

A Data-Driven Multi-Objective Optimization Framework for Rear-End Crash Compatibility Design of Truck Rear Underrun Protection Devices

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Abstract—To mitigate underride risks in passenger car–truck rear-end collisions caused by mismatches in vehicle mass, structural stiffness, and height, this study proposes a surrogate model-assisted multi-objective optimization framework for rear underride protection device design. A full-overlap passenger car–truck crash model was first established using explicit dynamic finite element analysis and validated against the static loading and dynamic barrier impact requirements of GB 11567-2017. The numerical reliability of the full-vehicle model was further assessed through energy balance, hourglass energy, and added mass ratio. A crash compatibility evaluation framework was then developed from three aspects: energy distribution, structural intrusion, and vehicle dynamic response. RUPD energy absorption ratio, maximum intrusion of the passenger car front longitudinal beam, and peak longitudinal acceleration at the lower B-pillar were selected as optimization responses, while crossbeam thickness, ground clearance, and longitudinal installation distance were considered as design variables. Optimal Latin hypercube sampling and Kriging surrogate modeling were combined with the NSGA-II algorithm to obtain Pareto-optimal designs. Compared with the baseline design, the optimal solution increased the RUPD energy absorption ratio from 26.3 percent to 33.8 percent and reduced maximum longitudinal beam intrusion from 173 mm to 117 mm, representing a 32.4 percent reduction. The results demonstrate that appropriate coordination of RUPD stiffness and geometric configuration can improve energy distribution and load transfer, thereby reducing underride risk and enhancing crash compatibility.

Keywords—Rear underride protection device; rear-end crash compatibility; explicit dynamic finite element method; Kriging surrogate model; NSGA-II; multi-objective optimization

I. INTRODUCTION

With the continuous expansion of road transportation networks and the increasing number of motor vehicles and freight transportation demands, heavy-duty trucks have become increasingly important in highway freight systems. Compared with passenger cars, heavy-duty trucks generally possess greater vehicle mass, higher structural stiffness, and elevated rear-end load-bearing structures. These differences can result in significant geometric and load-path mismatches during passenger car–truck rear-end collisions, preventing key energy-absorbing components of passenger cars, such as the front bumper, crash box, and longitudinal beams, from establishing stable contact with the rear structure of trucks. Previous accident statistics and studies have demonstrated that passenger car–truck rear-end collisions are associated with a high risk of fatal

injuries, primarily due to substantial disparities in vehicle mass, stiffness, structural height, and load-bearing configuration, which lead to insufficient crash compatibility between the two vehicles [1–3].

The rear underride protection device (RUPD) is an essential passive safety component designed to mitigate passenger car underride and improve structural engagement between colliding vehicles. A properly designed RUPD can reduce the effective load-transfer height of the truck rear structure, facilitate the participation of passenger car frontal energy-absorbing structures in progressive deformation, and improve crash energy distribution to some extent. However, many existing RUPDs are primarily designed to satisfy regulatory requirements regarding geometric dimensions and static load-bearing capacity. Under high-energy impact conditions or unfavorable structural engagement scenarios, limitations may still exist, including insufficient geometric compatibility, limited plastic energy absorption capability, and instability of connecting components [4]. Although regulations such as ECE R58 and FMVSS 223/224 specify requirements for RUPD dimensions, installation positions, and static strength performance, their consideration of full-vehicle dynamic crash compatibility and energy distribution characteristics between vehicles remains relatively limited.

In recent years, extensive experimental investigations and explicit finite element simulations have been conducted to study RUPD structural configurations, thickness parameters, installation dimensions, and underride protection performance [5–11]. These studies have provided valuable guidance for RUPD strength evaluation and structural improvement. However, systematic investigations into the evolution of load-transfer paths, energy distribution mechanisms, and overall crash compatibility during passenger car–truck rear-end collisions remain insufficient. Particularly for multi-objective optimization problems involving strongly coupled performance indicators, simply increasing RUPD stiffness may reduce structural intrusion but simultaneously increase vehicle acceleration responses. Conversely, insufficient stiffness may lead to premature structural instability of the protection device and aggravate underride deformation. Therefore, it is necessary to establish a comprehensive evaluation framework considering energy distribution, structural intrusion, and vehicle dynamic response, followed by multi-objective optimization to achieve improved crash compatibility.

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II. ESTABLISHMENT AND VALIDATION OF A PASSENGER CAR-TRUCK REAR-END COLLISION MODEL WITH RUPD

A. Finite Element Model Development and Regulatory Validation

Rear-end collisions are transient nonlinear dynamic processes involving large deformations, material plasticity, contact nonlinearity, and potential structural instability. Explicit dynamic finite element analysis is widely used to investigate such problems because it can efficiently handle contact-impact events and substantial plastic deformation [12]. Therefore, an explicit dynamic finite element method was adopted in this study to investigate the crash responses of vehicle structures. LS-DYNA has been applied in previous studies to simulate collisions between passenger cars and rear underrun protection devices and to evaluate structural deformation and crashworthiness [13]. In explicit dynamic analysis, the equations of motion are commonly integrated using the central difference method. When a lumped (diagonal) mass matrix is employed, nodal accelerations can be calculated directly from the residual nodal forces without solving a globally coupled system of equations at each time step, thereby improving computational efficiency for large-scale crash simulations [12].

