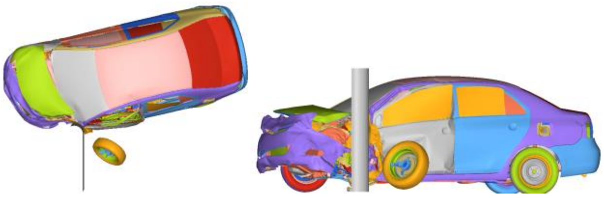


Figure. 13 Top and right views for the intrusion of the optimal model.

4.4.2 Intrusion analysis of the optimal model

According to the IIHS regulations on the crashworthiness of the small overlap impact structure, the intrusion amount of the relevant measurement points is measured as shown in Figure.14[17,37]. The simulation indicates that expected intrusion level is achieved for the optimized passenger compartment. The optimal results show that the intrusion for the lower hinge pillar, footrest, left toe-pan, brake pedal, parking brake and rocker panel was reduced by 60.16%, 44.96%, 74.57%, 47.74%, 63.20% and 61.22% respectively. The overall intrusion was decreased by an average of 58.64 %, and the measurement points reached a "good" level. The crashworthiness performance of the optimized vehicle has been significantly improved.

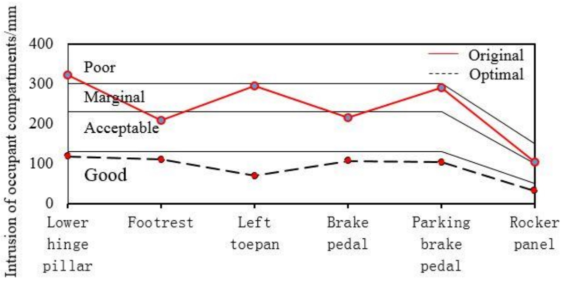


Figure. 14 Rating comparison for passenger compartment.

5 Conclusion

In this study, a small overlap impact simulation model was built up to develop a body optimization approach for better small overlap impact rating. This study indicates it is efficient to improve small overlap impact rating by engine-room energy management, suspension safety design and passenger compartment enhancement. Besides, the response surface methodology is beneficial for optimization of the A-pillar thickness to enhance the passenger compartment's stiffness. The optimization simulation results showed the intrusion of the measurement point had been upgraded to "good", and the overall intrusion was decreased by an average of 58.64 %. Different materials, as well as structural parameters, affect the vehicle crashworthiness performance a lot [47-49]. Thus, the future work will mainly focus on the evaluation of more design parameters (e.g. aspect ratio and material) on crashworthiness performance for better small overlap impact rating, as well as addressing the significant challenges of modeling manufacturing forming effects in the optimization process, as these will change local thinning and strain hardening

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