Research Article

Optimization Design on Functionally Graded Cem for Trains Based on LPM Model with Calibrated Parameters

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Lumped parameter modeling (LPM) combined with optimization techniques is an efficient approach for parametric configuration design of energy absorption to improve crashworthiness performance during train collision. This work proposed a simplified model by introducing a bar element to consider the influence of the carbody in a collision process. The optimization method is applied to calibrate the equivalent parameters of the bar element. Bar elements with calibrated parameters are adopted in establishing a one-dimensional (1D) model for the train crash. Subsequently, a novel crash energy management (CEM) mode with functionally graded energy (FGE) configuration is introduced to the train crash model for improving crashworthiness performance. The influence of parameters in graded function on interfacial force and peak acceleration is investigated and optimal design parameters are obtained by Nondominated Sorting Genetic Algorithm (NSGA-II). It is concluded that considering the behavior of the carbody can improve the accuracy of LPM in predicting the longitudinal response, and the gradient CEM is a potential energy configuration mode to improve the crashworthiness of the train in a collision.

1. Introduction

Passive safety design is the second protection for a train in case of the failure of active safety protection during a collision. It has been a critical issue to improve the crashworthiness and provide more survival possibilities by the reasonable design of energy absorption parameters. The first is to obtain enough response information of train during an occasional collision accident, as actual train collision experiments are always inadvisable as a result of poor repeatability and substantial consumption of resources. Numerical simulation based on explicit dynamical analysis technique is preferred in reproducing the dynamical response of train collision as various results and good repeatability. Nonetheless, the numerical analysis is still of disadvantages in several respects such as long computing time, large storage space, low analysis efficiency, especially for a large-scale model with numerous degrees of freedom.

Large numbers of attempts to improve computing efficiency for vehicle collision models have been implemented.

LPM is the main method widely applied for reducing model scale and improving solution efficiency in crash analyses. It forms formulating equations of systematic motion (including but not limited to spring, damper, and mass) and obtains structural responses. Different forms are configured and used in analyzing the crash response. For instance, the basic lumped mass model consists of masses and linear springs [1], LPM including beam elements, plastic hinges, nonlinear contact spring elements, and rigid solids [2], and models using nonlinear spring and damper [3, 4]. However, considering the mathematical complexity of a crash event representation, it was concluded that the determination of the force elements in the mathematical model has a remarkable influence on response prediction [5–8]. The LPM is largely lacked in providing an accurate response with no additional parameter calibration. It is found that parameters calibration based on experiments or FEA results is effective in enhancing the accuracy in analytical results of LPM compared to the full-sized model. The most used data in parameters calibration of LPM in previous works include

displacement, velocity, and acceleration pulse of the car measured by actual test during a car crash, where the energy by structural deformation is merely considered [9–13]. However, as a critical indicator of crashworthiness problems, the neglect of energy absorption in parameters calibration will lower the accuracy of LPM in collision analysis. On the other hand, due to the fact that the full-sized experiment of railway vehicles is always limited by multiple difficulties on cost and measurement, the parameters calibration of LPM for large structures like railway vehicles is highly dependent on the FEA results. Markiewicz et al. [14] utilized rigid elements and nonlinear beam elements and springs to simulate the behavior of a railway driver's cabin. The classical spring-mass model was once used to investigate the design energy requirements of crashworthy vehicles for understanding the collision behavior of two trains with a linear model, particularly the interaction between the vehicles [15]. Milho et al. [16, 17] presented a validated multibody-based model for the design of train crashworthy components including the anticlimber devices and discussed the modeling assumptions and its suitability in application to the design of train crashworthy components. Xie et al. [18] realized the model scale is reducing by substituting the middle part of the carbody with no plastic deformation using mass and beam elements, which has more accuracy than the multibody model and lowers computing time than the full-size finite element model. Nonetheless, computing efficiency is still an obstacle to the investigation and optimization of crashworthiness parameters. Tang et al. [19] obtained the stiffness and damping of the nonlinear spring in the spring-mass model for the train impact process by machine learning method, as the mass-spring model with validated parameters is unable to get the response of the carbody, which cannot deal with the train crash case of higher speed. Elmarakbi et al. [20] developed a mathematical model to optimize the crashworthiness in a frontal crash scenario and proved the application ability of the mathematical model in optimization studies. Almost above simplified models for train collision neglect the response of the carbody; however, the response of the carbody is a critical indicator of crashworthiness assessment, and the dynamical behavior of the carbody is necessary for collision process analysis.

Compared to the full-sized crash model of the train, the obvious advantage of the simplified model with parameters validation is to provide accurate analysis results using less computing time, with the advance in analyzing efficiency derived from model simplification, which makes crashworthiness optimization feasible and practical. CEM concept is early proposed to design the energy distribution [21]. In a uniform energy distribution, the equivalent energy absorber is equipped at connected interfaces. The energy at the first interface is obviously higher than other interfaces. It is believed that increasing the energy absorption at the first interface of the train is an effective way to enhance the crashworthiness of the train. Whereas the train is a typical multibody system, the energy absorption presents noteworthy progressive characteristics along with the propagation of impact wave. The energy dissipation path is inefficient when the energy at different interface of the train

is distributed in uniform manner, which has been validated by the valuable work from Dias and Pereira [21, 22], and the concept of that the energy absorption at adjacent interfaces should be configured as a graded manner was found theoretically based on the one-dimensional wave propagation model [23], which indicated a more advisable energy distribution, but the detailed parameters for graded energy distribution are not yet studied, and the parameters optimization of CEM discretely were conducted without controlled function tending to be time consuming due to much variables.