To ensure the numerical stability and accuracy of the simulations, three key aspects were considered. First, the time-step control was implemented, where the critical time step was determined by the element characteristics and structural parameters. Second, contact algorithms were carefully defined. A penalty-based contact algorithm in LS-DYNA was employed to prevent interpenetration between contacting surfaces during the collision process. Third, hourglass control was applied, and the hourglass energy was maintained below 5% of the total system energy, which is generally considered an acceptable criterion for ensuring numerical reliability in crash simulations [14,15].

The rear underride protection device (RUPD) is a critical passive safety component for mitigating passenger car underride and reducing the risk of severe structural intrusion. The accuracy of the RUPD model directly influences the reliability of crash simulation results [16–18]. In this study, the RUPD model was developed and validated according to GB 11567-2017 Requirements for Side and Rear Underride Protection Devices of Vehicles and Trailers. This regulation specifies requirements for the geometric configuration, static load-bearing capacity, and dynamic impact performance of rear underride protection devices. The major evaluation criteria include the maximum longitudinal displacement of the protection device (≤ 400 mm), rebound velocity of the impact barrier (≤ 2 m/s), and peak acceleration (≤ 40 g). These regulatory limits were adopted as the criteria for subsequent model validation.

A rear underride protection device of a domestic commercial truck was selected as the prototype. The key structural features were retained, while non-critical components were reasonably simplified to improve computational efficiency. The three-dimensional geometric model was developed using SolidWorks, followed by geometry cleanup, mesh generation, and property definition in HyperMesh. Considering that the present study mainly focused on the structural response of the RUPD and that

the deformation of the truck frame was relatively limited under the *investigated* conditions, the truck frame and impact barrier were modeled as rigid bodies. The thin-walled components of the RUPD were discretized using shell elements, and the material behavior was described using the piecewise linear plasticity model (*MAT_024) [19].

For the static regulatory evaluation, the maximum longitudinal displacements of the RUPD crossbeam under two-point and three-point loading conditions were 18.45 mm and 19.32 mm, respectively, both of which were significantly lower than the regulatory limit of 400 mm. The stress distribution obtained from the static validation is shown in Fig. 1.

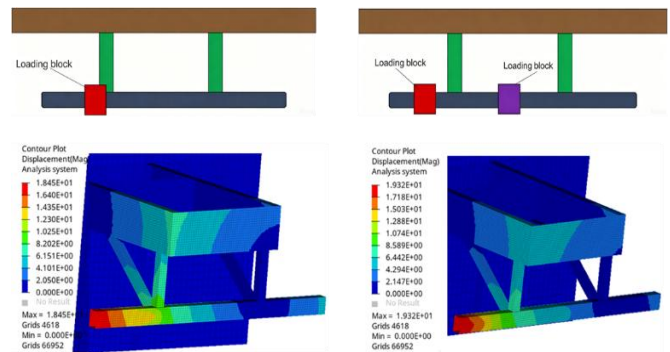


Fig. 1. Stress distribution under static validation conditions.

For the dynamic impact evaluation, the moving barrier impacted the RUPD at a velocity of 32 km/h according to the requirements specified in GB 11567-2017. After completing the finite element model development, the centroid acceleration history of the moving barrier, deformation response of the rear crossbeam, and velocity variation curve were obtained, as shown in Fig. 2.

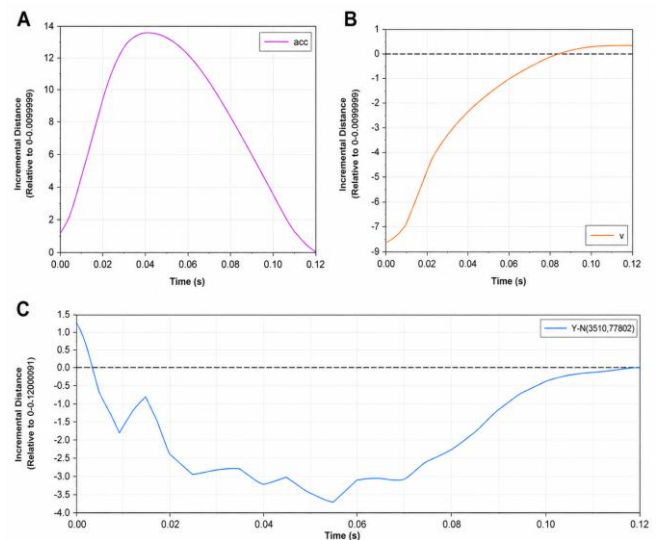


Fig. 2. Centroid acceleration history of the moving barrier, deformation response of the rear crossbeam, and velocity variation curve.

The simulation results showed that the peak acceleration of the barrier centroid was 13.67 g, which was below the regulatory limit of 40 g. The rebound velocity after impact was 0.25 m/s, satisfying the requirement of less than 2 m/s. In addition, no

overall detachment of the RUPD occurred, and the maximum longitudinal displacement of the crossbeam was 3.58 mm. These results demonstrate that the developed RUPD finite element model satisfies the specified regulatory requirements and can provide a reliable basis for subsequent full-vehicle crash simulations.

