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An FSRW numerical simplification approach for vehicle frontal crashworthiness analysis

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Abstract： Vehicle frontal crashworthiness analysis is an important topic in the field automotive community, as it relates to legislative requirements. Frontal crash models contain a large number of elements and therefore present a high computational cost, especially when performing crashworthiness structural performance optimisations. A new numerical methodology is proposed in this paper with the aim to increase computation speed by implementing a sub-modelling approach on the frontal structure-rear wheels (FSRW) method. In this new method, the vehicle body structure behind the rear seats is replaced by a point mass, with an equivalent mass to the rear structure removed, and attached to the rear frame of the front structure and its wheels. This new method was rigorously tested against validated full size vehicle computer models of different classes provided by NHTSA, and included MPV, SUV, van, and sedan, against fixed rigid barrier (FRB), small overlap and a mobile progressive deformable barrier (MPDB) tests. The research has demonstrated that this new sub-modelling approach correlated against all the full size NHTSA computer models in deformations, intrusions, velocities, and accelerations, as well as providing a runtime reduction between 7% and 23%. This new simplified method, which can be easily implemented, is innovative and will have an important impact vehicle design, as it will allow an easier use of optimisation techniques, which will lead to safer vehicles.

Key words： Crashworthiness analysis; Optimisation; Numerical methodology; Sub-modeling; Simplification

1. Introduction

Frontal impacts of vehicles due to the high frequency and severe occupant injury have been examined for many years [1-3]. Front Rigid Barrier (FRB) impact, 25% overlap (small overlap) impact, and Mobile Progressive Deformable Barrier (MPDB) impact evaluation rating protocols have been built up [4-5]. Crashworthiness analysis has, therefore, been one of the most critical research fields for the automotive community. Besides, simulation technology-supported significantly reduces the number of tests and saves time in developing new vehicle models [6]. Different materials, such as magnesium alloys, high strength steels, and hot stamping steel, have been used to enhance the vehicle crashworthiness [47-48]. Engine bay structures, including cashbox, shotgun were optimized to improve the vehicle performance in the impact [49-50].

Moreover, a series of multi-surrogate-assisted optimization was proposed to optimize vehicles [51-52]. However, establishing a multi-surrogate-assisted optimization model may require hundreds of simulations [53-54]. Furthermore, each vehicle analysis model can take up to dozens of hours or even weeks, making it a severe challenge for analyzing the vehicle's crashworthiness [7-8]. Owing to this, many researchers have done lots of work on studying vehicle crashworthiness simplification technology.

Over the years, researchers and the automotive community have proposed a series of practical simplification approaches to reduce computational time. For instance, Ni and Lin [9] evolved a lumped-mass-nonlinear-resistance system consisting of mass and spring characteristics from hybrid impact simulation technique and dynamic elastic-plastic analysis. In their research, the numerical results were in good agreement with experimental results. Thus, the simplification method was a practical way to improve the crash performance of the whole vehicle model. Chowdhury and Taguchi [10] shown the manner of selecting mass and spring parameters. Wood [11] proposed and verified a four springs simplification method to model the mutual crushing of the front and side structure using two uppers and two lower springs. The rollover velocity variation with the motorcycle's mass and height of the seat could be calculated using this four springs simplification approach. Finally, Gandhi and HU [12] investigated an analytical model consisting of a mass, spring, and damper. The verification results indicated that this approach could help develop a valuable simplified model.

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After that, researchers investigated and verified a series of more complex lumped-mass-spring-damper simplification methods [7,13-17]. The simplified vehicle models in which the structural regions were replaced by stiffness and load paths were employed in a frontal impact in their research. Moreover, 92% computational cost reduced of crash analysis could be obtained [7]. On the other hand, Ren [18] applied the equivalent static load method for structure optimization (ESLSO) in the vehicle simplified model. In the ESLSO method, the vehicle body structure was simplified into two sections of thin-walled structure with different thickness distributions. The maximum internal energy and the time of the ESLSO simplified collision model and full vehicle collision model are the same. Thus, those simplification approaches with the spring system and the ESLSO method could be a valuable approach for the early phase of vehicular crashworthiness design, which stage non-availability of date of component [17].

Nevertheless, certainties' effect could not be verified through the spring simplification model and ESLSO model. Moreover, the spring simplification model and ESLSO model need to complete a complex mathematical calculation to obtain the spring damping parameters, limiting the potential of the two simplified methods to be applied to engineering. Therefore, many researchers investigated simplified modeling for automobile frontal impact sleds. Upper vehicle body structures with minor deformation were replaced by using a mass point to overcome this limitation. For instance, Qiu et al. [19] proposed a simplified model with the frame, bumper, engine transmission, steering, water tank, and simplified instrument panel. In their study, 30° bent of the longitudinal beam was obtained, while a 24° bent as shown in the simplified model. Wang et al. [20] investigated frame dynamic slide tests as a simplified model. Different frame structure optimal designs were proposed through numerical analysis with the simplified model. Furthermore, the peak deceleration of the frame is reduced by 30.8%.

Besides, Dong et al. [21] presented a new sled impact with engine and drive system, suspension system, wheel, underbody to substitute whole car impact test. The varified results indicated good consistency between the front longitudinal beam deformation and body acceleration curve of the simplified sled model and complete car crash. Moreover, the number of elements, nodes, and solder joints has been reduced by nearly half of the whole vehicle model, reducing the computational cost. Finally, Chen et al. [22] suggested a similar but more complex simplified sled method. A mass point was adopted to replace the sidewall in their study, and a rigid connect way was used to simulate. The simplified minibus model showed a 53% computing time reduction with a 5% error of main energy-absorbing parts than the entire vehicle model.

A slight disadvantage of the simplified sled numerical method is the lack of representation of the deformation of the hinge-pillar and rear vehicle structures. Under severe frontal impact conditions, such as small overlap impact, the crash forces concentrate in the front wheel, suspension, firewall, and A-pillar, resulting in severe passenger compartment intrusions [23-25]. Thus, the Altair company presented a sub-structure modeling approach to simplify whole vehicle models [26]. All entities inside the boundary and the section force were used to build up the sub-model in the sub-structure model. Stein et al. [27] presented and verified five different sub-structure models. The results showed that 10 hours of computation cost reduction could be observed when the structures behind A-pillar were replaced by mass, and 12.5 hours of computation cost reduction could be saved when the engine bay structure and passenger compartment were replaced by mass.

Moreover, Yao et al. [28] selected the frontal structure, extremely the engine bay structure, to build up a simplified vehicle frontal impact model. In their research, the pick value of the collision acceleration was reduced by 11g by optimizing structure features of the frontal structure through the simplified model. This simplified model retains the engine bay and hinge pillar. Thus, the frontal structure deformation can be directly observed and facilitates the optimization of the crashworthiness of the car body. Many researchers employ this simplified approach to optimizing bus, sedan, and sport utility vehicles [29-31]. However, most of them suggested a simplified sled method applicable to a single vehicle, rather than a method suitable to MPV, SUV, van, and sedan. Besides, a disadvantage with this simplified approach is how a change in the A-pillar and front door affects the passenger compartment due to a lack of available completeness structures. On the other hand, the improvement of the A-pillar and front door structure is an important measure to optimize the crashworthiness of the car body [32-33].

According to the conclusions from the previous study above, the umped-mass-spring-damper simplification method and equivalent static loads method for structure optimization take less computation time than the sled simplification approach. However, the sled simplification simulation model could present the deformation of the front rail, which is conducive to improve vehicle crashworthiness performance. Besides, a more complex simplification model containing engine bay structure and hinge-pillar could provide more information about passenger compartment deformation. Therefore, those sub-modeling simplification approaches may present the effect of optimal hinge-pillar structure on the crashworthiness performance. Furthermore, more options could be provided for vehicle crashworthiness design using a simplification model with engine bay structure, A-pillar, front door, and wheels because A-pillar and front door are vital in a severe frontal collision. Moreover, this method can be applied to different models such as MPV, SUV, van, and sedan at the same time. However, to our best knowledge, the method of simplification based on frontal structure-rear wheels (FSRW) sub-modeling has not been widely investigated and reported.

