

Finite Element Analysis of Dynamic Buckling Evolution and Failure Risk in Tubular Strings under Impact Loading

Yafeng Lu¹, Geng Peng¹, Kui Zhang^{2,*}, Zheng Li¹, Hao Gong¹, Xiaodong Yang³ and Qiao Deng⁴

¹ Engineering Technology Research Institute, PetroChina Southwest Oil & Gasfield Company, Chengdu, China

² China Petroleum Engineering Technology Research Institute Co., Ltd., Beijing, China

³ Gas Production Management Office of North Central Sichuan, PetroChina Southwest Oil & Gasfield Company, Suining, China

⁴ State Key Laboratory of Low Carbon Catalysis and Carbon Dioxide Utilization (Yangtze University), School of Petroleum Engineering, Wuhan, China

INFORMATION

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ABSTRACT

As oil and gas exploration progressively extends into deeper and structurally complex reservoirs, dynamic buckling of down-hole tubular strings under impact loading has emerged as a critical issue, posing significant risks to drilling and completion integrity. This paper presents a fully coupled three-dimensional finite element framework that integrates a Cowper–Symonds rate-dependent plasticity model and an Arbitrary Lagrangian–Eulerian (ALE) formulation to capture both strain-rate sensitivity and fluid–structure interaction—two key aspects that have been largely overlooked in prior tubular string buckling analyses. The model incorporates tubular, casing, and fluid domains under representative down-hole conditions and is validated through systematic mesh convergence and energy balance checks. Simulation results indicate that impact loading produces intense stress concentration and abrupt displacement at the tubular string bottom. Axial deformation alternates periodically between compression and tension, reaching a maximum axial displacement of 0.437 cm and exhibiting pronounced hysteretic energy dissipation. Spatially, the response is non-uniform: severe velocity oscillations (peak 9.07 m/s) occur at the bottom, whereas packer-induced constraints attenuate the top velocity by 75.6%. Axial acceleration dominates the response (peak 460.44 m/s²), far exceeding radial components. The numerical analysis further identifies two critical vulnerability zones: the maximum equivalent stress (519.52 MPa) occurs at the top due to wave-reflection superposition, while the initial impact creates a local stress hotspot of 434.35 MPa at the bottom—both locations representing key sites for buckling initiation and subsequent fatigue damage. By quantifying the spatiotemporal evolution of deformation, motion, and stress fields, this study reveals how axial dynamics interact with boundary constraints to trigger dynamic buckling. The findings provide a theoretical basis for the optimum design and safe deployment of deep-well tubular strings, while demonstrating the predictive capability of advanced finite element modeling for complex downhole impact scenarios where experimental data are limited.

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1 Introduction

Driven by the declining availability of conventional resources, oil and gas exploration campaigns are shifting toward deep, geologically complex reservoirs that exhibit extreme temperatures and pressures. Consequently, down-hole tubular strings now experience increasingly severe service conditions, which can trigger complex and unpredictable mechanical responses under combined loading [1–3]. During tripping, impact loads can significantly impair the structural integrity and operational stability (as shown in Fig. 1). These impact loads originate from multiple sources, including shock waves generated by perforating gun detonation, mechanical impacts caused by rod-string failure or detachment, and sudden fluid discharge or jarring forces during fishing operations [4]. To this end, dynamic analysis methods have been increasingly adopted since the late 20th century to investigate the buckling of tubular strings under impact conditions.