The passenger car model was obtained from the publicly available finite element model database of the National Crash Analysis Center (NCAC) and was used for subsequent crash compatibility analyses. The total mass of the vehicle model is 1420 kg, with overall dimensions of 4330 mm × 1710 mm × 1370 mm. The basic structural parameters satisfy the requirements for passenger car-truck rear-end crash compatibility investigations. The finite element preprocessing was conducted using HyperMesh. Quadrilateral shell elements were preferentially adopted for mesh discretization, while triangular elements were only introduced in geometrically complex regions for local transition. Mesh quality was controlled based on several criteria, including aspect ratio, skew angle, warpage angle, Jacobian value, and element size.

B. Development and Validation of the Passenger Car-Truck Rear-End Crash Finite Element Model

After the independent validation of the RUPD model and passenger car model, a 100% overlap passenger car-truck rear-end collision model was developed. The ground was modeled as a rigid wall, and the rear structure of the truck was fully constrained to maintain a stationary condition. The passenger car was assigned an initial velocity of 56 km/h and impacted the stationary truck along the global X-direction [20]. Gravity loading was applied in the Z-direction. Automatic contact algorithms were employed to define the self-contact of the passenger car, self-contact of the RUPD, and contact interaction between the passenger car and RUPD. The truck body was constrained using fixed boundary conditions.

To evaluate the numerical stability of the crash model and achieve a balance between computational efficiency and solution accuracy, several key simulation parameters were carefully defined. The time-step scale factor was set to 0.9, and the termination time of the simulation was set to 0.12 s. An IHQ=4 hourglass control formulation was adopted with the hourglass coefficient set to 0.1. The energy monitoring function was activated to record the relevant simulation responses. The results showed that the total energy remained stable throughout the collision process. The hourglass energy accounted for 4.18% of the total energy, which was below the commonly accepted 5% threshold in crashworthiness simulations. The contact sliding energy accounted for 1.97%, indicating no abnormal contact-related energy dissipation. Furthermore, the additional mass introduced by mass scaling accounted for only 0.62% of the total model mass, which was within the commonly accepted engineering control range of 5%. These results demonstrate that the developed crash model possesses acceptable numerical stability and can be applied for subsequent crash compatibility evaluation and structural optimization. The energy variation history and mass change during the entire collision process are presented in Fig. 3.

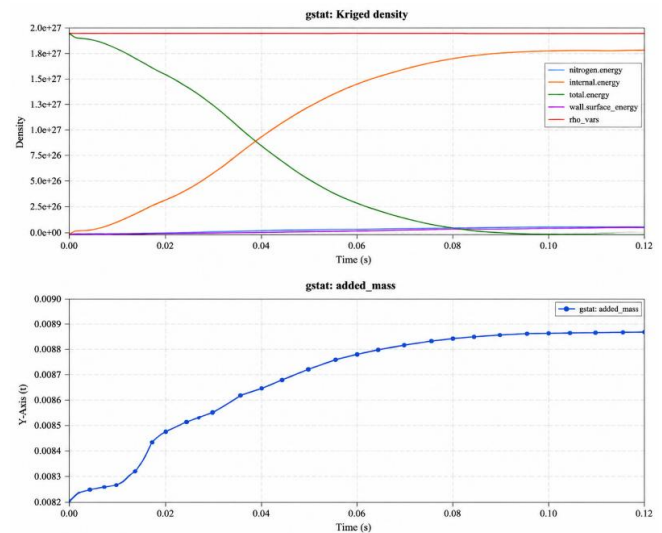


Fig. 3. Energy variation and mass scaling curves.

III. DEVELOPMENT OF CRASH COMPATIBILITY EVALUATION METRICS

Crash compatibility in rear-end collisions describes the degree of coordination among geometric matching, load transfer characteristics, and energy absorption behaviors between different vehicles during the collision process [21]. Existing studies have rarely provided a comprehensive evaluation method that simultaneously considers structural matching relationships and energy-sharing characteristics in rear-end collisions. To evaluate structural crashworthiness and damage characteristics during the collision process [22–23], this study considers both crash compatibility and crashworthiness requirements. The overall deformation behavior of the vehicles is analyzed to characterize the global impact response, while the deformation characteristics of key energy-absorbing components are further investigated to evaluate the structural crashworthiness performance [24].

A. Crash Response Analysis Under 100% Overlap Rear-End Collision Condition

Based on the developed full-vehicle finite element model, the structural response process at different impact stages under the 100% overlap rear-end collision condition was analyzed. The deformation patterns of the rear underride protection device (RUPD) and the front longitudinal beams of the passenger car at different impact stages are presented in Fig. 4.

The crash results indicate that the RUPD exhibits limited deformation participation during the initial impact stage. During the middle and later stages of the collision, plastic deformation develops in the brackets and connection regions, accompanied by a tendency toward local instability, resulting in reduced continuous load-carrying capability. Therefore, to quantitatively characterize rear-end crash compatibility from the perspective of structural interaction, an evaluation framework was established based on three aspects: energy distribution, intrusion mitigation, and occupant-related impact response.

