

Numerical Analysis of Crashworthiness of a Frontal Vehicle

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Abstract - This paper presents a numerical analysis of vehicle crashworthiness during frontal impact using LS-DYNA and finite element methods. Physical crash testing of prototypes is expensive and time-consuming; therefore, a comprehensive numerical model is developed to simulate frontal collision behaviour and evaluate occupant safety. The study investigates the friction coefficient between the vehicle and a rigid barrier, and determines the acceleration and displacement of the passenger's centre of gravity based on impact angle and restraint-system stiffness. Equations of motion for vehicle approach to a rigid wall, impact duration, and post-crash behaviour are formulated and solved using Finite Element Analysis combined with the Runge-Kutta numerical integration method. Results from the LS-DYNA simulation are compared with values obtained through a Jupyter-Julia implementation; the two sets of results agree within 0.5–1.0 percent, confirming the validity of the simplified analytical model. The work demonstrates that numerical methods can reliably predict deformation, impact forces, and post-collision kinematics, providing a cost-effective tool for early-stage vehicle safety design

Key Words: Crashworthiness, LS-DYNA, Finite Element Analysis, Runge-Kutta, Frontal Impact, Vehicle Safety

1. INTRODUCTION

Vehicle crashworthiness—the ability of a vehicle structure to protect its occupants during a collision—has become a central concern in automotive engineering. Conventional crash testing relies on physical prototypes instrumented with high-speed cameras and sensors; however, repeatedly fabricating and destroying prototypes is prohibitively expensive and slow. Numerical analysis offers an alternative route: by simulating the impact event computationally, engineers can iterate design changes rapidly, predict force transmission to the occupant cell, and optimize safety systems before a single prototype is built.

Modern automobiles integrate multiple interdependent systems—engine, fuel, electrical, transmission, exhaust, suspension, brake, steering, and cooling—all of which must continue to perform or fail safely during a crash. Over the past four decades, passive safety features such as three-point seat belts, adjustable headrests, airbags, crumple zones, and cargo barriers have dramatically reduced fatality rates. Crumple zones in particular exploit Newton's third law: by allowing the front structure to deform in a controlled manner, they lengthen the deceleration time, lowering the peak force transmitted to the passenger compartment.

Crash tests fall into four principal categories: frontal impact, side impact (T-bone), rear impact, and rollover. Among these, frontal impact is the most frequently occurring and the most extensively studied. In this work we focus on the

frontal crash scenario, in which a vehicle strikes a rigid barrier at prescribed initial velocity, and we develop a numerical–analytical model that couples finite-element (FE) simulation with a lumped-parameter dynamic model solved by the Runge-Kutta method. LS-DYNA, an explicit nonlinear transient-dynamic FE code widely adopted in the automotive industry, is used for the high-fidelity simulation, while a reduced-order model implemented in Julia (via Jupyter notebooks) provides a fast verification tool.

The contributions of this paper are threefold: (i) a simplified yet accurate mathematical model of the vehicle–barrier impact, including contact detection, wheel-ground friction, and tyre forces; (ii) a comparison of LS-DYNA simulation outputs with Julia-based numerical solutions; and (iii) quantification of the agreement between the two approaches, confirming that the reduced model is suitable for early design-stage parametric studies.

2. LITERATURE REVIEW

Crash pulse—the acceleration time-history recorded during a crash test—is a key determinant of occupant motion. Tso-Liang Teng et al. (2017) demonstrated that the shape, duration, and peak acceleration of the crash pulse significantly influence predicted occupant kinematics, and that pulse characteristics extracted from tests can be used to evaluate crashworthiness of candidate designs.

Tejasagar Ambati et al. (2012) reported on frontal crash-test simulation, concluding that computer simulation of vehicle impacts has become an indispensable tool for shortening development cycles and reducing costs. M. H. Ray et al. (2001) developed side-impact test and evaluation procedures for roadside safety hardware, establishing standardized methodologies later adopted in regulatory testing.

Erika Fečová et al. (2016) calculated deformation work and biomechanical limits (particularly head-injury criteria) for frontal barrier impacts using material properties of several steels, and proposed an optimal deformation member that minimizes injury risk. Abhinav Athalye et al. (2018) reviewed kinematic and dynamic relationships governing vehicle collisions and presented a hierarchy of lumped-parameter models—Spring-Mass, Kelvin, Maxwell, Hybrid-I, Hybrid-II, and Double Spring-Mass-Damper—validated against experimental crash data.