On the foundation of previous works, a simplified method based on a nonlinear bar element is proposed in modeling the carbody to realize the characterization of its crush behavior during train collision. Its parameters are calibrated by optimization, using the energy absorption as one of the responses in parameters calibration to improve the accuracy of equivalent parameters of LPM. The 1D collision model established using the nonlinear spring, mass, and bar element provides the ability to investigate the energy configuration parameters at the early stage of CEM design for the train. In order to propose more efficient energy absorption for the train, a novel energy absorption mode ruled by gradient function is presented to improve the crashworthiness performance. Based on the simplified model, the influence of gradient parameters in the control function on crashworthiness is studied, and optimal energy distribution parameters are obtained by solving the optimization problem aimed to lower the peak interfacial force, which realizes the continuous parameters optimization for train collision issue.

2. Simplified Model and Parameters Calibration

2.1. Single Carbody Analysis. In the simplified modeling of train collision for numerical analysis, the carbody of the vehicle is generally simplified to lumped mass element as a consideration of no deformation in a collision. However, the train is actually a multibody energy dissipation system in a collision, and the shock wave propagates from the front collision interface to the end interface. Each vehicle will be subjected to multiple loading of compression and tension with the reflection of the stress wave. The characteristics of the carbody need to be taken into consideration in crash analysis using a simplified model.

The carbody of the Tc car and M car is manufactured by assembling aluminum alloy extruded profiles with seam welding, which includes four components of the underframe, sidewall, endwall, and roof. The width and height of Tc and M are 3300 mm and 3900 mm, while the length is 21000 mm and 25000 mm, respectively. The mass of Tc and M is 51500 kg and 54200 kg, respectively. The distance between the center points of the two bogies is 17500 mm. The distance between the height of the coupler and underframe for M car is 325 mm, and that for Tc car is 260 mm and 325 mm at the head and end location, respectively. The mass of bogie for Tc and M is 7500 kg and 9800 kg, respectively. The Tc car and M car are full-sized modeled due to their different structural characters. The coupler of the Tc car is

generally designed with failure when the threshold load is reached; subsequent to the failure, the front end of the carbody comes to bear the crushing force. The compression of the first vehicle is more coincident to the axial loading, while the coupler fixed on the M car is not equipped without failure function. The vertical distance from the fixation center of the coupler to rail is 1000 mm. With the special structure, the impact force by collision is transferred from the coupler to the draft sill then to the underframe. As a result of the difference in the height of the coupler and underframe, it generates a certain bending moment to the carbody of the M car in a collision.

As a consideration of the difference in structure and loading characteristics, the head vehicle Tc and the intermediate vehicle M are selected as typical vehicles to analyze the response process of carbody subjected to impacting load. The numerical analysis is carried out using explicit analysis code PAM-CRASH to obtain dynamical response including longitudinal deformation, impact force, energy absorption of carbody, and the velocity of a rigid wall. Figure 1 shows the finite element analysis model of vehicle Tc and M, including the carbody, coupler, bogie, and track. The carbody is modeled by using Belytschko–Tsay four-node shell elements with five integration points along the thickness direction, and the coupler is simulated using the spring element. The platform forces of the coupler for Tc and M are both 1500 kN. The compression stroke is 680 mm and 340 mm, respectively. The bogie and wheel set are modeled using a hexahedral solid element and set as a rigid body due to no participation in crash energy absorption. The number of elements for Tc and M is 576854 and 653234, while that of nodes is 666578 and 772826, respectively.

Nonlinear 6DOF Spring Dashpot (MAT 220) in PAM-CRASH is created to describe the energy absorption property of the coupler. Carbody is modeled using Elastic-Plastic-Shell (MAT105) in PAM-CRASH. The carbody is manufactured with two aluminum alloys of 6005A-T6 and 6082A-T6. 6005A-T6 is mainly used for extruded section situated side wall, underframe, and roof, while 6082-T6 to slab structure such as end wall. The thickness of the thickest plate is 2.5 mm. Young's modulus is 70 GPa for both, Poisson's ratio is 0.3 for both, and plastic tangent modulus is 0.81 GPa and 0.83 GPa, the yield strength is 0.26 GPa and 0.29 GPa, respectively.