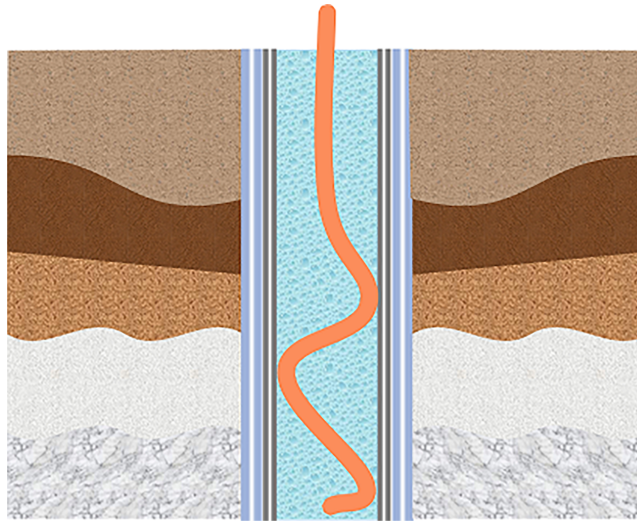


Figure 1: Schematic diagram of tubular string buckling

In 1987, Simitses [5] investigated the stability of undamped elastic structures under both static and dynamic loading, considering three types of dynamic loads: an ideal impulse (Dirac delta), a constant load of infinite duration, and a constant load of finite duration. Even so, the scenario of instantaneous loading followed by unloading isn't investigated. Building on this type of research, Ari-Gur and Elishakoff [6] in 1997 examined the effect of geometric imperfections on the dynamic instability of transversely isotropic columns owing to axial compressive pulses. By accounting for transverse shear deformation and inertial effects, they showed that the buckling load of low-shear-stiffness columns can decrease to two-thirds of the classical theoretical value. While this work contributes to understanding buckling behavior, it fails to elucidate the specific location of geometric imperfections in the tubular string during impact. A further development occurred in 1999, when Lepik [7] derived a general asymptotic expression for elastic–plastic critical buckling loads using a hybrid perturbation–weighted residual method. The accuracy of both theoretical and numerical predictions was validated using user-defined subroutines in ABAQUS, and the influence of thickness variation on buckling loads was examined. Building on these findings, Lepik [8] subsequently investigated the dynamic buckling of elastic–plastic beams under axial compression, considering four distinct loading scenarios. Using Hamilton's principle, he derived the governing equations of motion and proposed a numerical

integration scheme to track the evolution of post-critical deflection modes. In later work, Lepik [9] extended his analysis to include stress wave propagation effects on the dynamic buckling of elastic-plastic beams under axial compression. Assuming linear strain hardening, he developed a Galerkin-based solution applicable to arbitrary degrees of freedom. While Lepik et al. established foundational theories for the dynamic buckling of beams, their models did not account for the multi-component nature of downhole tubular strings.

In 2000, Karagiozova and Jones [10] investigated the axisymmetric dynamic buckling of cylindrical shells under axial impact from rigid projectiles. Based on numerical simulations of the deformation process, they developed an analytical model capturing the spatiotemporal coupling between stress wave propagation and buckling mode evolution. Their results showed that the buckling path is highly sensitive to the impactor's kinetic energy dissipation and to the shell's radius-to-thickness ratio. However, their study did not account for the interactions between multi-component structures. In 2012, Baumann et al. [11] utilized numerical models to predict transient pressure waves and structural loads on related components in oil and gas wells. They also assessed the sensitivity of tubular string design parameters to impact loads. A notable limitation of this study is that its research provided insufficient analysis of the dynamic buckling behavior of the tubular string itself. In 2015, Miller et al. [12] systematically characterized buckling mode transitions and post-buckling bifurcations of elastically constrained bars using a combined experimental and nonlinear finite element framework with contact-friction coupling. Nonetheless, their model did not account for material plasticity or strain-rate effects.

In 2018, Aarsnes and van de Wouw [13] performed a dynamic stability analysis of a distributed drill-string system. By deriving a minimal parameter set for linearized axial-torsional dynamics, they elucidated complex dynamic behaviors and emphasized the importance of distributed models in vibration suppression. Yet their study focused on vibration suppression and did not address dynamic buckling under impact loading. In 2021, Blostotsky et al. [14] built an experimental setup for dynamically determining the critical tensile load of elastic bars. Unfortunately, it overlooks the wave reflection, contact interactions, or buckling mode evolution. In 2023, Han [15] established models in ABAQUS for tubular string slippage and impact; the results indicate that, under the same friction, longer strings develop lower axial stress during slippage, whereas smaller wellbore inclination angles produce larger displacement, velocity, and stress during impact. Be that as it may, her model did not consider fluid-structure coupling.

In 2024, Guo [16] investigated the influence of perforation impact pressure on the axial force of tubular strings based on the fluid-structure coupling principle; however, it should be noted that the relevant research on the damage location of tubular strings and the time-history evolution of displacement is lacking. In 2025, Ding et al. [17] employed numerical simulation to reveal the dominant role of piston force in tubular string buckling when the expansion joint is in a stretched state. Yet, they did not study the piston force under impact conditions. In 2026, Nora Benmir and Akroum [18] established a theoretical analysis framework for the effect of continuous external forces on the stability of a tubular string, based on the Euler-Bernoulli beam theory combined with the wave propagation characteristics of periodic structures. Regrettably, their modeling strategy failed to consider nonlinear factors in the wellbore.