This paper presents a new FSRW sub-modeling as a simplified numerical approach for vehicle frontal crashworthiness analysis of different vehicles. The FSRW simplified approach does not require complicated mathematical calculations, making it an extremely high engineering application value. Unlike most of the traditional sub-structure, a mass point was used to replace the vehicle structure behind the rear seats. In addition, the rigid element was employed to connect the vehicle's frontal structures and the rear wheels. MPV, SUV, van, and sedan simplification models in FRB impact, small overlap impact, and MPDB impact were built up to verify the simplification numerical method. This approach may be an extremely excellent numerical simplification method in practical engineering.

2 Methods and Materials

Vehicle Finite element (FE) model of the Tauras, Explorer, Silverado, Accord, Econoline, and Caravan from the National Crash Analysis Center (NCAC) library were employed to build the frontal impact model in both complete vehicle and FSRW sub-modeling in this paper. Those whole vehicle FE models are based on the actual vehicles' structural definitions and were verified by comparing the crashworthiness performance between the FE models and crash tests [34-35]. The validations were proposed for FRB impact, small overlap impact, and MPDB impact to exam the simplified approach accuracy.

2.1 Frontal impacts

Road traffic crashes have caused millions of people to suffer casualties every year [36]. Furthermore, the frontal impact was the most common first harmful event of all the traffic crash types [37]. For this reason, vehicle assessment programs were established to promote automobile companies to improve the crashworthiness of automobile structures. Among them, the widely used collision forms include FRB impact, small overlap impact, and MPDB impact.

2.1.1 FRB impact

Fig.1 (a) shows that the vehicle hits the rigid wall at 56.3 km/h with 100% overlap in the FRB impact. The engine bay structures, firewalls could be potential targets for crashworthiness optimization. Besides, the damage suffered by the occupants in a collision accident is the focuses of the new car assessment program in various countries [38]. Thus, vehicle' deformation of the vehicle, firewall' intrusion, the velocity and acceleration of engine bottom and B-pillar are calculated as indicators for the consistency between the simplified and full-vehicle models [8], as shown in Fig. 1 (b).

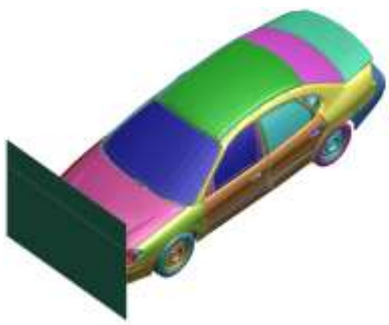


Fig.1. FRB impact simulation model.

2.1.2 small overlap impact

As shown in Fig. 2, the vehicle impacts the rigid barrier at 64 km/h with 25% overlap in the small overlap impact. Moreover, tens of intrusion measure points, including the upper hinge-pillar and steering column points, are selected as the indicators [39]. Unfortunately, there are no steering column and dashboard in the Caravan and Explore FE model proposed by NCAC. Thus, vehicle' deformation of the vehicle, firewall' intrusion, the velocity and acceleration of engine bottom, and B-pillar mentioned above were adopted as the consistency indicators.

Fig.2. FRB impact simulation model

2.1.3 MPDB impact

In the MPDB impact, the vehicle FE model and MPDB barrier built up by LS-DYNA company collide with each other at a speed of 50 km/h with 50% overlap, as shown in Fig. 3 [40]. The deformation, firewall' intrusion, velocity, and acceleration of engine bottom and B-pillar are compared to verify the consistency of the vehicle models herein.

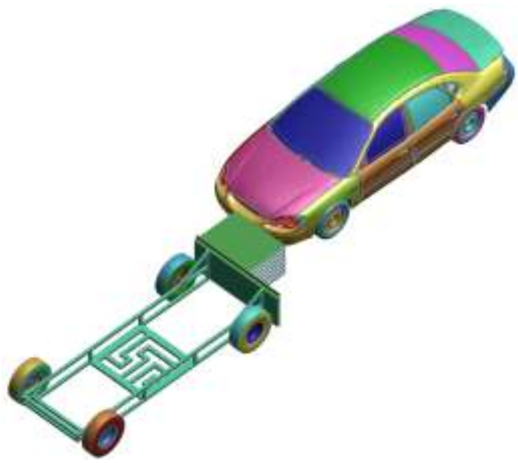


Fig.3. MPDB impact simulation model

2.2 FSRW sub-modeling simplified model

Dozens or hundreds of simulations are often required to optimize the crashworthiness of the vehicle. Furthermore, due to many detailed vehicle collision model elements, a significant amount of time is taken to utilized detailed vehicle models for simulation. Therefore, a three-step approach to building the FSRW sub-modeling simplified model could be applied to analyze the optimization scheme quickly. The three-step was described in Fig. 4. In the three steps, a mass point placed in the center of gravity of the removed structures was adopted to replace the mass of the removed structures. The car's rear wheels are retained to maintain the consistency of slip and rotation between the simplified and full vehicle models. Moreover, the frontal structure, rear wheels, and the mass point were connected by rigid elements.

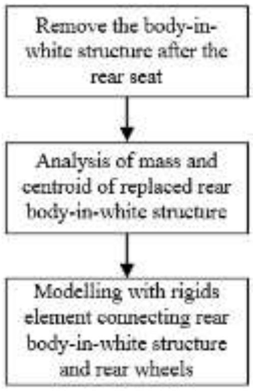


Fig.4. Three-step method for FSRW sub-modeling simplified model.

2.2.1 Remove the body-in-white structure

Fig. 5 shows a one-dimensional model of FRB impact, the two-dimensional model of small overlap impact, and MPDB impact. The collision force is far greater than the non-collision force, such as the friction and wind resistance of the ground against the tires in the frontal impacts. Therefore, it can be considered that the collision system is not affected by external forces [41]. Thus, the energy loss of the collision system caused by the body's plastic deformation is equal to the difference between the kinetic energy before and after the collision. Furthermore, the energy loss ΔE_{FRB} in the FRB impact can be defined as:

$$\Delta E_{FRB} = \frac{1}{2}m(v_0^2 - v_1^2) = \frac{1}{2}ks^2 \quad (1)$$

Where m is the vehicle's mass, the V_0 is the initial velocity of the car, V_1 is the speed after the collision, k is the plastic elastic coefficient of the front part of the car body, and the s is the deformation of automobile structure.

Besides, the force and reaction force is generated along the normal direction and the tangent direction of the contact surface in small overlap impact and MPDB impact, which will cause the car to rotate or slide, as shown in Fig.5 (b) and (c). Suppose further that the center of gravity and moment of inertia of the car remain unchanged during the collision. The synthetic impulse acts on the center of the collision at the end of the collision. Hence, the energy loss ΔE_{small} in the small overlap impact can be adopted as:

$$\Delta E_{small} = \frac{1}{2}m(v_0^2 - v_1^2) - \frac{1}{2}I\omega^2 = \frac{1}{2}ks^2 \tag{2}$$