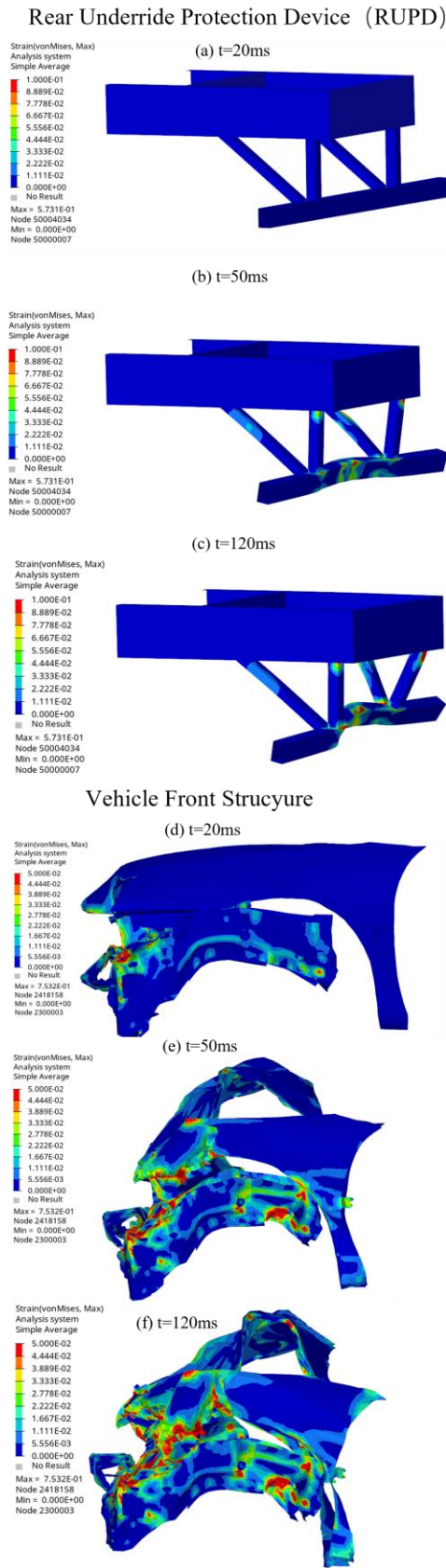


Fig. 4. Deformation patterns of the RUPD and passenger car front-end structure.

The selected evaluation metrics include the RUPD energy absorption ratio (η_R), the maximum intrusion of the passenger car front longitudinal beam (ΔL), and the peak longitudinal acceleration of the lower B-pillar (a_{max}).

B. Development of Crash Compatibility Evaluation Metrics

During rear-end collisions, geometric mismatches and differences in structural stiffness between passenger cars and heavy trucks can affect impact load transmission and structural deformation. The load-bearing capacity and deformation characteristics of the rear underrun protection device (RUPD) are therefore important design considerations, as they influence its stress distribution and energy absorption behavior [25–26].

To quantify the contribution of the RUPD to crash energy sharing, this study introduces the RUPD energy absorption ratio η_R . This parameter acts not only as a quantitative metric for evaluating the energy absorption efficiency of the RUPD structure, but also as a load-sharing indicator reflecting the crash compatibility of the two colliding vehicles. Specifically, E_R denotes the internal energy absorbed by the RUPD component assembly during the entire collision process, and E_c represents the internal energy absorbed by the energy-absorbing components in the front structure of the passenger car. The definition of η_R is given by Eq. (1):

$$\eta_R = \frac{E_R}{E_R + E_c} \times 100\% \quad (1)$$

The variation of internal energy absorbed by the truck rear protection structure and the passenger car front structure during the collision process is shown in Fig. 5.

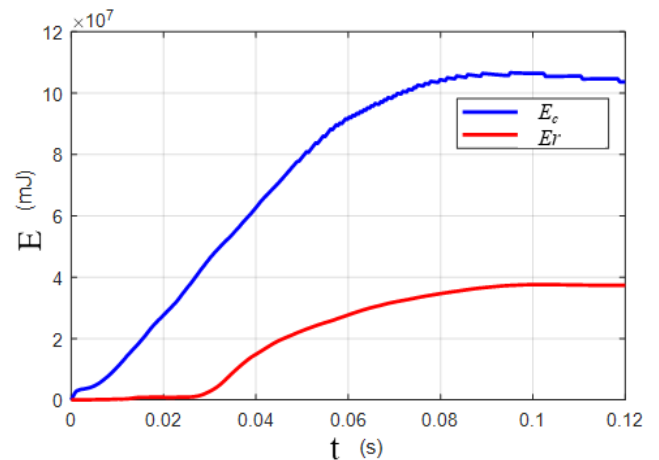


Fig. 5. Comparison of internal energy absorption between the passenger car front structure and truck rear protection device.

To evaluate the structural intrusion of the passenger car primary load-bearing path, the front longitudinal beam end nodes were selected as measurement points. The displacement histories along the global X-direction were extracted, and the maximum displacement was defined as the longitudinal beam intrusion ΔL . The variation of the X-direction displacement at the front longitudinal beam end is illustrated in Fig. 6. Under the baseline condition, the maximum intrusion ΔL was 173 mm.

The maximum intrusion of the passenger car front longitudinal beam along the vehicle traveling direction during

the collision was adopted as the intrusion evaluation metric, which is defined as:

$$\Delta L = \max(\Delta x_L(t)) \quad (2)$$

where, Δx_L represents the displacement of the front longitudinal beam measurement point during the collision.

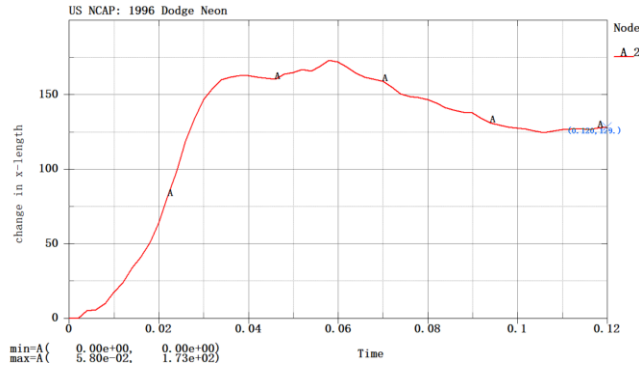


Fig. 6. Time history curve of X-direction displacement at the front longitudinal beam end.