The Master-Slave-Node-Segments contact model of No. 33 is established to describe the contact behavior between wheelset and track. The same contact type is considered between the coupler and rigid wall block; all the contact thicknesses are set to 0.5. Only the initial external faces of the solid elements are taken into account, and the node-edge contact is active. The initial penetration is set to be removed by adjusting the local contact thickness. The precise contact pressure computation is active. Considering the rolling friction contact relation between wheelset and track in a general rear-end train collision, the friction is neglected and its column coefficient is set to 0. In general, its friction coefficient is not enough to generate influence to the crash process, and the crash energy is mainly dissipated by a special safe system such as energy absorbing structure and

FIGURE 1: Finite element model of a single vehicle.

coupler. In order to illustrate the effect of friction, finite models of a single carbody were reanalyzed using the wheel-rail friction coefficient of 0.1 [24, 25]. It is found that the energy dissipated by friction is less than that by structural deformation during a crash. The energy dissipated by friction during the collision to a rigid wall is about 4 kJ and 3 kJ for Tc car and M car, respectively. Compared to corresponding energy absorbed by structural deformation, the relative ratio of energy by wheel-rail friction is 0.27% and 0.33%, respectively. The axial deformation of the carbody is approximately calculated by the coordinate of two nodes at the end location of the carbody. Also the impacting force needs to be output. The rigid wall block with a mass of $20t$ is set to crash the vehicle at an initial speed of 10 m/s. All the model is subjected to the gravity acceleration field of 9.8 m/s^2 , and the track is constrained at six freedoms.

The convergence analysis of the FE model was carried out by using different mesh sizes of 20 mm, 40 mm, 80 mm, and 120 mm, respectively. The full-sized crash models with different mesh sizes for Tc and M were solved on HP Z840 Workstation (CPU E5-2670, 2.3 GHz, 64 GB RAM). As shown in Figure 2, the stiffness curve tends to be converged with the reduction of mesh size, and the consuming CPU time of analysis should be considered for a balance between efficiency and accuracy. So the mesh size for the final full-sized crash model is determined as about 20–40 mm to fulfill the common requirements of efficiency and solution accuracy for subsequent analysis.

2.2. Simplified Model Description. It is obviously difficult to obtain the theoretical solution of the stiffness and damping parameters for irregular and complex structures like the carbody in a certain direction. One feasible way is to get an approximate numerical solution by combining the optimization method with the finite element analysis.

In order to obtain the equivalent parameters of the nonlinear bar element for characterizing the longitudinal behavior of the carbody under a compression load, the crash model of the vehicle is simplified, as shown in Figure 3, actual carbody is substituted by a nonlinear bar element, the coupler spring and rigid wall are retained while the bogie and track are removed. The boundary condition of the simplified model is completely consistent with the actual full-sized model in Section 2.1. The nonlinear bar element shown in Figure 3 can be regarded as a generalized Kelvin model, of which the mechanical behavior of spring and

FIGURE 2: Convergence analysis of longitudinal stiffness for the carbody. (a) Tc. (b) M.

FIGURE 3: Simplified model for a single vehicle.

damper can be defined by a nonlinear function [26]. Its axial force is expressed as equation (1); the first and second terms denote nonlinear spring force and nonlinear damping force subjected to an impacting load, respectively:

$$F(\delta) = K(\delta) * \delta + C(\dot{\delta}) * \dot{\delta}, \quad (1)$$

$$\delta = L - L_0, \quad (2)$$

$$\dot{\delta} = \frac{d(L - L_0)}{dt}, \quad (3)$$

where the longitudinal deformation of the bar element is calculated by the current length L and the initial length L_0 , $\dot{\delta}$ is the axial deformation rate, K represents the nonlinear function of stiffness with axial deformation, while C describes the relation between damping and deformation rates. The equivalent parameters of the bar element in the simplified model, the stiffness and damping, can be captured by reaching the consistency of the longitudinal response of the rigid wall from the simplified model and full-sized model. The parameter calibration of a simplified model for a

single vehicle subjected to axial compression loading during collision is realized.

2.3. Parameters Calibration. The equivalent parameters calibration problem of the carbody constructed by a simplified bar element can be considered as a typical inverse analysis. By matching response data derived from the simplified model and actual model, it is feasible to solve the inverse analysis problem adopting the optimization method. At first, it is necessary to extract the response information from two kinds of the model mentioned above, here. Two types of responses were taken into account for data matching, including the energy absorption of the carbody and velocity of the rigid wall during the vehicle collision with a rigid wall. Subsequently, the Sum of Squared Differences (SSD) of two responses can be calculated using the results from the simplified 1D model and actual 3D model via the following equation:

$$SSD = \sum_{i=1}^n [Y(T_i, K_i, C_m) - Y_i]^2, \quad (4)$$

where i represents the sequence number of the data point on the response curve, n is the total number of data points. T_i is the output moment of i th data, K_l and C_m are the 1st value of the stiffness curve and m th value of the damping curve, respectively. $Y(T_i, K_l, C_m)$ and Y_i denote the current response and desired response value, respectively, herein including energy absorption of the carbody and velocity of the rigid wall.

It is evident that parameters of stiffness and damping have a remarkable effect on SSD, so it is the key to find the most suitable parameters by solving the inverse analysis problem. Here the optimization method is selected to adjust parameters in realizing the best data matching. The optimization objective is minimizing the SSD of collision response. K_l and C_m are resolved as the design variables for the optimization problem. There is a total of 10 points selected from the stiffness and damping curves of the carbody. The initial stiffness curve is shown in Figure 2, and the damping curve is assumed by referring to literature [19].