In summary, impact-induced dynamic buckling of tubular strings remains a central research theme. However, key challenges persist in accurately predicting critical loads and characterizing post-buckling deformation. In transient dynamic buckling, mode evolution is governed by the coupling between temporal load characteristics and inertial effects, resulting in bifurcation with competing instability paths. In contrast, quasi-static stability analysis follows the framework of eigenvalue decomposition theory, predicting stability boundaries by identifying critical parameters associated with the lowest-order buckling mode.

Dynamic buckling responses are highly sensitive to load characteristics, with buckling behavior strongly dependent on the loading type. Moreover, impact-induced dynamic buckling exhibits pronounced time-dependent behavior: transient high-energy loads trigger inertial effects through stress wave propagation, resulting in dynamically evolving buckling modes. Critical load criteria are governed by the synergistic effects of loading rate, boundary dynamics, and material strain-rate sensitivity. Numerous studies have employed theoretical, numerical, and experimental approaches to examine the effects of boundary conditions, step loading, axial inertia, shear deformation, and other factors on critical load, time, and length in rod or tubular string dynamic buckling. However, research specifically addressing the dynamic buckling of tubular strings under impact loading remains scarce, highlighting the need for further investigation.

Accordingly, this study develops a fully coupled 3D finite element model using the Cowper–Symonds constitutive law and ALE formulation to capture strain-rate effects and fluid–structure interaction, which have been largely overlooked in prior tubular string analyses. By quantifying spatiotemporal deformation, motion, and stress distribution, this work identifies critical vulnerability zones (bottom and top sections) and reveals how wave reflection and superposition drive dynamic buckling—findings that extend beyond traditional quasi-static design frameworks.

To provide a clear overview of the study’s logical progression, [Fig. 2](#) presents a flowchart outlining the key stages of the research. The investigation begins with problem definition and identification of research gaps through a literature review. Subsequently, a three-dimensional finite element model is developed, incorporating geometric simplifications, material constitutive laws, boundary conditions, and fluid-structure interaction via the ALE method. Numerical simulations are then performed using an explicit solver, with rigorous validation through mesh convergence and energy balance checks. The simulated results are systematically analyzed in terms of deformation, motion, and strength characteristics to identify critical vulnerability zones. Finally, engineering recommendations and conclusions are drawn based on the findings. This flowchart serves as a roadmap for the methodology and analysis presented in the following sections.