In full-vehicle crash simulations, the vehicle acceleration history can reflect the buffering effect of load transfer and energy absorption processes to some extent and is closely associated with the impact environment experienced by occupants [27]. In this study, the structural node located at the lower B-pillar on the collision side of the passenger car was selected as the measurement point, and its longitudinal acceleration history $a_B(t)$ was extracted.

To ensure consistency among different design cases, the acceleration signal $a_B(t)$ was processed using a unified CFC60 filter. The absolute peak value during the main contact and structural response phase was selected as the acceleration evaluation metric $a_{\max}$, which is defined as:

$$a_{\max} = \max |a_B(t)| \quad (3)$$

where, a_B represent the beginning and ending times of the primary contact and structural response stage during the rear-end collision, respectively. The symbol $|\cdot|$ denotes the absolute value operation, and the unit of $a_{\max}$ is expressed as g.

IV. OPTIMIZATION DESIGN OF RUPD FOR CRASH COMPATIBILITY IMPROVEMENT

Vehicle crash optimization problems generally exhibit strong nonlinear characteristics, multi-variable coupling effects, and high computational costs. The complex relationship between design variables and crash responses makes it difficult to establish explicit analytical expressions. Therefore, to reduce the computational burden of finite element simulations and achieve coordinated improvement of crash performance, an optimization framework integrating finite element simulations, surrogate modeling, and multi-objective optimization algorithms was developed in this study.

A. Selection of Key Structural Parameters

To achieve a balance between computational cost and optimization accuracy, key structural parameters of the truck rear underidee protection device (RUPD) that significantly

influence crash compatibility were selected as design variables. Based on the structural characteristics and crash response mechanisms of the RUPD, three design variables were considered: the crossbeam thickness t , the ground clearance of the crossbeam h , and the distance between the RUPD and the rear end of the truck d . The design variable vector x is defined as follows:

$$x = [t, h, d]^T \quad (4)$$

In this study, the developed crash compatibility evaluation metrics were adopted as optimization objectives. A multi-objective optimization problem was formulated by maximizing the RUPD energy absorption ratio η_R and minimizing the front longitudinal beam intrusion (ΔL) and the lower B-pillar peak acceleration ($a_{\max}$). Additionally, the constraint $a_{\max} \leq 40g$ was introduced to ensure occupant impact safety while achieving a balance between crash compatibility improvement and vehicle structural response control. The mathematical formulation is given as follows:

$$\begin{aligned} \min \quad & a_{\max}(t, h, d) \\ \max \quad & \eta_R(t, h, d) \\ \min \quad & \Delta L(t, h, d) \\ \text{s.t.} \quad & a_{\max} \leq 40g \\ & 4mm \leq t \leq 8mm \\ & 330mm \leq h \leq 500mm \\ & 0mm \leq d \leq 305mm \end{aligned} \quad (5)$$

To ensure the comparability and consistency of the evaluation results across all sample cases, the extraction rules, node set definitions, and data statistical methods for the three indicators are detailed in Table I. The extracted results are automatically compiled into the sample database, thereby guaranteeing the consistency and reproducibility of the sample data and providing a reliable data basis for surrogate model training and optimization analysis.

TABLE I. DEFINITION, DATA SOURCE AND STATISTICAL RULES OF RESPONSE INDICATORS

Indicator	Symbol	Definition	Data Source	Statistical Rule
RUPD energy absorption ratio (%)	η_R	$\eta_R = \frac{E_R}{E_R + E_C}$	matsu m	Cumulative internal energy at the end of crash
Maximum front longitudinal beam intrusion (mm)	ΔL	$\Delta L = \max(\Delta x_L(t))$	nodout	Maximum displacement during the entire crash process
Peak acceleration of lower B-pillar (g)	$a_{\max}$	$a_{\max} = \max_{t \in [0, T]} a_x^{CFC}(t)$	nodout	Peak value after CFC filtering

B. Experimental Design

In this study, the optimal Latin hypercube sampling (Optimal LHS) method was adopted to generate sample points.

Compared with conventional sampling strategies, Optimal LHS provides better spatial uniformity and distribution balance within the design space, effectively avoiding local clustering of samples and improving sampling efficiency and the stability of surrogate model construction. Within the predefined design variable ranges, the MATLAB platform was employed, and the sample distribution was optimized using the maximin criterion. A total of 30 training samples were generated. The distribution of the design variables is illustrated in Fig. 7.

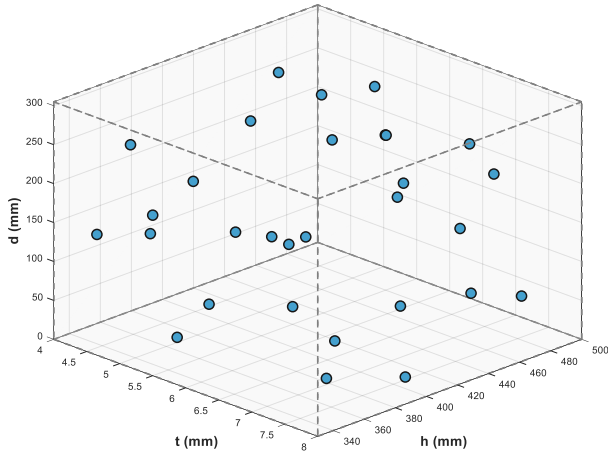


Fig. 7. Three-dimensional distribution of OLHS samples.