Then the optimization problem for inverse analysis on equivalent parameters of the carbody can be mathematically described as follows:

$$\begin{aligned} \min \quad & \text{SSD} \\ \text{s.t.} \quad & 0 \leq K_l \leq 12000 \quad l = 1, 2, \dots, 10 \\ & 0 \leq C_m \leq 12000 \quad m = 1, 2, \dots, 10. \end{aligned} \quad (5)$$

NSGA-II algorithm is selected in solving the optimization problem to get the optimal stiffness and damping parameters. In order to validate the accuracy of the simplified 1D model with optimal parameters in predicting collision response, of which dynamical response results are compared to that of the actual 3D model, including the velocity and displacement of the rigid wall and energy absorption and longitudinal deformation of the carbody. As shown in Figure 4, the responses from the 1D model agree closely with those from the 3D model. What's more, it is found in Figure 5 that the response of the carbody constructed using a 1D model also presents a good coincidence with the 3D model. Table 1 lists the results comparing energy absorption and longitudinal deformation of carbody derived from a 1D simplified model and 3D full-sized model. The prediction error of the 1D model of T_c and M in longitudinal deformation is -6.18% and 8.42% , while that of energy absorption is 2.66% and 9.83% , respectively. The relative error of the 1D model for T_c carbody response is slightly lower than that of M carbody in predicting collision, which is largely due to the structural difference between T_c and M carbodies. The two end couplers of T_c carbody more than M are much closer to the concentricity situation, so there is a certain moment generated on the carbody of M subjected to collision. In this work, a simplified 1D nonlinear element is established to mainly focus on the longitudinal energy distribution, so it does not have enough consideration of the moment effect as its slight effect on the prediction accuracy. The bar element is more suitable to characterize the mechanical behavior of the carbody under pure axial compression load.

3. Collision Analysis for Simplified Train Model

3.1. Collision Model. As mentioned above, the 1D bar element can be used to simulate the carbody behavior with an acceptable accuracy during a single vehicle collision. Nonetheless, it is necessary to revalidate its applicability when introduced to train collision analysis. The 1D train collision model consists of four vehicles that are formed in terms of responding 3D actual model, where the carbody and coupler are simulated with a 1D bar element and nonlinear spring element, respectively. Figure 6 shows the train models for collision analysis established from different perspectives. The analytical scenario is set as a frontal collision between two trains with a relative speed of 36 km/h in terms of the C-I category in EN15227 standard [27].

3.2. Response Comparison with 3D Model. In order to validate the accuracy of the 1D train model using calibrated bar elements in analyzing longitudinal collision, the 1D model is compared to the 3D model from the response of velocity trend and impact force at each interface. Figure 7 shows the comparison of the velocity curves of different vehicles from the moving train under collision conditions. It is apparent that the velocity curves analyzed by the 1D model agree well with that of the 3D model. As the energy absorbing components and coupler completely collapsed, the velocity of vehicle A_1 and A_2 fluctuates several times near the value of 5 m/s, at which moment the carbody stands a larger impacting load, and the closeness between velocity curves obtained from the 1D model and 3D model also reflects the ability of simplified bar element in characterizing the longitudinal characteristics of carbody.

The interfacial force is a critical indicator to evaluate crashworthiness performance during train collision, so it is essential to validate the calculation accuracy of the simplified 1D model proposed. Figure 8 compares the impact forces of the moving train obtained from the 1D model and 3D model at the connectional interfaces S_3 to S_4 . Corresponding to Figure 7, the full stroke of the coupler is completed firstly at the interface of S_1 and S_2 , so there appears a large peak force and it decreases with the end of the collision process. While there is still a residual stroke at the interfaces S_3 and S_4 , so its peak of the interfacial force does not exceed the stable crush force of the coupler.

As shown in Figure 8, the interfacial forces predicted by the 1D model present good coincidence with that by the 3D model on both peak value and curves trend. For instance, the peak forces at interfaces S_1 and S_2 by the 3D model are 4262 kN and 4361 kN, while those by the 1D model are 4744 kN and 4481 kN, respectively. The prediction errors of peak force at interfaces S_1 and S_2 by the 1D model are 11.31% and 2.75%, respectively. The relative error of peak force at interface S_2 is significantly lower than that of S_1 .

4. Optimization Design of Crash Energy Distribution

4.1. Functionally Graded Energy Absorption. CEM is a concept widely adopted in determining the distribution of

FIGURE 4: Comparison of velocity and displacement by the 3D model and calibrated 1D model. (a) Tc (b) M.

FIGURE 5: Comparison of energy and deformation by 3D model calibrated and 1D model. (a) Tc (b) M.

TABLE 1: Response comparison of the 1D model and 3D model.

Response	Tc			M		
	1D	3D	Error (%)	1D	3D	Error (%)
Longitudinal deformation (mm)	27.3	29.1	-6.18	52.8	48.7	+8.42
Energy absorption (kJ)	463	451	+2.66	402	366	+9.83

FIGURE 6: 3D finite element 1D model used for train collision analysis.