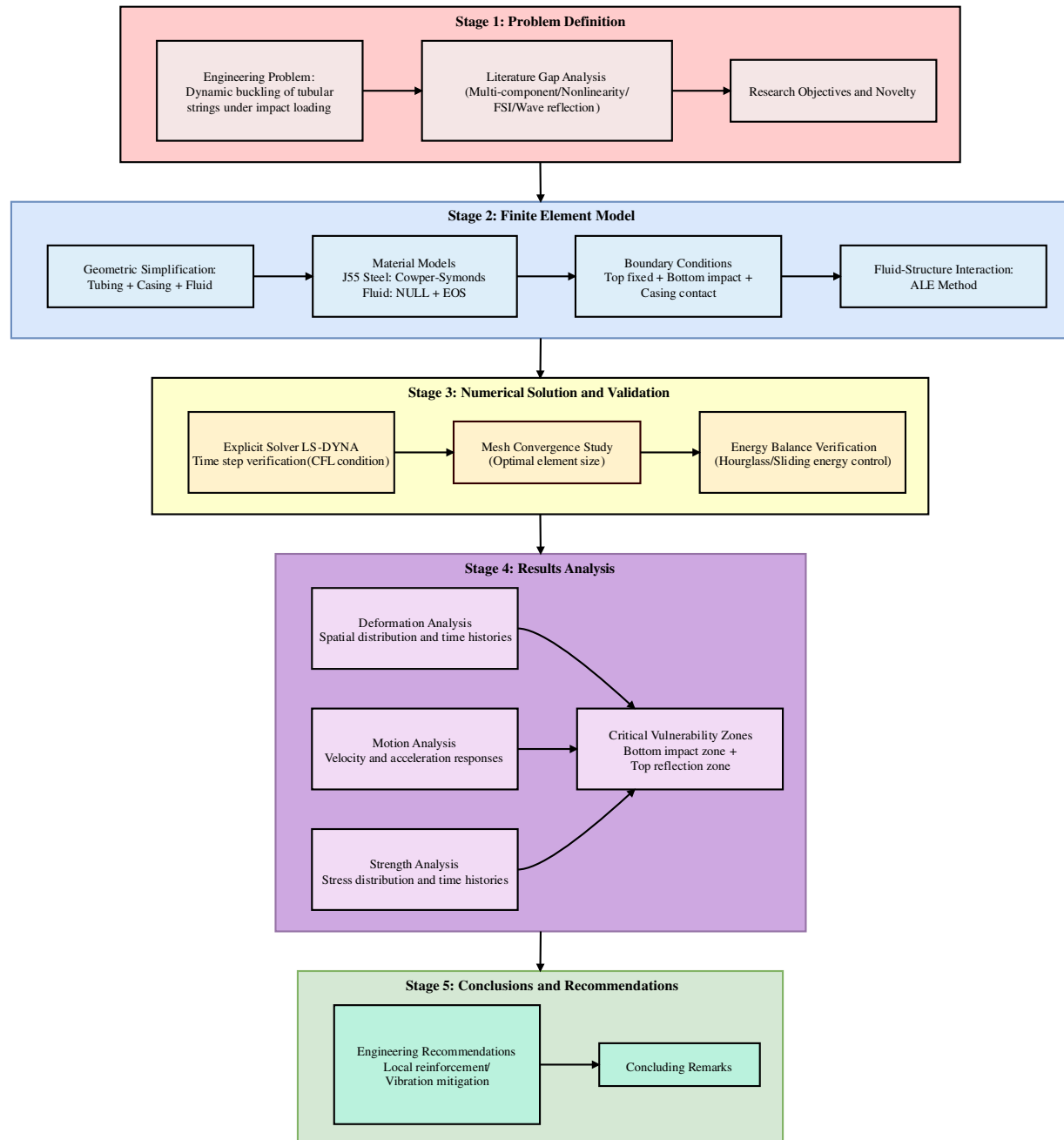


Figure 2: Flowchart of the research framework

2 Establishment of Finite Element Model

2.1 Geometric Model

During dynamic impact loading, energy dissipation in downhole tubular string systems induces several coupled dynamic mechanisms. The resulting structural response is multi-scale. This response

appears in three main aspects: (a) The tubular string, which consists of multi-stage threaded connections, efficiently transmits mechanical loads. (b) A variety of down-hole components—completion tools, gauges, and auxiliary devices—mechanically couple to form a complex multi-body dynamical system. (c) The constitutive behavior of these components differs markedly due to contrasting material compositions: connectors and shock absorbers are usually fabricated from high-strength alloys, which offer superior load-bearing capabilities, whereas tubular strings and packers are typically made of ductile materials, favoring flexibility and deformation resistance. Consequently, model simplification tailored to specific operating conditions is essential prior to simulation.

Analysis of actual downhole operations indicates that a typical tubular string contains casing, tubular, packer, high-strength connector, pressure control valve, and perforating gun. A representative field configuration of such a tubular string is illustrated in Fig. 3. Given its relatively low yield strength and susceptibility to bending deformation, the tubular segment is identified as the critical component for buckling analysis in this dynamic numerical study.