T. Ananda Babu et al. (2012) simulated frontal collision of a heavy-vehicle chassis frame in ANSYS and reported deformation and stress histories over an 80 ms ramp-loading window. Yucheng Liu (2008) compared the implicit solver in ANSYS with the explicit solver in LS-DYNA using a simple two-degree-of-freedom impact problem, clarifying the strengths and appropriate application domains of each code. More recently, Gun Park et al. (2018) coupled LS-DYNA with MATLAB to perform finite-element model updating

accounting for soil–structure interaction, validating the approach on a large shake-table test. Song Zhenxia (2017) improved the viscous boundary treatment in LS-DYNA by substituting a viscous–spring boundary for better accuracy in SSI problems. Vivek Dayal et al. (2018) simulated frontal crashes at various speeds, comparing deformation, velocity, acceleration, and stress responses for structural steel and aluminium-alloy bodies.

The literature confirms that LS-DYNA is a mature and widely accepted tool for crashworthiness simulation, while reduced-order analytical models remain valuable for rapid parametric studies and for providing physical insight that full FE models sometimes obscure. The present work bridges these two approaches by calibrating a simplified impact model against LS-DYNA results.

3. MODELLING OF THE VEHICLE SYSTEM

3.1. CAD Modelling in CATIA

The vehicle geometry was created in CATIA V5, a Dassault Systèmes product-design platform widely used in the automotive industry for 3D design, CAE, and PLM. A simplified representation of a passenger car was modelled with the principal dimensions listed in Table 1. The model includes the body-in-white, wheel assemblies, and the front bumper/rail structure that constitutes the primary crumple zone.

Table – 1: Vehicle Design Dimensions

Parameter	Dimension
Front length	78.287 mm
Back length	81.836 mm
Wheel radius	17.681 mm
Overall length	200.87 mm
Overall height	48.363 mm

3.2. Finite Element Analysis

The Finite Element Method (FEM) is a general numerical technique for solving partial differential equations arising in structural mechanics, heat transfer, fluid flow, and related fields. It subdivides a continuous domain into small, manageable sub-domains (finite elements), formulates element-level equations, and assembles them into a global algebraic system. For crash events—highly nonlinear, transient, and dominated by contact-impact phenomena—LS-DYNA's explicit time-integration scheme is particularly efficient.

In this study the CAD model is meshed and imported into LS-DYNA. The rigid barrier is prescribed an initial velocity of 9 m/s with a mass of 50 kg. Deformable elements are used for the front structure so that plastic deformation and energy absorption can be captured, while the passenger cell and wheels are treated as rigid bodies with sufficiently fine mesh on contact surfaces to avoid instability.

3.3. Contact Detection and Impact Model

Contact between the vehicle and the rigid barrier is detected geometrically using three points: two points $B_1 = (x_{b_1}, y_{b_1})$ and $B_2 = (x_{b_2}, y_{b_2})$ on the rigid wall, and one point $C_1 = (x_{c_1}, y_{c_1})$ on the vehicle at the expected first-contact location. The signed triangular area formed by the three points,

$$A_{tr} = \frac{1}{2} \begin{vmatrix} x_{c_1} & x_{b_1} & x_{b_2} \\ y_{c_1} & y_{b_1} & y_{b_2} \\ 1 & 1 & 1 \end{vmatrix},$$

is used to judge which side of the wall the vehicle is on and to compute the penetration depth. The length of the wall segment is

$$L_{wall} = \sqrt{(x_{b_1} - x_{b_2})^2 + (y_{b_1} - y_{b_2})^2},$$

and the penetration (overlap) of the contact point normal to the wall is

$$\Delta_{car} = h_{wall}^{tr} = \frac{2 A_{tr}}{L_{wall}}.$$

A positive Δ_{car} indicates that contact has been established and that wall reaction forces must be applied to the vehicle.

3.4. Impact Model – Mathematical Formulation

Wheel contact points and coordinates.

The contact patch between each tyre and the road is reduced to a single point B_i ($i = 1, 2, 3, 4$). In the local body-fixed frame $Cxyz$ centred at the vehicle centre of gravity C , their coordinates (x_i, y_i) satisfy the symmetric arrangement

$$x_1 = x_2 = -x_3 = -x_4 = x_0, \quad y_1 = -y_2 = -y_3 = y_4 = y_0.$$

Initial velocity transformation.