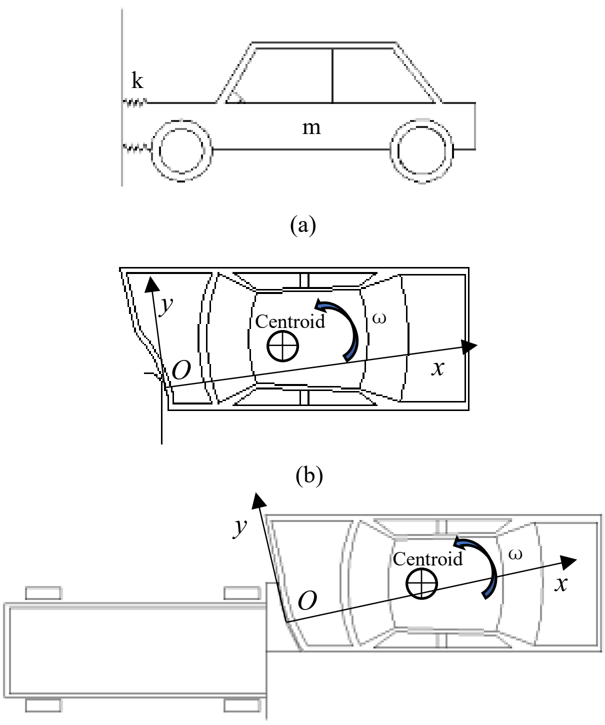
Where I is the moment of inertia, the ω is angular velocity. the energy loss ΔE_{MPDB} in the MPDB impact can be adopted as:

$$\Delta E_{MPDB} = \frac{1}{2}m_1(v_2^2 - v_3^2) + \frac{1}{2}m(v_0^2 - v_1^2) - \frac{1}{2}(I_1{\omega_1}^2 + I\omega^2)$$

$$= \frac{1}{2}(k_1{s_1}^2 + ks^2)$$

(3)

Where m_1 is the mass of the barrier, the v_2 is the initial velocity of the barrier, the v_3 is the speed of the barrier after the collision, I_1 is the moment of inertia of the barrier, the ω_1 is the angular velocity of the barrier after impact, k_1 is the plastic elastic coefficient of the front part of the barrier, and the s_1 is the deformation of the barrier structure.



(c)

Fig. 5. One-dimensional model of FRB impact (a), two-dimensional model of small overlap impact (b), and MPDB impact (c).

Thus, the structure of the passenger compartment determines the deformation of the passenger compartment in a collision. The lower the structural strength of the frontal structure of the passenger compartment, the greater the deformation of the car body. Besides, mass points are generally resorted to replacing the rear structure of the passenger compartment with minimal deformation when establishing a simplified sub-modeling model [26]. In a traditional simplified scheme, a mass point replaced the centroid of the vehicle structure behind the B-pillar, as shown in Fig. 6.

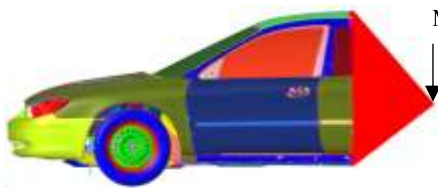


Fig. 6. Schematic diagram of typical simplified schemes

However, those crash forces may cause serious deformation of the passenger compartment arch with a severe deformation near the B-pillar when the crash forces transmitted to the rear structure, as shown in Fig. 7. Furthermore, when the passenger compartment is severely bent and deformed, the simplified model could be different from the whole model's falling without rear wheels as support.

(a) (b)

Fig. 7. Side (a) and bottom (b) view for the small overlap impact simulation model

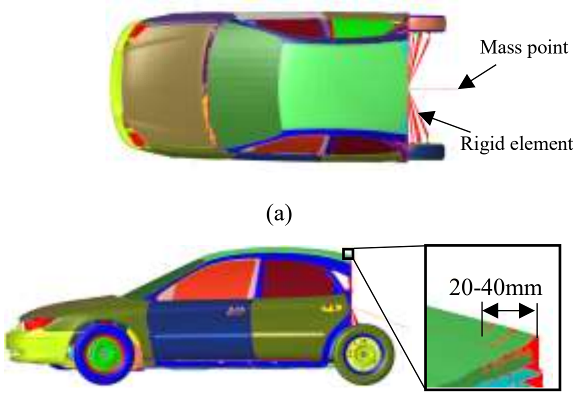
Moreover, it seems that there is no way to judge the deformation of the vehicle body and establish a simplified model in the absence of vehicle body structure. Fortunately, we have found in engineering that the structure behind the rear seats rarely deforms under the frontal impact conditions because the beams at the rear seat mounting points on body-in-white enhance the rigidity of the nearby structure. For this reason, the structure after the rear seat could be replaced by a mass point in a frontal structure-rear wheels (FSRW) sub-modeling.

2.2.2 Analysis of mass and centroid of replaced structures

Ensuring the same mass distribution is essential for ensuring consistency between the simplified and full models [7]. Thus, the mass and center of gravity of the removed structures, which could be obtained by computer-assisted engineering software, such as Hypermesh and Oasys primer, have to be determined. A mass point is adopted at the centroid of the structures to replace those structures. Hence, the simplified model has the same mass distribution as the whole vehicle model.

2.2.3 Modelling with rigid element connecting rear body-in-white structure and rear wheels

The frontal structures, rear-wheel, and mass point determined above connect by rigid elements in the FSRW sub-modeling simplification numerical approach. When connecting these parameters, the grid in the 20-40mm width range on the body-in-white, the rim or lower arm mounting holes of the rear wheels, and the mass point are connected by the rigid element. Fig. 8 shows an FSRW sub-modeling proposed through the accord FE-model.



(b)

Fig.8. Top (a) and side (b) views of a schematic diagram of FSRW sub-modeling simplified schemes

3 Results and discussion

this article aims to establish a simple frontal impact simplification method, which avoids indulging in mathematical deduction and forgetting the engineering application background and does not lose the core value of the subject by stating technical details. In order to verify the FSRW sub-modeling numerical simplification approach, Accord, Explorer, Silverado, Tauras, Econdine, Caravan were adopted as MPV, SUV, van, and sedan to propose FRB impact, small overlap impact, and MPDB impact. The verification was carried out by comparing acceleration and front door deformation between the simplified and full vehicle models under different front impacts.

3.1 Verification of the FSRW sub-modeling in FRB impact

Fig. 9 to 12 show the deformation pattern of the FSRW sub-modeling simplified and the whole vehicle. The results show that the simplified models of these four vehicles have the same deformation trends as the whole vehicle model in FRB collisions. Especially in the front structure of the car body, including the front fender, front longitudinal, sub-frame, the front wheel, and the A-pillar, the simplified model and the whole vehicle model have almost the same bending and folding. Moreover, the simplified model and the complete vehicle model have reasonable consistency in the intrusion of each part of the firewall. Previous studies have shown that the bumper beam, the upper beam of the water tank, the shotgun, the front longitudinal, hinge-pillar, and A-pillar bear most of the collision force in FRB impact [42]. Thus, the simplified model can replace the whole vehicle model when analyzing the force and deformation of the body.

Moreover, the vehicles' velocity and acceleration on the engine's bottom and the left lower B-pillar were obtained through the sensors installed on those vehicle finite element models. The velocity of the simplified model and the whole vehicle model is shown in Fig. 9-12. Except for the slight difference in velocity between the simplified models and whole models of the Explorer and Silverado, the velocity of the simplified models and the vehicle model of the remaining models have a remarkable consistency. Among all the models, the Silverado simplified, and the full models have the most significant difference in engine bottom and B-pillar acceleration, reaching 9.1% and 16.7%. The average errors of acceleration at the engine bottom and B-pillar of these four cars are 3.0% and 12.3%, respectively. There is no seat belt retractor at the lower B-pillar of the model, and the lower end of the B-pillar was deformed and shaken during the collision so that the acceleration error of the B-pillar was greater than 10%. The vehicle's remarkable consistency and acceleration indicate that those vehicles' deformation amount and deformation speed in the collision process has a high consistency.

Besides, the energy plot indicated that the simplified model and the vehicle model have high consistency in total energy, kinetic and internal energy. The most significant difference in total energy and internal energy occurs in the Silverado model, but only 1.5% and 2.6%, respectively. The average error of the simplified model's total energy and internal energy and the vehicle model is 0.4% and 0.8%, respectively. Therefore, the FSRW simplified model could effectively represent the full model's crash behavior during FRB impact.