To ensure that the surrogate model can accurately capture nonlinear responses while maintaining good generalization capability, a Kriging approximation model with a constant regression term was adopted. A Gaussian correlation function was selected to characterize the spatial correlation between sample points [28]. Based on the sample data obtained from the design of experiments (DOE), the proposed approach can reduce prediction errors caused by overfitting to training samples and improve the reliability of the surrogate model in subsequent multi-objective optimization.

To further evaluate the robustness of the developed model, an independent validation dataset was employed to assess the prediction accuracy of the Kriging model. The coefficient of determination (R^2) was selected as the evaluation metric, and its calculation equation is given as follows:

$$R^2 = 1 - \frac{\sum_{i=1}^n (y_i - \hat{y}_i)^2}{\sum_{i=1}^n (y_i - \bar{y})^2} \quad (6)$$

where, y_i represents the actual response value, denotes the predicted response value obtained from the Kriging model, is the mean value of the actual responses, and n represents the number of validation samples.

In engineering applications, a surrogate model is generally considered to possess sufficient predictive capability when $R^2 > 0.9$. In this study, the developed Kriging surrogate models were evaluated for the prediction of three crash compatibility responses, including the peak acceleration at the lower B-pillar ($a_{\max}$), the RUPD energy absorption ratio (η_R), and the maximum intrusion of the front longitudinal beam (ΔL). The prediction errors of the surrogate models are illustrated in Fig. 8.

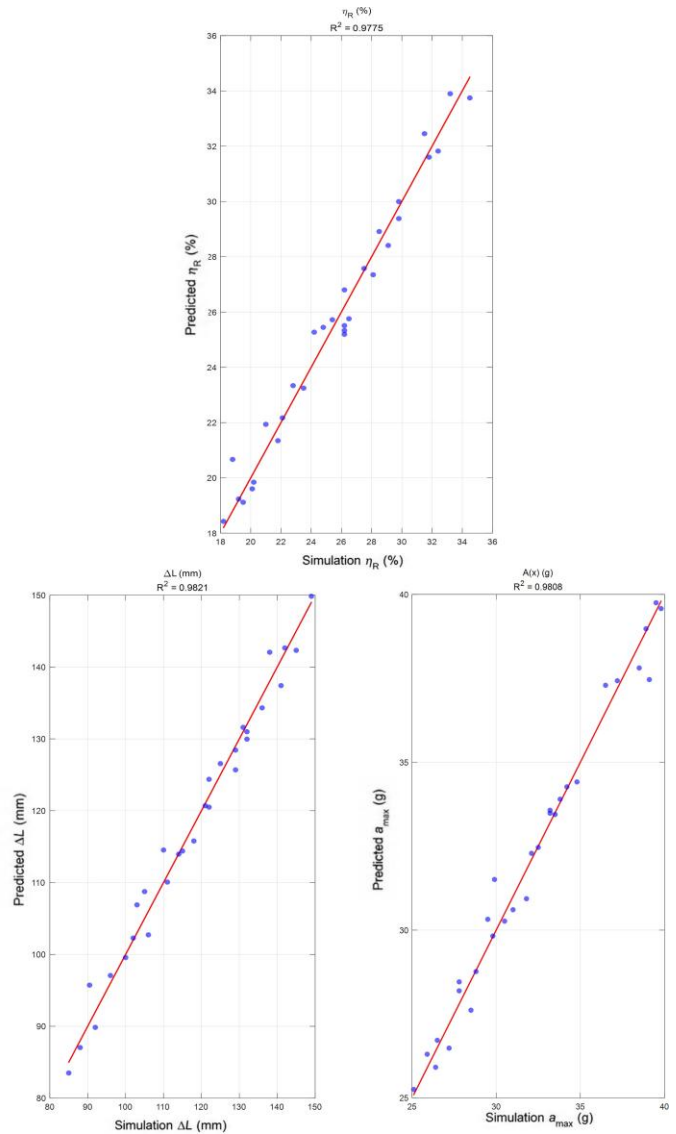


Fig. 8. Scatter plots of surrogate model prediction errors.

The results show that the coefficients of determination (R^2) for $a_{\max}$, η_R , ΔL are 0.9808, 0.9775, and 0.9821, respectively. These results demonstrate that the developed Kriging models can accurately characterize the mapping relationship between the design variables and crash responses, providing a reliable basis for subsequent multi-objective optimization.

To further investigate the sensitivity of design variables to the output responses, two-dimensional response surfaces were constructed based on the validated Kriging surrogate models. During the response surface generation process, one design variable was fixed while the other two variables were varied within the design domain. The fixed values were selected as the baseline parameters: $t=t_0=6$ mm, $h=h_0=475$ mm, $d=d_0=100$ mm. This approach enables a clear visualization of the main effects and interaction effects of design variables, providing theoretical support for subsequent multi-objective optimization analysis.