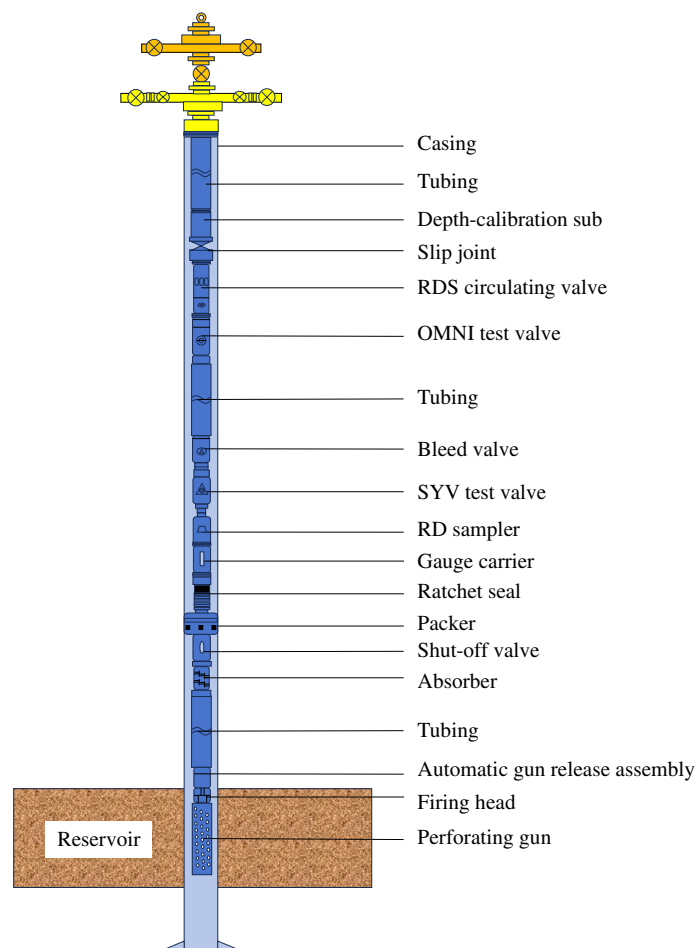


Figure 3: Test, perforation, acid pressure, plugging, completion and production integration string

Although downhole tubular strings exhibit structural diversity, their functional units are mechanically coupled—e.g., through threaded connections—into an integrated composite structure capable of continuous load transfer.

In accordance with oilfield operational requirements and industry standards, a simplified tubular string model is developed using structural equivalence principles, assuming isotropic material behavior.

The resulting mechanically equivalent assembly is illustrated in Fig. 4, consisting of three core components: tubing, casing, and the fluid. The boundary conditions in the model are imposed as follows, based on actual downhole physical constraints: (a) Top boundary (packer location), the top end of the tubular string is fully fixed, simulating the constraint imposed by a packer that anchors the string to the casing. This is implemented using nodal displacement constraints applied to all nodes at the top cross-section of the string. This simplification is consistent with field conditions where packers provide high-stiffness radial and axial restraint. (b) Vertical displacement, the bottom nodes are free to move in the axial direction, allowing the string to respond to impact loading. (c) Rotational constraints, vertical rotation (i.e., rotation about horizontal axes), are constrained to maintain structural stability, while free rotation about the vertical axis is permitted, reflecting the actual downhole condition where torsional freedom exists. (d) Casing constraint, the outer surface of the tubular string is in potential contact with the inner surface of the casing. Radial expansion and lateral displacement of the string are constrained through contact interaction with the casing. The casing itself is modeled with its outer boundary fixed in all directions, simulating the restraint provided by the cement sheath and surrounding formation.

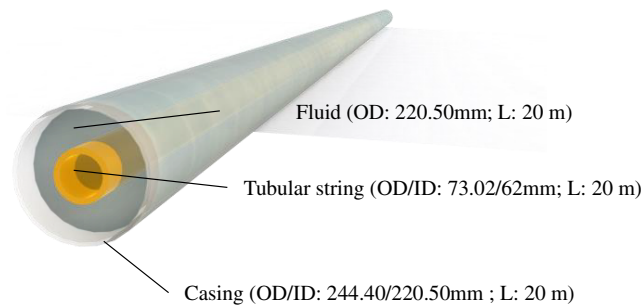


Figure 4: Finite element model of tubular string system

The impact loading applied to the tubular string is defined as a pressure load on the bottom-end cross-section, with a specific load-time history derived from field measurements and previous experimental studies. The detailed definition is as follows: (a) Uniform pressure applied to all elements on the bottom-end surface of the tubular string. (b) The pressure varies with time in two phases: Phase 1 (0–500 μ s), pressure increases rapidly from 0 MPa to the peak value of 168.72 MPa; Phase 2 (500–5000 μ s), pressure gradually decays from the peak value to zero. The decay represents the dissipation of impact energy after the initial shock.

To ensure the reliability and accuracy of the finite element simulations, tubing and casing materials are assigned the J55 steel, consistent with applicable tubular string design standards and field practice. Detailed material parameters are provided in Table 1.

Table 1: Material parameters of tubular system

Component	Elasticity modulus (GPa)	Yield strength (MPa)	Poisson's ratio (-)
String	210	552	0.3
Casing	210	450	0.25

2.2 Constitutive Models and State Equations

In finite-element simulations, result accuracy depends not only on an efficient and robust numerical framework, but also on appropriate constitutive models and precisely calibrated material parameters [19,20]. Under impact loading, the response of the tubular string is multi-scale and highly nonlinear. Three features govern the dynamic mechanical response of tubular string materials: (a) Yield strength increases significantly under dynamic loading; (b) Instantaneous stress is highly strain-rate sensitive. (c) Deformation exhibits strain-rate history dependence, necessitating special treatment.