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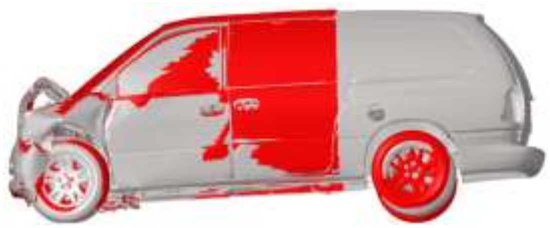
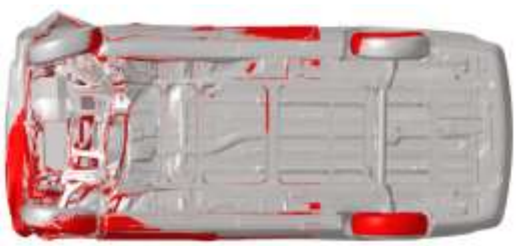
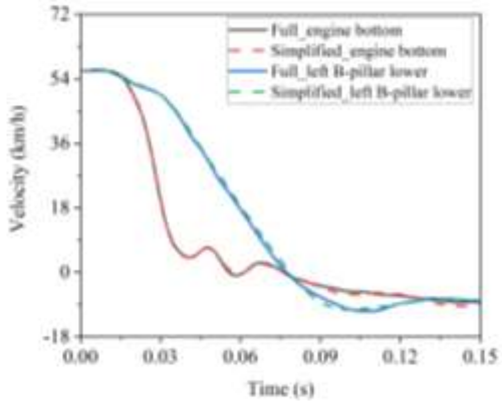


(a) Side view of vehicle deformation

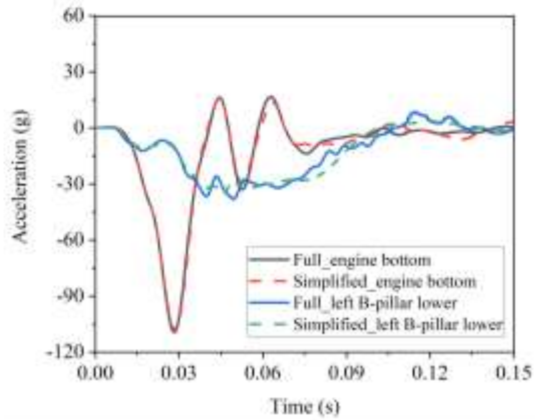


(b) Bottom view of vehicle deformation

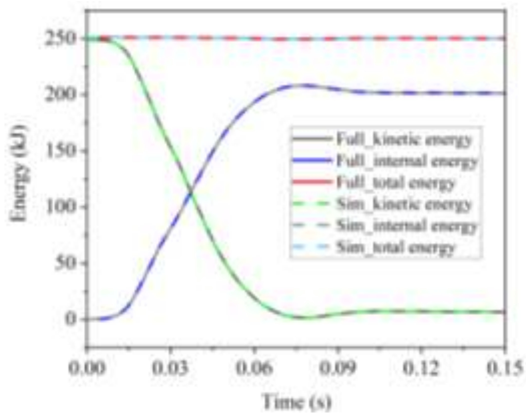
(c) Deformation of the firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar

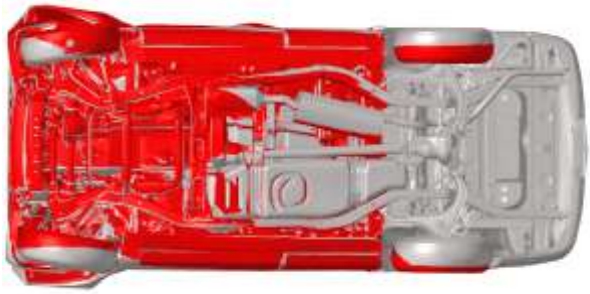


(f) Total, kinetic and internal energy

FSRW simplified model Full model

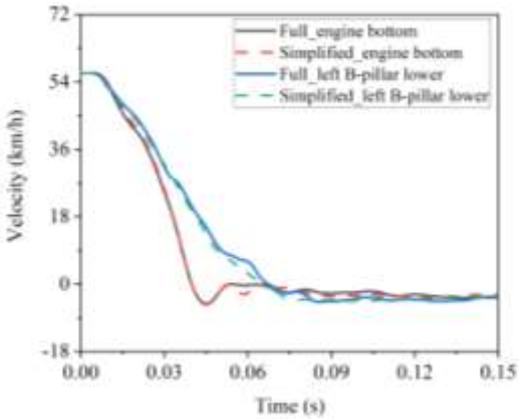
Fig. 9. Comparison FSRW simplified model and full model of Caravan at FRB

(a) Side view of vehicle deformation

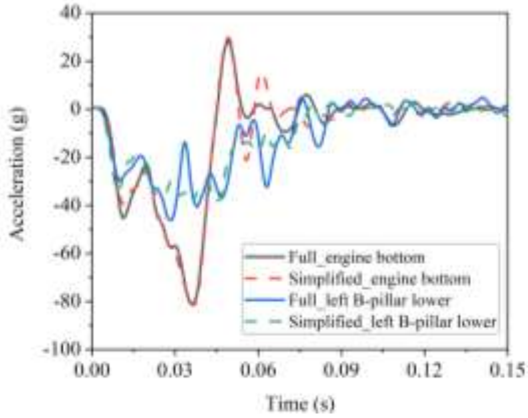


(b) Bottom view of vehicle deformation

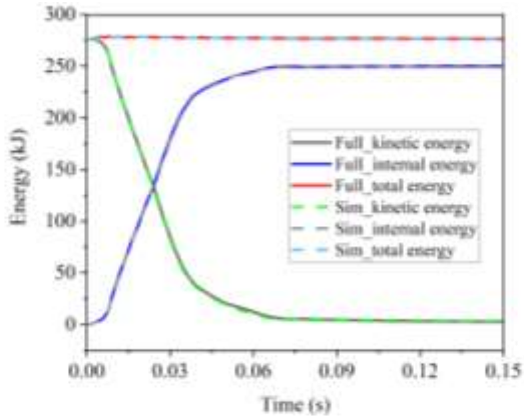
(c) Deformation of the firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar

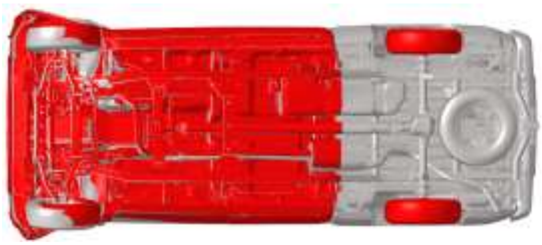


(f) Total, kinetic and internal energy

FSRW simplified model Full model

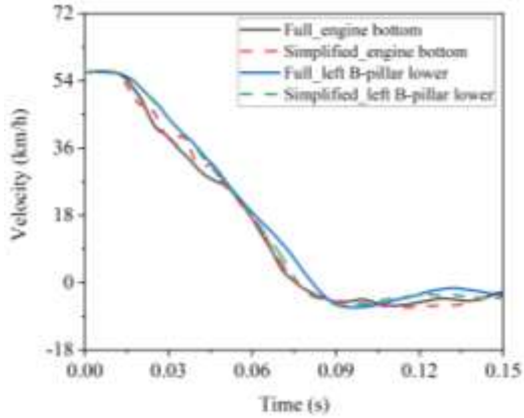
Fig. 10. Comparison FSRW simplified model and full model of Explorer at FRB