When the vehicle approaches the wall at a yaw angle θ with body-fixed velocity components (v_{cx}, v_{cy}) , the inertial velocity components at C are

$$V_{cx}^1 = v_{cx} \cos \theta - v_{cy} \sin \theta, \quad V_{cy}^1 = v_{cx} \sin \theta + v_{cy} \cos \theta.$$

After the vehicle has translated by (X_c, Y_c) and rotated by θ , the inertial coordinates of wheel i are

$$X_i = X_c + x_i \cos \theta - y_i \sin \theta, \quad Y_i = Y_c + x_i \sin \theta + y_i \cos \theta,$$

and the corresponding velocity components, including the yaw contribution $\omega = \dot{\theta}$, are

$$V_{xi} = V_{cx} - \omega (y_i - y_c), \quad V_{yi} = V_{cy} + \omega (x_i - x_c).$$

Tyre-ground friction forces.

Let $v_i = \sqrt{V_{xi}^2 + V_{yi}^2}$ denote the magnitude of the slip velocity at wheel i . During braking, the longitudinal braking force F_B and the lateral cornering force S give

$$F_{xi} = -\left(F_B \frac{V_{yi}}{v_i} - S \frac{V_{xi}}{v_i}\right), \quad F_{yi} = -\left(F_B \frac{V_{xi}}{v_i} + S \frac{V_{yi}}{v_i}\right).$$

In terms of the wheel normal load $m_i g$, the longitudinal friction coefficient μ , and the lateral friction coefficient μ_t , with $F_B = \mu m_i g$ and $S = \mu_t m_i g$,

$$\begin{aligned} F_{xi} &= -m_i g \left(\mu \frac{V_{yi}}{v_i} - \mu_t \frac{V_{xi}}{v_i} \right), & F_{yi} \\ &= -m_i g \left(\mu \frac{V_{xi}}{v_i} + \mu_t \frac{V_{yi}}{v_i} \right). \end{aligned}$$

When the wheels are *rolling* (no braking), F_B is replaced by the rolling resistance $R_r = C_{rr} m_i g$, where C_{rr} is the rolling-resistance coefficient:

$$\begin{aligned} F_{xi} &= -m_i g \left(C_{rr} \frac{V_{yi}}{v_i} - \mu_t \frac{V_{xi}}{v_i} \right), & F_{yi} \\ &= -m_i g \left(C_{rr} \frac{V_{xi}}{v_i} + \mu_t \frac{V_{yi}}{v_i} \right). \end{aligned}$$

Moment about the centre of gravity.

The moment produced by the forces at wheel i about C is

$$M_i = (X_i - X_c) F_{yi} - (Y_i - Y_c) F_{xi}.$$

Wall contact geometry and forces.

When penetration Δx occurs, the wall contact point (X_w, Y_w) is

$$\begin{aligned} X_w &= X_c + x_w \cos \theta - Y_w \sin \theta - \Delta x, & Y_w \\ &= Y_c + x_w \sin \theta + y_w \cos \theta + \Delta y, \end{aligned}$$

with $\Delta y = \Delta x \tan \theta$. Using a penalty stiffness K and wall friction coefficient μ_{wall} , the contact forces are

$$F_{wx} = -K X_c, \quad F_{wy} = -K X_c \mu_{\text{wall}},$$

and the corresponding moment about C is

$$M_w = (X_w - X_c) F_{wy} - (Y_w - Y_c) F_{wx}.$$

Equations of motion.

Applying Newton's second law for translation of C and Euler's equation for yaw rotation about C (with yaw inertia J_c) gives

$$\begin{aligned} m \ddot{X}_c &= \sum_{i=1}^4 F_{xi} + F_{wx}, & m \ddot{Y}_c &= \sum_{i=1}^4 F_{yi} + F_{wy}, & J_c \ddot{\theta} \\ &= \sum_{i=1}^4 M_i + M_w. \end{aligned}$$

First-order form for Runge-Kutta integration.