The first two phenomena, collectively termed the strain-rate effect, are central to the study of dynamic material behavior [21,22]. Under dynamic impact, metallic materials exhibit a significant increase in yield strength with increasing strain rate—a phenomenon known as strain-rate hardening. The Cowper–Symonds model is one of the most widely adopted constitutive relations for describing this behavior in engineering materials subjected to high-rate loading. It expresses the dynamic yield stress as a function of the quasi-static yield stress and the effective strain rate, making it particularly suitable for problems involving stress wave propagation and transient dynamic response. The Cowper–Symonds model has been extensively validated for steel materials under impact conditions. Jones [23] demonstrated the applicability of the Cowper–Symonds relation for structural impact problems, providing experimental validation for various steel grades. Within this framework, the yield stress is expressed by Eqs. (1) and (2) [24]:

$$\sigma_m = \left[1 + \left(\frac{\varepsilon_m}{Y_1} \right)^{\frac{1}{Y_2}} \right] (\sigma_0 + \alpha_m E_{mp} \varepsilon_{mp}^{eff}) \quad (1)$$

$$E_{mp} = \frac{E_{tan} E}{(E - E_{tan})} \quad (2)$$

where, σ_m is strain-rate-dependent yield stress; ε_m is the strain rate; Y_1 and Y_2 are material-specific strain-rate parameters; σ_0 is the quasi-static yield stress (i.e., the value measured at low strain rates); E_{mp} is the plastic hardening modulus; ε_{mp}^{eff} is the effective plastic strain.

Unlike solid materials, fluids (gases and liquids) do not possess a yield strength or shear resistance under static conditions. The NULL material model is specifically designed for materials that sustain no deviatoric stresses and require only an equation of state to define the pressure–volume relationship [25]. This makes it ideally suited for modeling air and water in fluid–structure interaction problems.

The LINEAR_POLYNOMIAL equation of state (EOS) was chosen to describe the thermodynamic behavior of the air domain, which expresses the pressure–volume–temperature (PVT) as a function of the density ratio and internal energy [26]. The general form is given in Eq. (3):

$$P_{gas} = (A_1 + A_2 \mu_{gas} + A_3 \mu_{gas}^2 + A_4 \mu_{gas}^3) + (A_5 + A_6 \mu_{gas} + A_7 \mu_{gas}^2) E_{gas} \quad (3)$$

where, P_{gas} is the gas pressure; A_1 – A_7 are the constants of the equation of state, set to 0, 0, 0, 0, 0.4, 0.4, respectively; μ_{gas} is the density ratio, the density of air is 1.225 kg/m³.

The Grüneisen EOS is commonly used to capture the response of liquids at high pressure, especially under impulsive loading such as impact-generated shocks [27], and is written as Eq. (4):

$$P_H = (\gamma_0 + \delta\alpha_m) E_m + \frac{\rho_m C_m^2 \alpha_m \left[1 + \left(1 - \frac{\gamma_0}{2} \right) \alpha_m - \frac{\delta}{2} \alpha_m^2 \right]}{\left[1 - (S_1 - 1) \alpha_m - S_2 \frac{\alpha_m^2}{\alpha_m^2 + 1} - S_3 \frac{\alpha_m^3}{\alpha_m^2 + 1} \right]} \quad (4)$$

where, P_H is annulus pressure, γ_0 is taken as a constant with a value of 0.35; δ is the correction factor for the equation of state; ρ_m is the initial density of the medium; C_m is the intercept of the shock wave velocity curve; S_1 – S_3 are the slope coefficients of the stress wave velocity curve in the material; α_m is the medium's compression coefficient.

Downhole impact simulations are conventionally based on either a Lagrangian or Eulerian description. The Lagrangian algorithm is prone to severe mesh distortion under large deformation, whereas the Eulerian algorithm encounters difficulties in tracking material interfaces, often necessitating excessively refined meshes and a consequent increase in computational cost.