(a) Side view of vehicle deformation

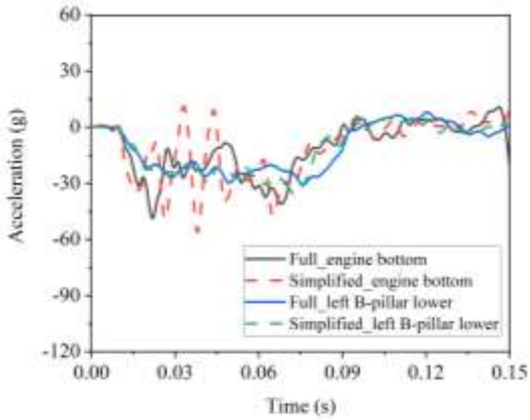


(b) Bottom view of vehicle deformation

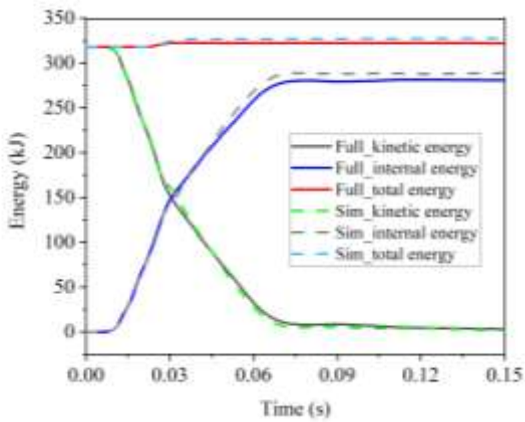
(c) Deformation of the firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar

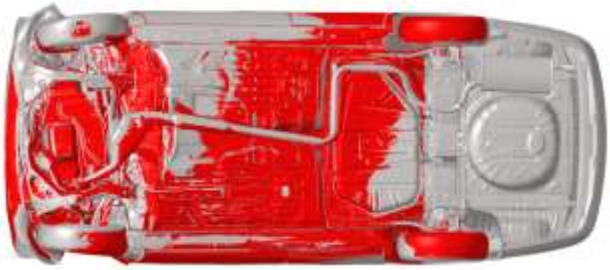


(f) Total, kinetic and internal energy

FSRW simplified model   Full model

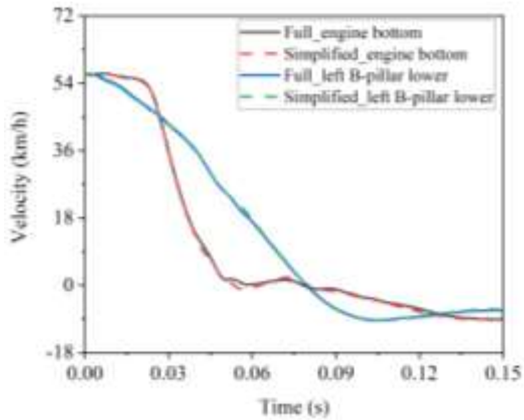
Fig. 11. Comparison FSRW simplified model and full model of Silverado at FRB

(a) Side view of vehicle deformation

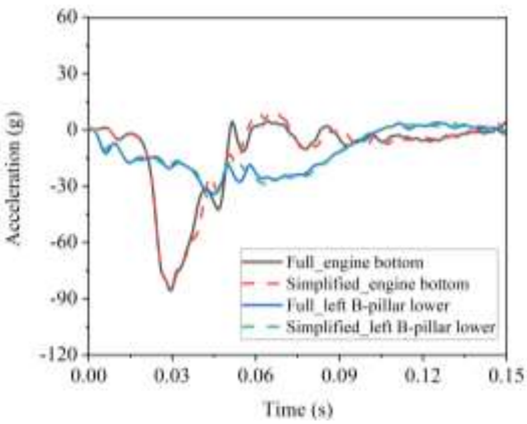


(b) Bottom view of vehicle deformation

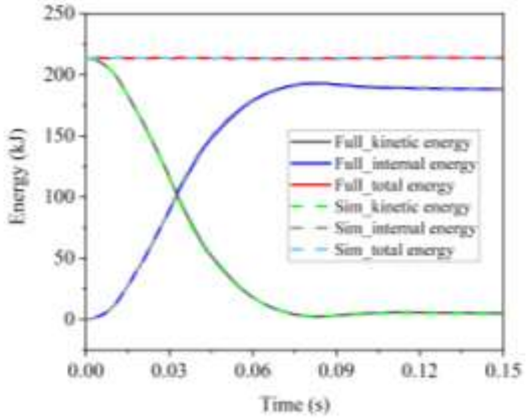
(c) Deformation of firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar



(f) Total, kinetic and internal energy

FSRW simplified model   Full model

Fig. 12. Comparison FSRW simplified model and full model of Taurus at FRB

3.2 Verification of the FSRW sub-modeling in small overlap impact

The deformation pattern between the FSRW sub-modeling simplified and the whole vehicle was graphed in Fig. 13-16. It can be seen from Fig. 10-16 that the simplified model and the whole vehicle model have a remarkable consistency in the deformation and rotation of the front structure. Moreover, the results indicated that the front structure of the simplified models of Explorer and Silverado is also consistent with the whole vehicle models. In contrast, the two simplified models showed a slightly larger rotation around the barrier than the full models. Furthermore, an impressive consistency in the firewall intrusion of the simplified and full model could be seen from the results. In summary, the engine bay structure and passenger compartment structure, including the shotgun, front wheels, firewall, A-pillar, hinge-pillar, and rock padel, have an extremely high deformation consistency between the simplified models and the complete vehicle models[25,43-45].

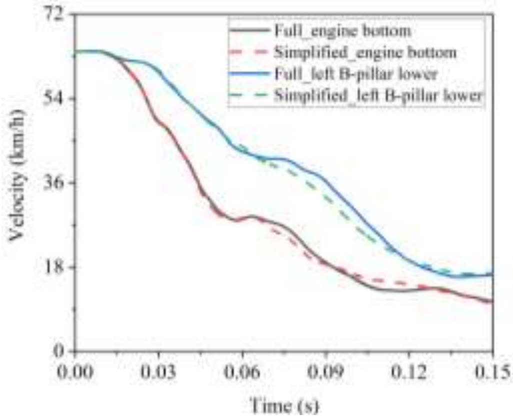
Furthermore, the velocity of the FSRW simplified model is in good agreement with those of the full models. The Caravan's engine bottom acceleration error is the largest among all models, reaching 5.3%; Explorer's left B-pillar lower acceleration error is the largest, reaching 13.4%. At the same time, the average acceleration errors of the engine bottom and the left B-pillar lower are 3.7% and 11.0%, respectively.

Moreover, the energy diagram shown in Figure 13-16 indicated a high consistency in total energy, kinetic and internal energy between the FSRW simplified model and the whole model. The maximum error and average error of total energy are 0.1% and 0.05%, respectively, while the maximum error and average error of internal energy are 0.5% and 0.2%, respectively. Thus, in the small overlap impact, the FSRW simplified model could represent the crash behavior of the entire model characterized by the identical trend in total, kinetic and internal energy. Therefore, those satisfactory correlations provide adequate evidence for the FSRW sub-modeling method adopted in the small overlap impact crashworthiness optimization.