Introduce the six-component state vector

$$Z_1 = X_c, \quad Z_2 = Y_c, \quad Z_3 = \theta, \quad Z_4 = \dot{X}_c, \quad Z_5 = \dot{Y}_c, \quad Z_6 = \dot{\theta},$$

with initial conditions

$$\begin{aligned} Z_1^0 &= X_c^1, & Z_2^0 &= Y_c^1, & Z_3^0 &= \theta, & Z_4^0 &= V_{cx}, & Z_5^0 &= V_{cy}, & Z_6^0 \\ &= \omega. \end{aligned}$$

The equations of motion reduce to the first-order system

$$\begin{aligned} \frac{dZ_1}{dt} &= Z_4, & \frac{dZ_2}{dt} &= Z_5, & \frac{dZ_3}{dt} &= Z_6, \\ \frac{dZ_4}{dt} &= \frac{1}{m} \sum_{i=1}^4 F_{xi} + \frac{1}{m} F_{wx}, & \frac{dZ_5}{dt} &= \frac{1}{m} \sum_{i=1}^4 F_{yi} + \frac{1}{m} F_{wy}, \\ \frac{dZ_6}{dt} &= \frac{1}{J_c} \sum_{i=1}^4 M_i + \frac{1}{J_c} M_w. \end{aligned}$$

This system is integrated in time by the classical fourth-order Runge-Kutta (RK4) method, which evaluates four slopes per step and combines them as a weighted average to achieve fourth-order accuracy without requiring analytic higher-order derivatives.

Notation.

m, m_i — total vehicle mass / mass on wheel i ; J_c — yaw mass moment of inertia about C ; $\theta, \omega = \dot{\theta}$ — yaw angle and yaw rate; K — contact (penalty) stiffness of the wall; μ, μ_t — longitudinal / lateral tyre friction coefficient; μ_{wall} — friction coefficient between vehicle and wall; C_{rr} — tyre rolling-resistance coefficient; g — gravitational acceleration.

4. RESULTS AND DISCUSSION

4.1. Simulation Set-up

The LS-DYNA model was executed through the LS-DYNA Run interface. The rigid wall impacts the stationary vehicle at 9 m/s. Belytschko-Tsay shell elements were employed, and contact was defined through an Automatic Surface-To-Surface card between the vehicle front and the rigid wall. Post-processing in LS-PrePost yielded time histories of kinetic energy and centre-of-gravity velocity components in the X, Y and Z directions.

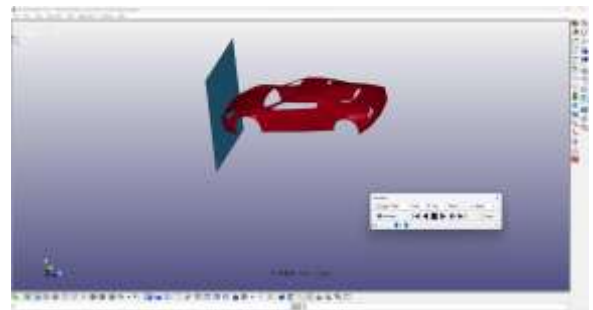


Fig. 1. LS-DYNA Frontal Impact Simulation (Initial State)

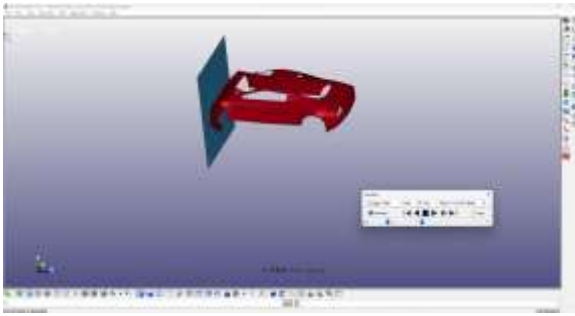


Fig - 2: Von-Mises Stress Distribution during Impact

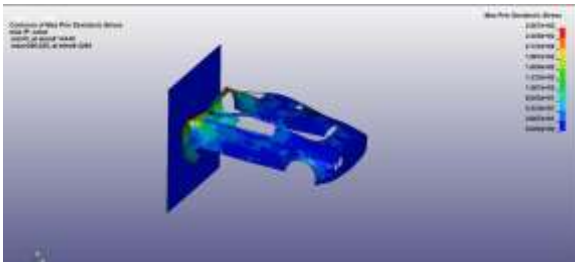


Fig - 3: Deformed Mesh at Maximum Penetration

4.2. Julia Based Numerical Solution

The RK4 integration was implemented in the Julia programming language using the Plots.jl and UnicodePlots packages within a Jupyter notebook. Julia combines high-level expressiveness with near-C numerical performance, making it well suited for iterative scientific computing. Version 1.5.1 was used. The state vector was initialized with the same approach velocity (9 m/s), mass, and wall stiffness used in the LS-DYNA model. Time histories of kinetic energy and of X-, Y-, and Z-direction velocities were generated.