FIGURE 7: Vehicle velocity comparison of the 1D model and 3D model. (a) A1. (b) A2. (c) A3. (d) A4.

energy absorption at different interfaces to improve the crashworthiness of the train. In general, the energy absorption configuration at middle interfaces such as S_2 to S_6 is equal except the end location of S_1 and S_8 . Different from the uniform energy absorption of the traditional CEM scheme, it is found that the nonuniform energy absorption tends to bring obvious improvement in energy absorption efficiency. As shown in Figure 9, a new CEM design with nonuniform energy distribution, named FGE, was proposed to improve the crashworthiness performance of the train during a collision. The maximum and minimum energy absorption was configured at interfaces S_1 and S_4 considering the symmetrical operation of the train, respectively.

The energy absorption of interfaces S_1 and S_4 can be calculated by the following equations:

$$E_{\max} = S_{\max} * F_{\text{ini}} * D_{\max}, \quad (6)$$

$$E_{\min} = S_{\min} * F_{\text{ini}} * D_{\min}, \quad (7)$$

where E , S , and D denote the energy distributed, the scale factor of crush force, and corresponding crush stroke at connection interfaces in Figure 9, respectively. The symbol with subscript max and min represents the parameters for interfaces S_1 and S_4 , respectively. The value of $D_{\max}$ and $D_{\min}$ is 680 mm and 340 mm, respectively. F_{ini} is the initial crush force of 1500 kN at all interfaces.

The gradient trend of energy distribution is controlled by the function shown as follows:

$$S_i = S_{\min} + (S_{\max} - S_{\min}) * \left(\frac{L_i}{L}\right)^n, \quad (8)$$

where S_i denotes the scale factor of crush force at the i th interface, and n is the gradient exponent. As shown in Figure 10, the gradient trend of energy absorption is various

with a different value of n . L and L_i is the distance from the interface S_1 and the i th interface to S_7 , respectively. The value of L is set to $1.5 E + 05$ mm.

4.2. Parameters Analysis. It is indicated by equation (8) that the parameters $S_{\max}$, $S_{\min}$, and n are the potential factors involved in train crashworthiness performance, so the parameters study with three factors and five levels was conducted for further investigation of their effects on crashworthiness performance. The maximum of interfacial crush force scale factor $S_{\max}$ is set to range from 1.5 to 3.0, $S_{\min}$ ranges from 0.75 to 1.5, and the gradient exponent n ranges from 0.1 to 10. The sample uniformity is guaranteed by logarithm operation in the sampling of gradient n . All parameters groupings were determined and listed in Table 2. Every set of parameters was adopted for collision analysis using the calibrated 1D train model in Section 3, and the peak value of acceleration and interfacial force of each vehicle was plotted in Figure 11.

As shown in Figure 11, the peak value of vehicle acceleration and interfacial force shows almost the same with the changes in energy gradient parameters. It can be seen from Figures 11(a) and 11(b) that the vehicle acceleration and interfacial force of the vehicle A_1 to A_3 sharply decrease with $S_{\max}$ increase, and this trend alleviates when $S_{\max}$ is close to 2.0. It is evident from Figures 11(c) and 11(d) that both the acceleration and interfacial forces of the vehicle A_1 to A_3 increase as $S_{\min}$ increases, while other vehicle indicators do not vary significantly. From Figures 11(e) and 11(f), it can be found that the indicators change of the vehicle A_1 to A_3 is still remarkable. It should be noted that the trend of indicators for A_1 is obviously diverse from A_2 and A_3 ; much larger exponent n results in decreasing the vehicle acceleration of A_1 and interfacial force F_1 . However,

FIGURE 8: Interfacial forces comparison by calibrated 1D model and 3D model. (a) S1. (b) S2. (c) S3. (d) S4.

FIGURE 9: Functionally graded crash energy distribution.

as shown in Figures 11(e) and 11(f) at the point of $n = 1$, the trend of vehicle acceleration and interfacial force begins to change except for the vehicle A_1 . When n ranges from 0 to 1, the peak acceleration and the interfacial force present a downward trend with the increase of n , while an upward trend within the interval of 1 to 10.

4.3. Surrogate Model. It is illustrated by a previous investigation in Section 4.2 that there exist optimal parameters for the train with functionally graded energy distribution. It is available to establish and solve the optimization problem for parameters determination. An approximate model is herein implemented to establish the relationship between

FIGURE 10: Relationship between normalized distance and normalized scaling factor under different gradient exponent.

TABLE 2: FGE parameters and factorial levels.

No.	$S_{\max}$	$S_{\min}$	n
1	1.3	1	1
2	1.6	1	1
3	1.9	1	1
4	2.2	1	1
5	2.5	1	1
6	1.9	0.8	1
7	1.9	0.9	1
8	1.9	1	1
9	1.9	1.1	1
10	1.9	1.2	1
11	1.9	1	0.2
12	1.9	1	0.5
13	1.9	1	1
14	1.9	1	2
15	1.9	1	5

parameters and crashworthiness indicators by experimental design. The optimal Latin Hypercube technique was conducted to generate 500 sample points in the parameters space with interval 1.5 to 3.0 of $S_{\max}$, 0.75 to 1.5 of $S_{\min}$, and 0.1 to 10 of n .