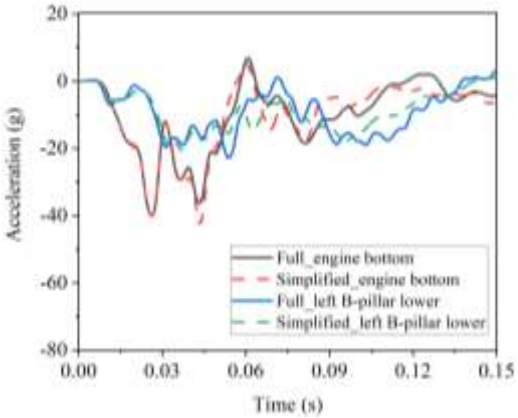
(a) Side view of vehicle deformation

(b) Bottom view of vehicle deformation

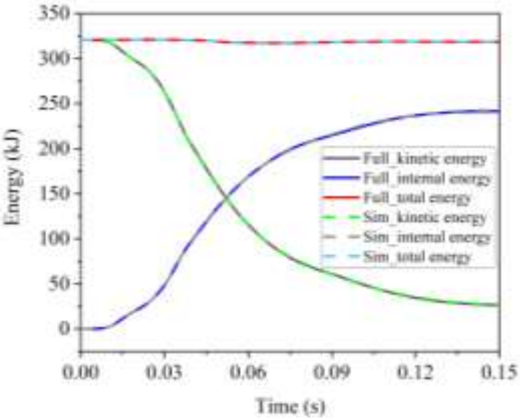
(c) Deformation of firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar



(f) Total, kinetic and internal energy

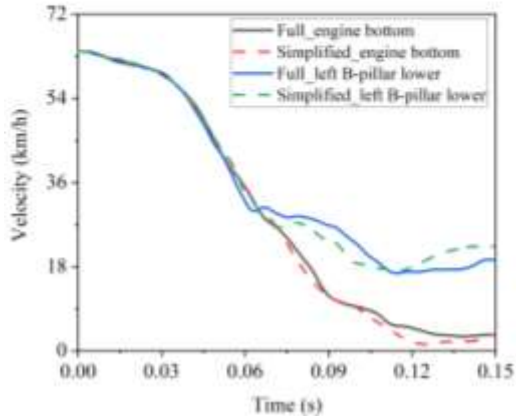
FSRW simplified model Full model

Fig. 13. Comparison FSRW simplified model and complete model of Caravan at small overlap impact

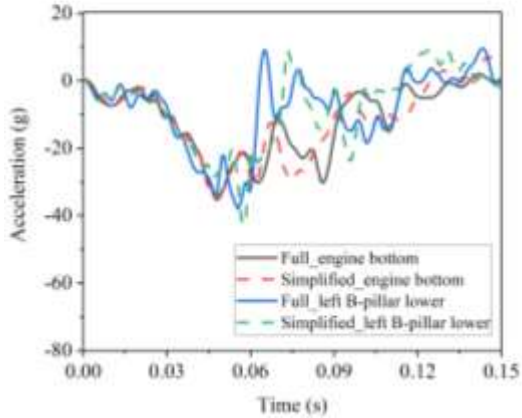
(a) Side view of vehicle deformation

(b) Bottom view of vehicle deformation

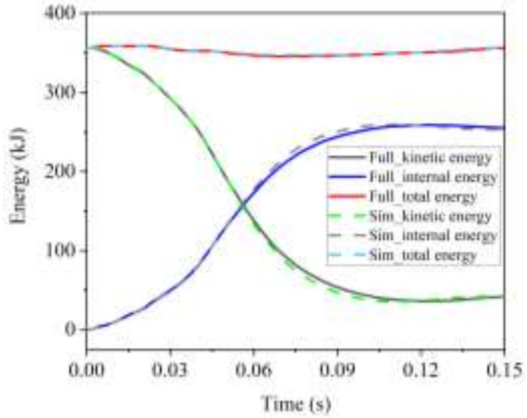
(c) Deformation of firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar

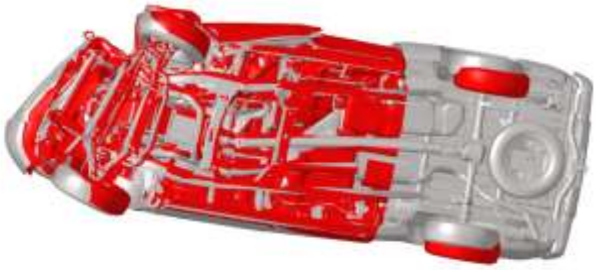


(f) Total, kinetic and internal energy

FSRW simplified model   Full model

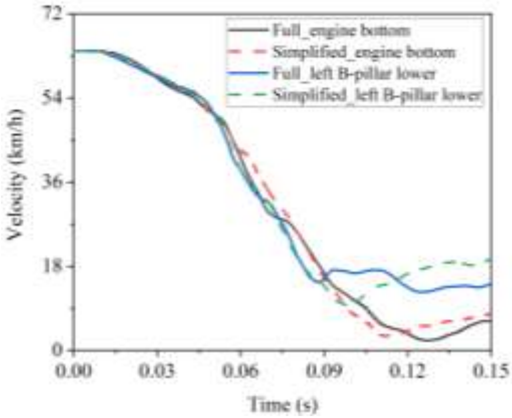
Fig. 14. Comparison of Explorer FSRW simplified model and full model of Econoline at small overlap impact

(a) Side view of vehicle deformation

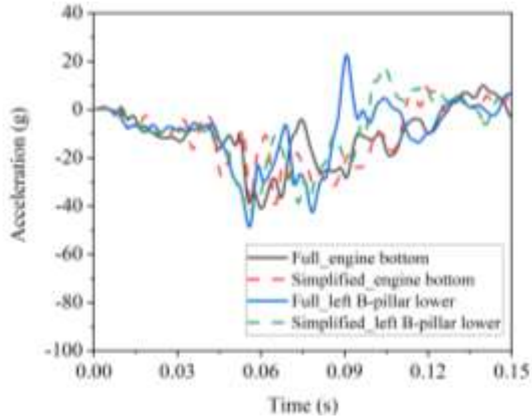


(b) Bottom view of vehicle deformation

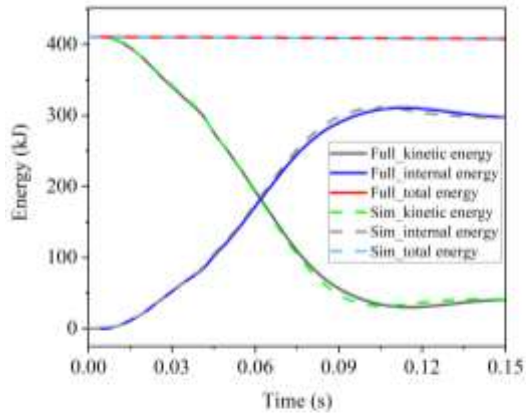
(c) Deformation of firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar

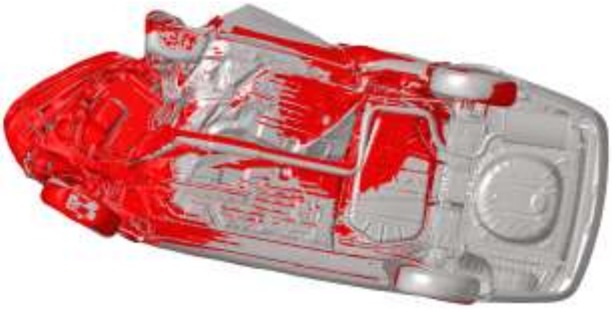


(f) Total, kinetic and internal energy

FSRW simplified model   Full model

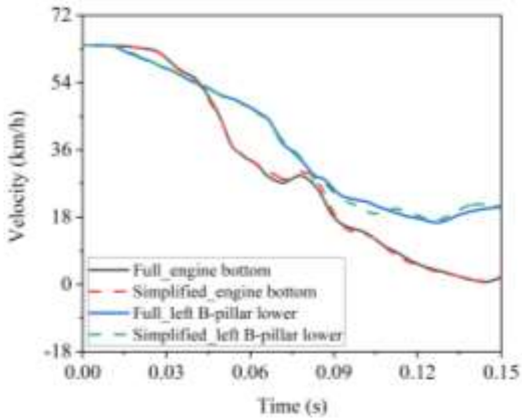
Fig. 15. Comparison FSRW simplified model and full model of Silverado at small overlap impact.