The results indicate that η_R generally increases with increasing crossbeam thickness t , while variations in h and d

introduce certain nonlinear fluctuations. The maximum intrusion ΔL decreases overall with increasing t within the investigated design space but exhibits nonlinear coupling effects with h and d . The peak acceleration $a_{\max}$ is highly sensitive to t . An increase in t may increase $a_{\max}$ due to enhanced structural stiffness and stronger load transmission, whereas insufficient t may also result in increased acceleration owing to premature RUPD failure and reduced energy absorption capability. Therefore, coordinated optimization of h and d is required to maintain effective RUPD load-carrying capacity while simultaneously controlling ΔL and $a_{\max}$. Consequently, a multi-objective optimization approach is necessary to achieve an optimal balance among the three competing objectives and obtain an overall improved structural configuration.

Based on the developed surrogate models, the NSGA-II algorithm was employed for multi-objective optimization. During each iteration, NSGA-II evaluates and selects individuals through non-dominated sorting and crowding distance calculations. New populations are subsequently generated through selection, crossover, and mutation operations, gradually converging toward the Pareto-optimal front and producing a set of non-dominated solutions representing the trade-offs among different objectives [29].

The parameters of NSGA-II were configured as follows: the population size was set to $N=60$, the maximum number of generations was set to $G=100$, the crossover probability was $P_c=0.9$, and the mutation probability was $P_m=0.1$. The optimization process was terminated when the Pareto front reached a stable state over successive generations or when the predefined maximum number of iterations was reached.

Based on the obtained Pareto solutions and the trade-off relationships among optimization objectives, the three-dimensional distribution and two-dimensional projection of the DOE samples and Pareto solutions in the objective space were plotted, as shown in Fig. 9.

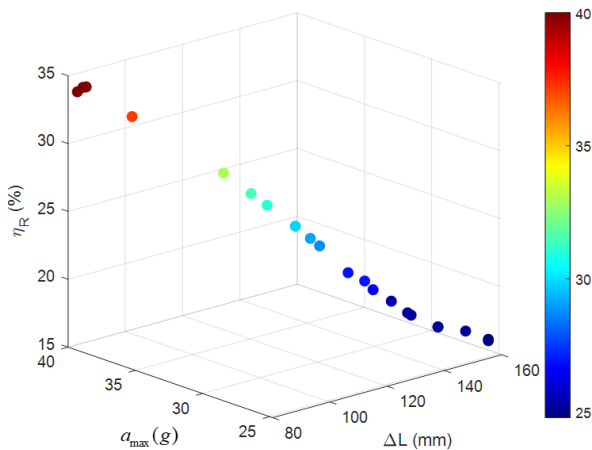


Fig. 9. Distribution of Pareto optimal solutions.

The two-dimensional projection of the Pareto non-dominated solutions shows an approximately one-dimensional band distribution, indicating strong mechanical coupling and competition among the optimization objectives. The results reveal that η_R and ΔL exhibit a negative correlation trend within the current design space. Increasing the energy absorption

contribution of the RUPD can effectively reduce the load burden on the passenger car longitudinal beams and mitigate structural intrusion.

A negative correlation trend is also observed between ΔL and $a_{\max}$, reflecting the inherent trade-off between intrusion reduction and impact pulse characteristics. Specifically, reducing structural intrusion generally requires stronger load transmission, which may deteriorate the acceleration response. Meanwhile, η_R and $a_{\max}$ show a positive correlation tendency, indicating that increasing RUPD stiffness to enhance energy absorption capacity may simultaneously intensify load transfer to the passenger compartment. Therefore, the three optimization objectives are mutually constrained, and it is difficult to achieve simultaneous optimality for all individual objectives.

Based on the distribution characteristics of the Pareto non-dominated solutions, and considering both structural performance improvement and engineering feasibility, the final optimal design was selected using the minimum Euclidean distance criterion to the ideal point. The selected compromise solution significantly improves the energy absorption capability of the RUPD while effectively reducing passenger car structural intrusion under the constraint of the B-pillar peak acceleration.

Considering practical manufacturing and assembly tolerances, the continuous optimization variables obtained from the optimization process were rounded to engineering-feasible values. The final optimized design parameters are determined as follows: $t = 6.4$ mm, $h = 390$ mm, and $d = 38$ mm.

V. RESULTS AND DISCUSSION

To evaluate the effectiveness of the proposed optimization design in improving rear-end crash compatibility, finite element simulation results before and after optimization were compared. The key evaluation responses of the two design schemes are summarized in Table II. The results indicate that the energy absorption ratio of the RUPD increased from 26.3% to 33.8%, representing an improvement of approximately 28.5%. Meanwhile, the maximum intrusion of the passenger car front longitudinal beam was reduced from 173 mm to 117 mm, corresponding to a reduction of 32.4%.

The optimized design improved the energy-sharing capability of the RUPD while maintaining the vehicle acceleration response within an acceptable engineering range. Consequently, the overall crash performance was improved compared with the baseline design.

TABLE II. COMPARISON OF KEY RESPONSE INDICATORS BEFORE AND AFTER OPTIMIZATION

Design	t (mm)	h (mm)	d (mm)	η_R	ΔL (mm)	$a_{\max}$
Baseline	6.0	475	100	26.3%	173	32.9
Optimized	6.4	390	38	33.8%	117	38.4

Fig. 10 presents the deformation and strain distributions of the optimized RUPD and the passenger car front structure. Compared with the baseline configuration, more pronounced plastic deformation and high-strain regions are observed in the connection areas between the diagonal supports and the upper/lower load-bearing beams of the optimized RUPD. This

indicates that the energy absorption participation of the RUPD has been enhanced.