The Radial Basis Function (RBF) is used to establish the surrogate model as well accuracy on the nonlinear approximation, which is used to describe the functional relation of the peak acceleration of vehicle A_1 – A_6 and interfacial force F_1 – F_6 to the gradient energy configuration parameters in the 1D train model. The RBF model mentioned is a three layers feedforward neural network; it consists of three neuron layers including an input layer with 3 nodes, an output layer with 6 nodes, and a hidden layer with 8 nodes. The Gaussian radial basis functions are used as the transfer functions between the input and hidden layers. The acceleration A_1 – A_6 and interfacial force F_1 – F_6 are individually predicted by the RBF.

The prediction error is the key indicators to determine the feasibility of the surrogate model for further utilization. The error validation was conducted for the approximate model using the following four commonly used metrics,

including R square (R^2), Relative Average Absolute Error (RAAE), and Relative Maximum Absolute Error (RMAE) calculated as in the following equations [28]:

$$\text{RMSE} = \sqrt{\frac{\sum_{i=1}^m (y_i - \hat{y}_i)^2}{m}}, \quad (9)$$

$$R^2 = 1 - \frac{\sum_{i=1}^m (y_i - \hat{y}_i)^2}{\sum_{i=1}^m (y_i - \bar{y})^2}, \quad (10)$$

$$\text{RAAE} = \frac{\sum_{i=1}^m |y_i - \hat{y}_i|}{\sum_{i=1}^m |y_i - \bar{y}|}, \quad (11)$$

$$\text{RMAE} = \frac{\max\{|y_1 - \hat{y}_1|, \dots, |y_m - \hat{y}_m|\}}{\sum_{i=1}^m |y_i - \bar{y}|}, \quad (12)$$

where m is the number of error validation points; y_i is the exact value by FEA; $\hat{y}_i$ is the value predicted by surrogate model; $\bar{y}$ is the mean value from FEA.

In general, higher R^2 and lower RAAE and RMAE indicate better accuracy of the surrogate model. The correlation coefficients are calculated by equations (9)–(12) and summarized in Table 3. Figure 12 gives a straight comparison of prediction accuracy on acceleration and interfacial force. It is obvious that the RBF model has higher prediction accuracy for the interfacial force more than acceleration. The R^2 of interfacial force generally exceeds 0.9 but for that of F_6 at the last interface, which is largely due to the interfacial force at the interface S_6 tends to stay at a steady level as incomplete energy absorption in the current collision condition. In general, the unreasonable energy configuration of the train always leads to poor crashworthiness performance during a collision. For instance, the sharply increased impact force at the vehicle interface tends to generate a high level acceleration. So it is essentially the same purpose of improving crashworthiness by decreasing either the interfacial force or acceleration. As shown in Figure 13, it can be easily found that the predicted data of

FIGURE 11: Influence of parameters on peak crash force and acceleration. (a) Acceleration— $S_{\max}$. (b) Acceleration— $S_{\min}$. (c) Acceleration— $\log(n)$. (d) Force— $S_{\max}$. (e) Force— $S_{\min}$. (f) Force— $\log(n)$.

TABLE 3: Prediction accuracy criteria of the surrogate model.

	RSME	R^2	RAAE	RMAE
A_1	0.07676	0.89427	0.04384	0.35779
A_2	0.0929	0.87506	0.06388	0.34854
A_3	0.09257	0.86847	0.06738	0.31286
A_4	0.10964	0.84381	0.07869	0.48723
A_5	0.14451	0.7878	0.11366	0.44218
A_6	0.11715	0.75014	0.07769	0.45686
F_1	0.07559	0.94785	0.04143	0.39939
F_2	0.0672	0.95057	0.04726	0.3013
F_3	0.07876	0.93542	0.0513	0.25993
F_4	0.07216	0.94311	0.05133	0.23806
F_5	0.04035	0.97145	0.02033	0.22219
F_6	0.08886	0.86227	0.05324	0.28304

acceleration distribute relatively disperse, while the interfacial force is more concentrated, so the interfacial force seems to be more suitable in selection as the optimization target in the subsequent optimization solution.

4.4. Optimization Design. From the analysis results in Section 4.2, the interfacial force can be reduced by determining the optimal group of parameters including $S_{\max}$, $S_{\min}$, and exponent n .

The mathematical description of the optimization problem on energy configuration parameters for train collision can be expressed as equation (13), where the interfacial force at interfaces S_1 to S_3 , minimizing F_1 to F_3 , is set as the objectives function of the multiobjective optimization problem. Crush force factors $S_{\max}$, $S_{\min}$, and gradient exponent n are set as the design variable, and their variation range is kept consistent with the value in parameters analysis.

$$\begin{aligned}
 &\min F_1(S_{\max}, S_{\min}, n), F_2(S_{\max}, S_{\min}, n) \\
 &\quad F_3(S_{\max}, S_{\min}, n) \\
 &\text{s.t. } 1.5 \leq S_{\max} \leq 3.0 \\
 &\quad 0.75 \leq S_{\min} \leq 1.5 \\
 &\quad 0.1 \leq n \leq 10.
 \end{aligned} \tag{13}$$

The multiobjective optimization problem mentioned above is solved using the NSGA-II algorithm, where the parameters adopted are listed in Table 4.