(a) Side view of vehicle deformation

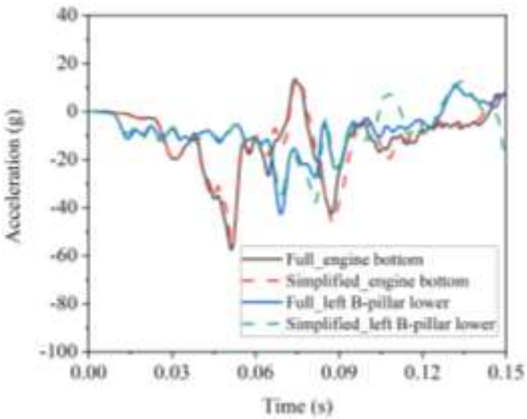


(b) Bottom view of vehicle deformation

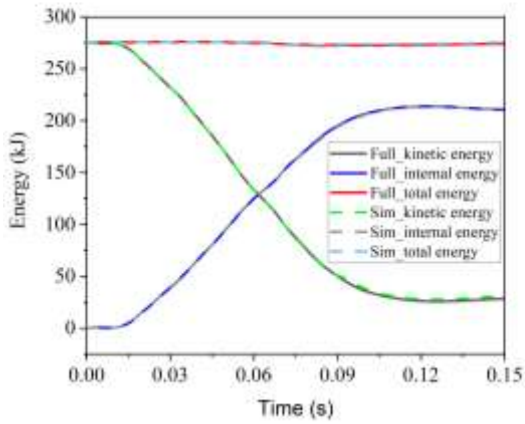
(c) Deformation of the firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar



(f) Total, kinetic and internal energy

FSRW simplified model   Full model

Fig. 16. Comparison of FSRW simplified model and full model of Taurus at small overlap impact.

3.3 Verification of the FSRW sub-modeling in MPDB impact

The deformation pattern between the FSRW sub-modeling simplified and the whole vehicle was shown in Fig. 17-20. A suitable consistency in the deformation and rotation of the front structure of the body could be seen from those simulation results. A slightly larger rotation around the barrier than the full models could be found in the Explorer FSRW simplified model.

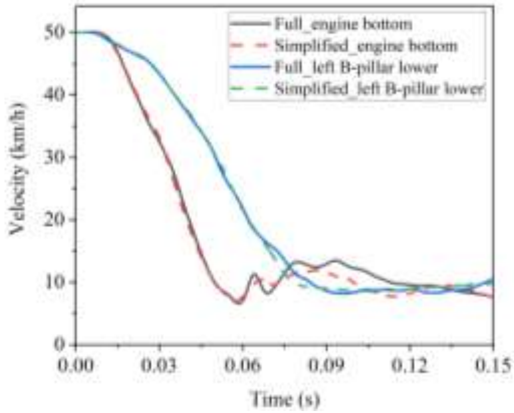
The engine bay structure, A-pillar structure, and hinge pillar structure in the FSRW simplified model and the whole vehicle model, which can significantly affect the MPDB collision results, have a relatively high consistency of deformation form. On the other hand, the obtained deformation in the firewall demonstrated that the FSRW simplified model could represent the intrusion behavior of the whole model.

Furthermore, the velocity of the FSRW simplified model is in good agreement with those of the full models. Moreover, the maximum error of engine bottom acceleration could be seen in the Caravan, reaching 7.1%, while the maximum error of left lower B-pillar acceleration can be found in Explorer, reaching 10.2%. Moreover, the average error of engine bottom acceleration and left lower B-pillar acceleration were 4.6% and 4.7%, respectively. Besides, the maximum error of total energy and internal energy could be seen as 0.06% and 3.4%, respectively, while the average error of the two energy was 0.03% and 1.2%, respectively. Thus, those positive correlations provide a piece of adequate evidence for the FSRW sub-modeling method adopted in the small overlap impact crashworthiness optimization.

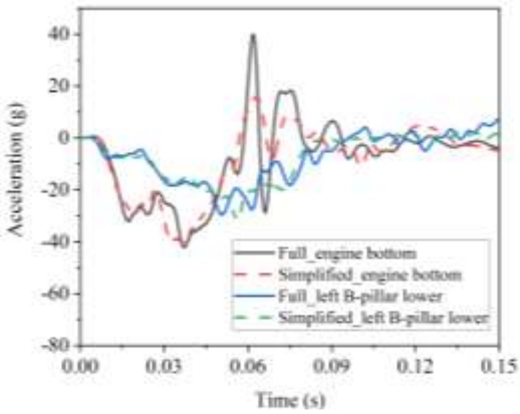
(a) Side view of vehicle deformation

(b) Bottom view of vehicle deformation

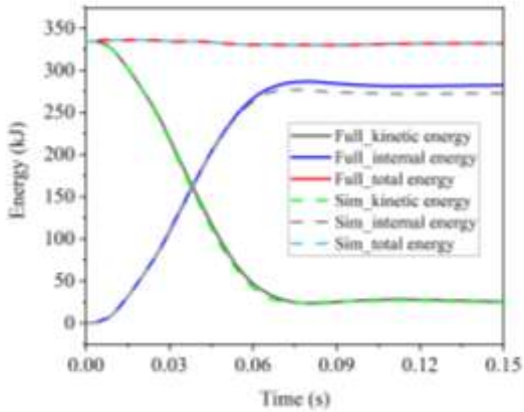
(c) Deformation of the firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar



(f) Total, kinetic and internal energy

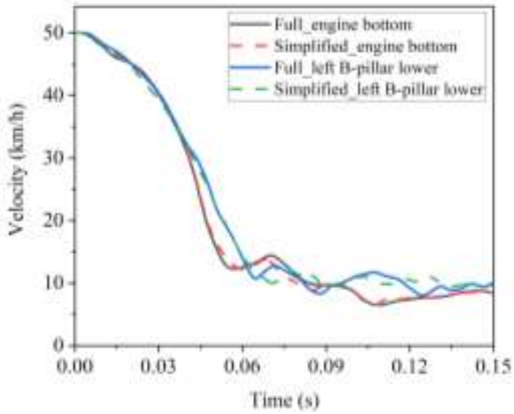
FSRW simplified model Full model

Fig. 16. Comparison of FSRW simplified model and full model of Caravan at small overlap impact.

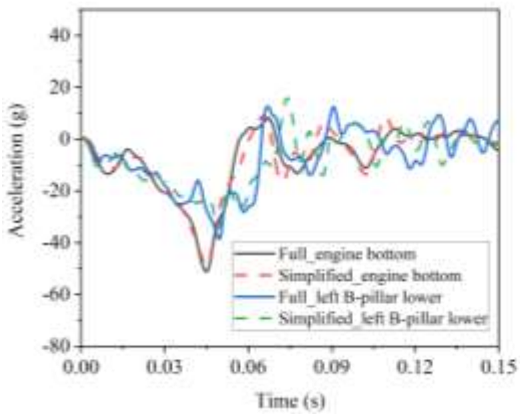
(a) Side view of vehicle deformation

(b) Bottom view of vehicle deformation

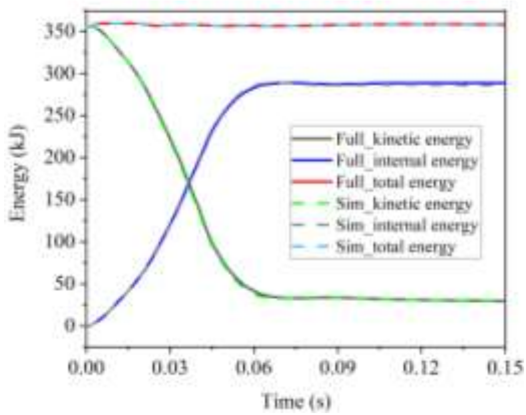
(c) Deformation of the firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar



(f) Total, kinetic and internal energy

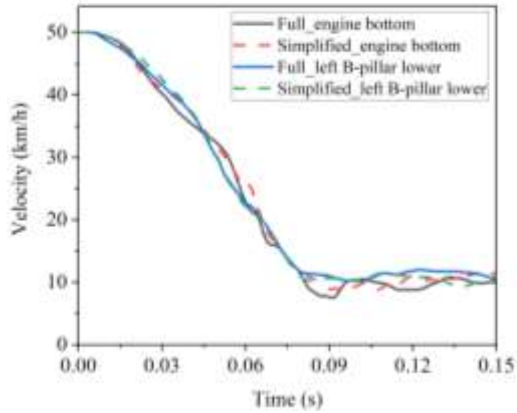
FSRW simplified model Full model

Fig. 16. Comparison of FSRW simplified model and full model of Explorer at small overlap impact.