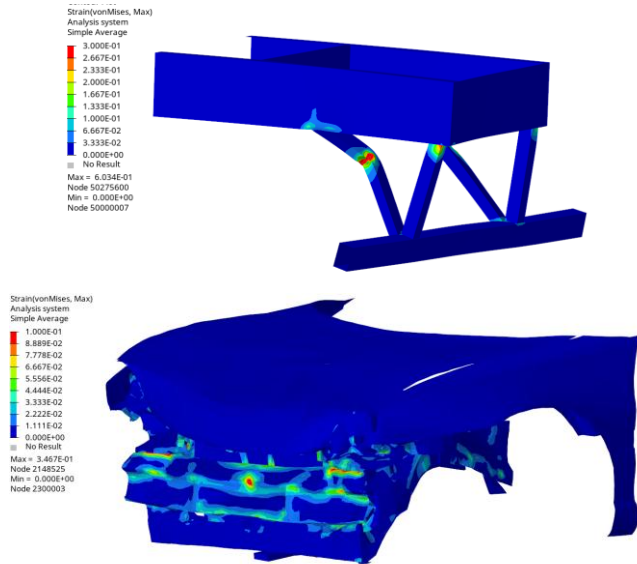


Fig. 10. Deformation and strain distributions of the optimized RUPD and passenger car front structure.

Meanwhile, the deformation process of the passenger car front energy-absorbing components becomes more progressive and controllable, resulting in reduced intrusion of the front longitudinal beams. The optimized RUPD can therefore absorb a larger proportion of the crash energy, reducing the energy burden transferred to the primary load-bearing structures of the passenger car. As a result, the longitudinal beam intrusion is effectively suppressed, and crash compatibility is improved.

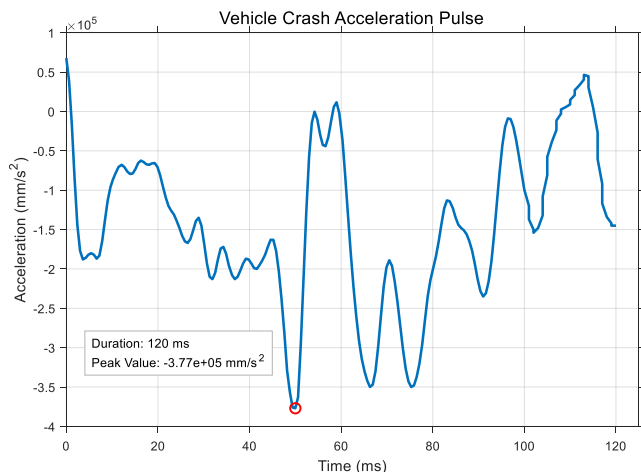


Fig. 11. Optimized B-pillar acceleration histories.

Fig. 11 presents the X-direction acceleration time-history curves and their peak values at the measuring points below the optimized B-pillar. From the perspective of pulse characteristics, the optimized acceleration curve exhibits reduced oscillatory behavior compared with the baseline condition. This improvement indicates that the optimized configuration, particularly the adjusted crossbeam height ($h=390$), establishes a more direct and stable load transfer path,

thereby reducing complex structural vibrations caused by improper geometric matching during impact.

Although the peak acceleration slightly increases after optimization, the maximum value remains below the typical engineering safety limit of 40 g, indicating acceptable occupant impact performance. This result demonstrates the capability of the multi-objective optimization framework to achieve a compromise between structural intrusion control, energy absorption enhancement, and occupant impact response. Therefore, the selected Pareto-optimal solution possesses practical engineering significance for rear-end crash compatibility design.

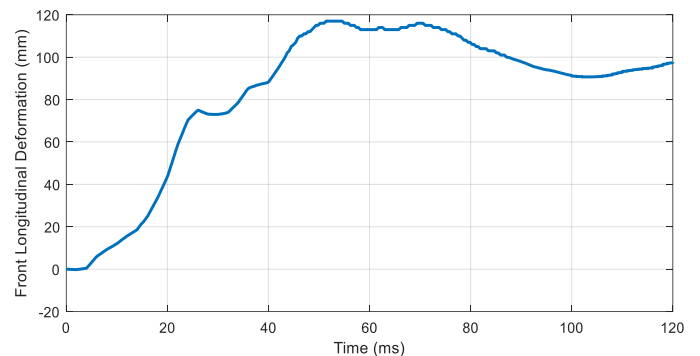


Fig. 12. Optimized intrusion response curves.

Fig. 12 presents the X-direction displacement time-history and the maximum intrusion of the optimized passenger-car front longitudinal beam. The results show that the baseline design exhibits a faster intrusion development rate and a considerably larger final intrusion value.

After optimization, the intrusion progression is effectively restrained, and the maximum intrusion is significantly reduced. This confirms that the optimized RUPD configuration improves the compatibility between the truck rear structure and passenger car front structure, thereby reducing the risk of severe structural intrusion.

The proposed methodology, which integrates parametric modeling, surrogate model prediction, and multi-objective optimization, effectively enhances the protective performance of the RUPD and improves rear-end crash compatibility. The proposed framework provides a numerical basis for comparative RUPD design optimization under the investigated impact conditions.

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