The Pareto frontier of F_1 , F_2 , and F_3 is obtained and shown in Figure 14. From the projection view of optimal points, it can be illustrated that there exists a nondominant relationship between the peak interfacial force at interface S_1

FIGURE 12: Prediction error comparison of peak acceleration and interfacial force. (a) RSME. (b) R^2 . (c) RAAE. (d) RMAE.

and that of interfaces S_2 and S_3 , respectively, while the peak interfacial force at interface S_2 is dominant to that of interface S_3 . In other words, the peak value of interfacial force at interfaces S_2 and S_3 can be reduced to an optimal minimum, but for that between interfaces S_1 and S_2 or S_3 .

5. Validation of Optimal Design

A set of optimization results are obtained, and the optimal value of F_1 to F_3 is 3876 kN, 4376 kN, and 4979 kN, respectively. The corresponding design variable is $S_{\max} = 1.950$, $S_{\min} = 0.995$, $n = 1.546$. The stable crush force of the energy absorber can be calculated via equation (8). The results at interfaces S_1 to S_4 are 2925 kN, 2258 kN, 1754 kN, and 1492 kN, respectively. The optimal values of interfacial platform forces derived from this work also validated the conclusion by theoretical investigation in literature [29] that the compressive strength of the front energy absorber should

be higher than its adjacent rear energy absorber to the benefit of effective energy absorption for the distributed energy absorption system of the train.

In order to investigate the improvement of crashworthiness when introducing the FGE mode with optimal parameters, corresponding energy distribution parameters for uniform energy (UE) absorbing mode were calculated under the rule of equal total energy configuration. The parameters detail of energy configuration for two kind modes of FGE and UE are shown in Table 5, in which data of interface S_7 is not listed as its no participation in energy absorption. It can be found that the energy absorption at interface S_1 is 1707 kJ under the case of UE mode, whereas it increases to 2048 kJ under the FGE mode and the energy absorbing efficiency rises from 71% to 98.1%. What's more, for the UE mode, all the designed energy at interfaces S_2 to S_4 is absorbed completely and that of interface S_5 is absorbed partially, and the energy absorption component of interface S_6 does not

FIGURE 13: Comparison of predicted and actual peak acceleration and force.

TABLE 4: Parameters adopted for NSGA-II.

Parameters	Value
Population size	60
Number of generation	100
Crossover probability	0.9
Crossover distribution index	10
Mutation distribution index	20

TABLE 5: Parameters comparison of different energy configurations.

	Mode	S ₁	S ₂	S ₃	S ₄	S ₅	S ₆
Platform load (kN)	UE	3429	1500	1500	1500	1500	1500
	FGE	2925	2258	1754	1492	1754	2258
Designed energy (kJ)	UE	2400	525	525	525	525	525
	FGE	2048	790	614	522	614	790
Absorbed energy (kJ)	UE	1707	507	511	520	191	3.1
	FGE	2008	783	606	452	1.8	0.9
PC (%)	UE	71.1	96.6	97.3	99.0	36.4	0.6
	FGE	98.1	99.1	98.7	86.6	0.3	0.1

Note: PC = (Absorbed energy / designed energy) × 100%.

FIGURE 14: Pareto frontier of interfacial force with optimal energy distribution parameters.

work. When it comes to FGE mode, the energy at interface S₂ and S₃ is absorbed completely, that of interface S₄ is partially absorbed, and there exists remaining stroke; almost no energy absorption is absorbed in the interfaces S₅ and S₆.

Figure 15 gives an intuitive understanding of the difference of two energy modes by comparing the energy absorption of each interface. The energy absorption at the first and second interfaces under the FGE mode is significantly higher than that of UE mode.

In order to have a further study on the influence of the gradient energy mode on the interfacial force during the process of a train collision, the interfacial forces under two kinds of energy modes were extracted and compared. As shown in Figure 16, as a result of incomplete energy absorption at the first interface S₁, the peak impact force for UE and FGE has been at a stable level of crushing force. The

FIGURE 15: Comparison of interfacial energy absorption under two configuration modes.

interfacial force at S_1 appears multiple times of loading and unloading, while the force curve under FGE mode is relatively smooth and continuous at the interval of 0–400 ms. This difference demonstrates that the energy absorption process is more steady under FGE mode than UE mode. As full energy absorption at interface S_2 , the impacting force appears obvious peak value. From Figure 16(b), under UE mode, the energy absorption reaches the full stroke at 139 ms and results in the peak impact force of 7165 kN at 180 ms. When it comes to UGE mode, the corresponding peak force of 4769 kN appears at 431 ms with a completely full stroke at 330 ms. Compared to UE mode, the peak value of interfacial force at S_2 is remarkably decreased by 33.4%, and its appearance time is delayed by 251 ms for the FGE mode; the increase of time consumed in energy absorption is beneficial to the crashworthiness of the train.