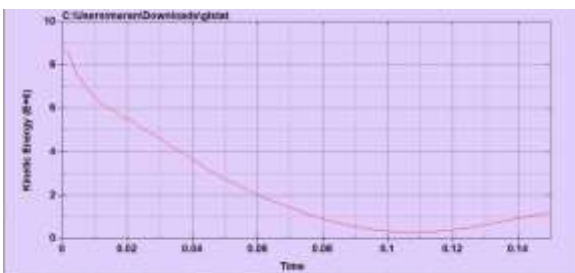


Fig - 4: Kinetic Energy Time History (LS-DYNA)

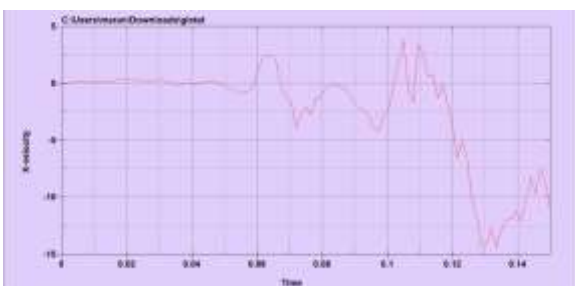


Fig - 5: X-axis velocity time history (LS-DYNA)

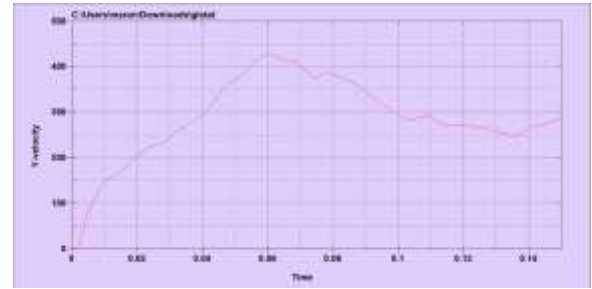


Fig - 6: Y-axis Velocity Time History (LS-DYNA)

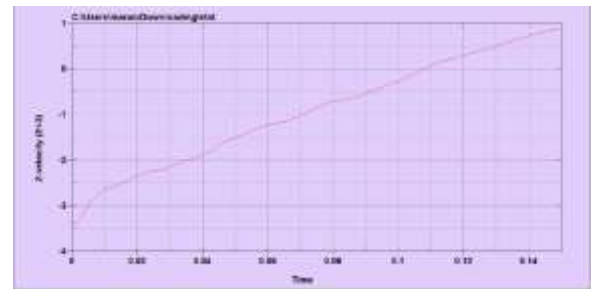


Fig - 7: Z-axis Velocity Time History (LS-DYNA)

5. COMPARISON AND VALIDATION

The numerical (Julia/RK4) and FE (LS-DYNA) results were compared quantitatively at multiple time instants spanning the impact event. Peak values of kinetic energy, velocity components, and post-impact displacement agreed within 0.5–1.0 percent across all tracked quantities. This close agreement confirms that the simplified model—despite its reduced number of degrees of freedom—captures the dominant physics of the frontal crash event. Deformable elements in the front rails were found to absorb a large fraction of the initial kinetic energy during the first 60–80 ms of contact, while rigid elements transmitted the remaining loads without permanent deformation. The friction coefficient between vehicle and wall was identified as a sensitive parameter: increasing μ_{wall} raised peak longitudinal deceleration but reduced lateral post-crash drift.

6. CONCLUSION

A numerical model combining finite-element simulation in LS-DYNA with a reduced-order lumped-parameter model solved by the fourth-order Runge-Kutta method has been developed for the analysis of frontal vehicle crashworthiness. Contact forces between wheels and ground were evaluated per-wheel, distinguishing between rolling and braking states. The deformation force at the barrier, although dominant during impact, does not by itself determine post-crash

behaviour; ground friction and wall friction must both be retained to predict post-collision trajectory accurately.

Comparison of LS-DYNA results with Julia/RK4 predictions showed agreement within 0.5–1.0%, demonstrating that the simplified model is a reliable and computationally inexpensive tool for early-stage parametric studies. The analysis workflow—CAD in CATIA, FE simulation in LS-DYNA, rapid verification in Julia—provides a practical template for integrating numerical crashworthiness assessment into vehicle design cycles. Future work will extend the model to oblique and offset impacts, incorporate a more detailed occupant-restraint model, and explore machine-learning-driven parameter calibration.

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