To mitigate these drawbacks, a topology-based adaptive-mesh algorithm that monitors the deformation-gradient tensor in real time is employed: refinement is introduced in regions of high non-linearity, while coarsening is applied where strain gradients are low, thereby simultaneously enhancing accuracy and efficiency.

Furthermore, the arbitrary Lagrangian–Eulerian (ALE) formulation combines the advantages of both Lagrangian and Eulerian descriptions by introducing a reference frame that moves independently of material particles and spatial coordinates [28]. This allows: (a) Accurate tracking of fluid–structure interfaces. (b) Accommodation of large deformations in the fluid domain without mesh distortion. (c) Efficient coupling between structural and fluid solvers.

2.3 Mesh Convergence and Model Validation

The string and casing are discretized with Lagrangian meshes, whereas the annular fluid domain is represented by ALE meshes. The complete model consists of hexahedral elements; a cross-section of the simulation model meshes is given in Fig. 5.

To reproduce the physics of down-hole impact, the numerical model must resolve the load pulse faithfully, yet the mesh must be designed so that accuracy and computational cost are optimally balanced—an inherently complex task that places heavy demands on resources. A systematic mesh convergence study was conducted prior to the final simulations to determine an appropriate element size that balances computational accuracy and efficiency. The study involved discretizing the tubular string, casing, and fluid domain using four different mesh resolutions, with characteristic element lengths of 3, 5, 6, 7, and 10 mm. For each mesh, the same boundary conditions, material parameters, and impact loading were applied, and the simulations were run for the full 5000 μ s duration. The convergence behavior was evaluated by monitoring the peak von Mises stress at the top end, which is critical to the dynamic buckling analysis. The convergence criterion adopted was that the relative change in the key output parameter should fall below 1% when refining the mesh. As shown in Table 2:

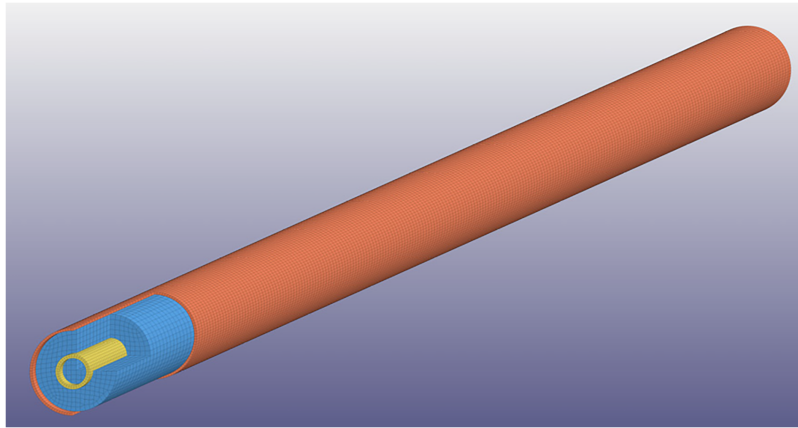


Figure 5: Sectional diagram of mesh model

Table 2: Mesh convergence analysis

Element Size (mm)	Peak Von Mises Stress (MPa)	Relative Change (%)
10	491.3	/
7	515.1	+4.8
6	519.4	+0.83
5	520.5	+0.21
3	520.9	+0.08

Refining from 10 to 7 mm produced changes of approximately 4.8%, indicating that the 10 mm mesh was too coarse to capture the response accurately.

Refining from 7 to 6 mm reduced the relative changes to below 1% (0.83% for stress), meeting the convergence criterion.

Further refinement from 6 to 5 mm and from 5 to 3 mm produced minimal additional changes (all below 0.25%), indicating that the solution had effectively converged.

Based on these results, the 6 mm element size was identified as the optimal choice: it satisfies the convergence criterion while maintaining a reasonable computational cost. The 5 and 3 mm meshes, although slightly more accurate, required substantially more computational resources without providing meaningful improvements in accuracy.

An explicit time-integration scheme is adopted. The ALE algorithm proceeds through a staged coupling strategy: at each time step, a Lagrangian update is executed first; boundary conditions of the resulting deformed mesh are preserved while the fluid domain is remapped in the ALE region. Throughout, mesh topological consistency is maintained, and the updated Lagrangian field variables are transferred to the newly generated ALE mesh, guaranteeing both numerical accuracy and physical consistency.