Similar to the situation of interface S_2 , the impact force at the interface S_3 shown in Figure 16(c) is also significantly different for two kinds of energy absorption modes. For the UE mode, the energy absorber reaches its full stroke at 274 ms, and then the peak value of 6467 kN appears at 312 ms. However, the moment of full stroke and peak impacting force under FGE mode is 379 ms and 418 ms, which is obviously later than UE mode by 105 ms and 106 ms, respectively. The peak value of the interfacial force also decreases to 4447 kN by 31.2%. In addition, it should be noted that the peak value of interfacial forces S_2 and S_3 appears in sequence at 180 ms and 312 ms under UE mode; the corresponding phenomenon is not observed under FGE mode; the moment of peak force at interface S_2 and S_3 is 431 ms and 418 ms, respectively. Even the moment related to the peak impacting force of interface S_3 is earlier than that of S_2 , which is difficult to be observed under UE mode. The coincidence of moment indicates that the FGE mode is conducive to improve the simultaneity of energy absorption at adjacent interfaces when a train collision occurs.

In summary, the mechanism of energy absorption under FGE mode is remarkably different from that under UE mode. Figure 17 presents a more intuitional perception of the difference of the energy absorption process under UE and FGE mode by outputting their historical energy absorption curves of each interface. As presented in Figure 17(a), the energy absorption under UE mode behaves as a strong character of gradual sequence with the propagation of the shock wave generated by train collision. For instance, after the energy absorption from interface S_1 reaches a certain level, the interface S_2 begins to absorb energy at 36 ms and finishes at 153 ms, the moment of full stroke completed. This phenomenon is also observed at other interfaces, such as the beginning moment for energy absorption of S_2 and S_3 is 166 ms and 302 ms, while the end moment is 287 ms and 434 ms, respectively. The energy configuration at interface S_5 is partially absorbed. It is found that the energy absorbing curve at interface S_1 shows three states of platform phenomena with energy absorption of 851 kJ, 1283 kJ, and 1548 kJ, respectively. The energy absorption at interface S_1 tends to stop when the energy absorber at rear interfaces comes into subsequent action, and the duration of each platform is approximately the same as the energy absorption interval of interfaces S_2 , S_3 , and S_4 . However, it is demonstrated from Figure 17(b) that the energy absorption process is more stable for interface S_1 , the platform phenomenon of energy absorbing process at the first interface is weakened under FGE mode, the corresponding energy level of the three platforms is 1772 kJ, 1839 kJ, and 1983 kJ. What's more, the energy absorption of interface S_2 to S_4 begins at 34 ms, 62 ms, and 93 ms and finishes at 343 ms, 393 ms, and 420 ms, respectively. The start time interval of adjacent interfaces between S_2 , S_3 , and S_4 decreases approximately from 130 ms of UE mode to 30 ms. It can be illustrated that the mechanism improving the

FIGURE 16: Collision force comparison under uniform and graded energy configurations (a) S_1 . (b) S_2 . (c) S_3 . (d) S_4 . (e) S_5 . (f) S_6 .

FIGURE 17: History of interfacial energy absorption under different energy configurations (a) UE (b) FGE.

crashworthiness of FGE mode is promoting the synchronization of the energy absorption at all interfaces.

6. Conclusion

In this paper, a simplified 1D model based on a nonlinear bar element was proposed to characterize the longitudinal behavior of the carbody in crashworthiness analysis for a train, and equivalent parameters of the bar element were calibrated by optimization method. The simplified model of the carbody was extended to the application of the 1D model of train collision, and its feasibility was validated by analytical results of the corresponding 3D model. In addition, a gradient energy absorption design concept is proposed to promote the crashworthiness of the train, and the effect of gradient parameters on crash response was investigated, and the optimal design parameters are solved on the basis of a simplified 1D train model. Some valuable conclusions are obtained for reference:

- (1) It is feasible to characterize the response of the carbody under longitudinal impact load by the nonlinear 1D element, but it is imperative to calibrate its equivalent parameters to guarantee the analyzing accuracy. On the indicators selection for parameters calibration of a simplified 1D model, consideration of each aspect including the internal response of the carbody itself and the external response of the impactor is helpful to obtain better equivalent parameters. The simplified model with calibrated parameters can be used for the assessment of CEM and optimization design of train crashworthiness.
- (2) The crash energy configuration with the gradient design concept can significantly improve the crashworthiness performance of a train. The energy absorption at each interface of the train presents

good simultaneity under FGE mode, which also eliminates the inefficient energy absorption due to the delay phenomenon of energy absorption at adjacent interfaces under UE mode. Under the case of equivalent energy capacity, the gradient energy mode can significantly reduce the maximum impacting force at the collision interface and lengthen the time of duration for energy absorption at each interface, and the appearance of the peak force is delayed. All of this is beneficial to enhance energy absorption efficiency and improve the crashworthiness, so the FGE mode is a potential and practical design concept of crash energy configuration.

Data Availability

Some or all data, models, or code generated or used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare no conflicts of interest.

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