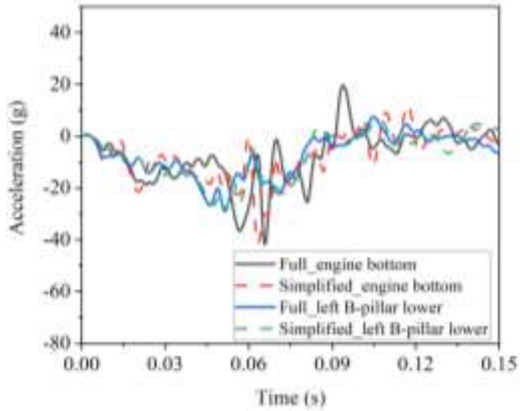
(a) Side view of vehicle deformation

(b) Bottom view of vehicle deformation

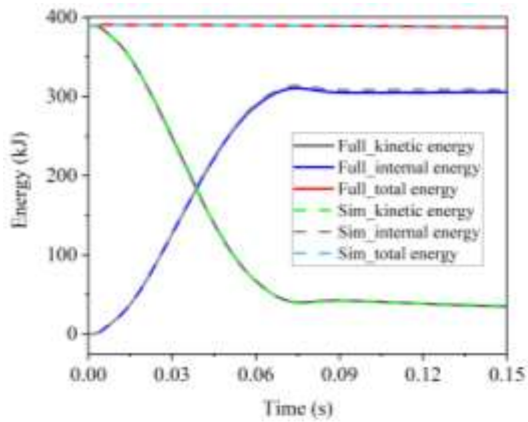
(c) Deformation of firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar



(f) Total, kinetic and internal energy

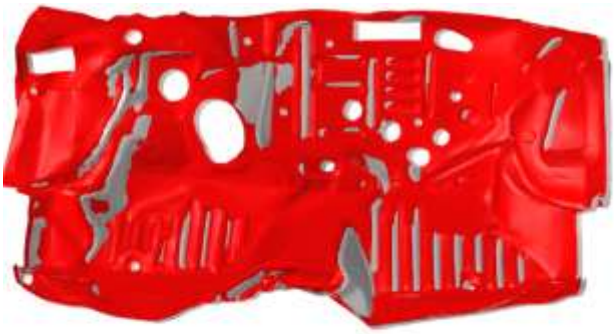
FSRW simplified model Full model

Fig. 16. Comparison of FSRW simplified model and full model

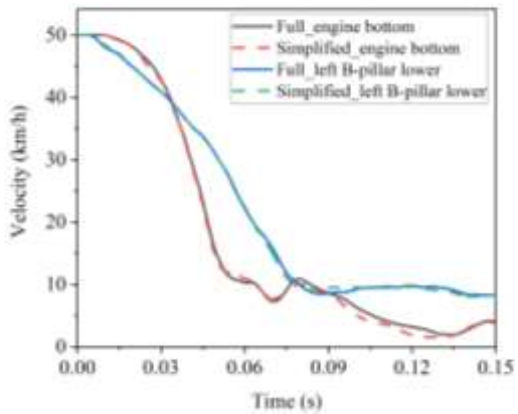
of Silverado at small overlap impact.

(a) Side view of vehicle deformation

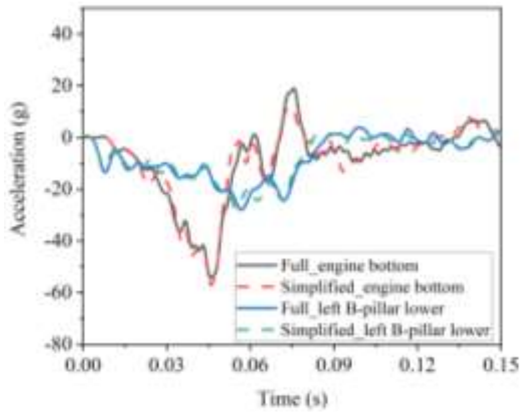
(b) Bottom view of vehicle deformation



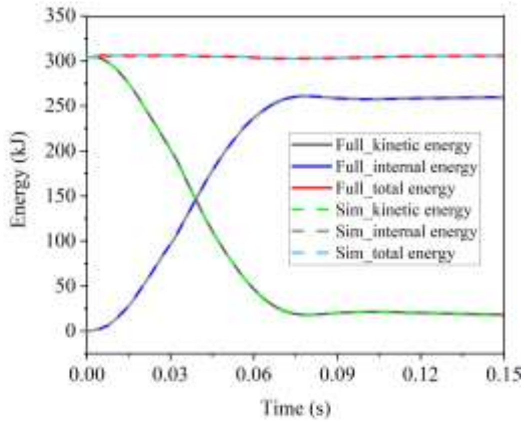
(c) Deformation of firewall



(d) Velocity of engine bottom and left lower B-pillar of gravity



(e) Acceleration of engine bottom and left lower B-pillar



(f) Total, kinetic and internal energy

FSRW simplified model Full model

Fig. 16. Comparison of FSRW simplified model and full model of Taurus at small overlap impact.

3.4 The impact on the computational time

Some crash models may need dozens of hours or even weeks of computational time [7-8]. Thus, the high computational time could be a significant challenge to a vehicle crashworthiness analysis [7]. The FSRW simplified method proposes a simplified method that does not require complicated mathematical deductions and is suitable for engineering applications. This method can reduce the computational time without having a significant influence on the simulation results. All crash simulations were performed on the same workstation with 24 CPUs to evaluate the computational time reduction potential. The impact of the FSRW simplified models on the computational time is shown in Table 1.

Table 1. Reduction of computational time.

Crash type	Vehicle	Simplified/h	Full/h	Reduction/%
FRB	Caravan	2.208	2.907	24.045%
	Explorer	5.388	5.843	7.787%

	Silverado	8.012	9.569	16.271%
	Taurus	10.602	13.617	22.141%
Small overlap impact	Caravan	2.039	2.595	21.426%
	Explorer	4.241	8.504	50.129%
	Silverado	7.717	8.651	10.794%
	Taurus	11.540	12.797	9.823%
MPDB	Caravan	13.957	15.520	10.071%
	Explorer	13.690	14.030	2.423%
	Silverado	14.482	16.273	11.006%
	Taurus	22.189	23.633	6.110%

From Table 1, it can be seen that the computational time for an FRB impact analysis can be reduced by up to 24.045%, with a 17.561% average computational time reduction for those models through using FSRW simplified method. In small overlap impact, the simulation time can be reduced by 23.044% on average and can be reduced by 50.129% at most. Besides, the model with the most significant reduction in computational time in the MPDB impact was reduced by 11.06%, with an average of 7.403% reduces the four models. Those high computational time reduction through using FSRW simplified method enables efficient simulation on frontal impact. In particular, the number of simulations required to build approximation models may be as high as hundreds of times [46]. At this time, the advantages of the FSRW simplified method could be more obvious.

4 Conclusion

In this paper, the FSRW simplified FRB impact, small overlap impact, and MPDB impact were illustrated. Four different kinds of vehicles were adopted to build up FSRW simplified models. The simulation results demonstrated only minor differences in vehicle deformation, firewall intrusion, engine bottom, and B-pillar acceleration using the simplified and vehicle models. However, the calculation time can be significantly reduced. In addition, using this method in FRB impact, small overlap impact, and MPDB impact, the average can be reduced by 17.561%, 23.044%, and 7.403%, respectively. Thus, the FSRW approach has a very high potential to reduce the simulation time and can be applied to the front impact analysis.

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