To ensure the reliability of explicit numerical simulations, the time step size must be set correctly. The actual computational time step was set to a fixed value of $1\ \mu\text{s}$ throughout the entire simulation. This choice was made after verifying the Courant–Friedrichs–Lewy (CFL) stability condition. For the smallest element size of 6 mm and the wave speed of steel $\left(c = \sqrt{\frac{E}{\rho}} \approx 5170\ \text{m/s}\right)$, the theoretical critical time step is:

$$\Delta t = \frac{L_{\min}}{c} = \frac{0.006\ \text{m}}{5170\ \text{m/s}} \approx 1.6\ \mu\text{s}$$

The adopted fixed time step of $1\ \mu\text{s}$ is therefore below this critical limit, providing a safety margin of approximately 16% and ensuring numerical stability. With this time step, the stress wave propagates about 5.17 mm per increment, which is less than the characteristic element length, guaranteeing that wave propagation is resolved accurately.

No mass scaling was applied in any of the simulations. Mass scaling artificially increases the density of selected elements to allow a larger time step, but it also alters the inertial response and can distort wave propagation phenomena. Because the dynamic buckling mechanism investigated here is highly sensitive to inertial effects and stress wave propagation, preserving the true material density was essential. The computational cost with a fixed $1\ \mu\text{s}$ time step was acceptable: the full 5000 μs simulation required approximately 9 h with the adopted 6 mm mesh. Hence, no artificial acceleration of the solution was needed.

In the explicit numerical simulations of impact dynamics, the energy composition and evolutionary characteristics of the system serve as one of the core quantitative indices for evaluating the validity of numerical calculations, the credibility of results, and the rationality of the model, which directly reflect the presence of non-physical energy distortion and numerical errors during the calculation process. For this purpose, the variation curves of three key energy components—total energy, hourglass energy, and sliding energy—during the simulation were accurately extracted (as shown in Fig. 6). As clearly indicated by the curves, the hourglass energy remained close to zero throughout the simulation, with its proportion to the total energy stably below 5%. This demonstrates that the mesh division and algorithm settings of the model effectively suppressed the hourglass distortion prone to occur in explicit calculations, thus avoiding non-physical deformation and spurious energy dissipation caused by the mesh zero-energy mode. Meanwhile, the sliding energy exhibited no negative values over the entire calculation cycle, which is consistent with the basic physical law of energy conservation, and its proportion to the total energy was also controlled within 5%. This indicates that the simulation of energy transfer and dissipation during the contact-sliding process of the tubular string conforms to the actual mechanical behavior, with no abnormal energy generation or loss observed. In summary, the energy evolution characteristics of the numerical simulation in this study are fully consistent with the validity criteria for explicit calculations, which fully verifies the reliability and accuracy of the simulation results. This provides a solid and credible numerical reference basis for the subsequent in-depth analysis of the dynamic mechanical response, buckling evolution law, and failure risk assessment of the tubular string under impact loading.

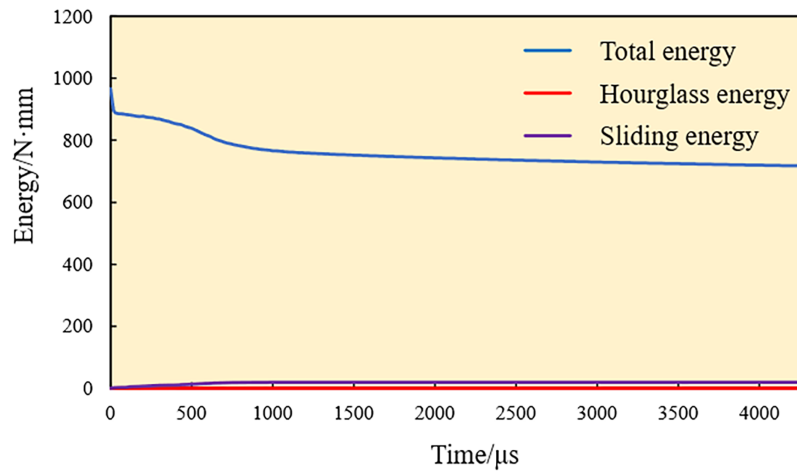


Figure 6: Energy balance check

3 Simulation Results and Discussion

3.1 Deformation Analysis

Based on the finite-element simulations (performed using a unit system where centimeters, grams, and microseconds are the base units), displacement contour plots showing regions with equal movement of the tubular string are extracted at different time intervals. These contour plots provide an intuitive view of the string's dynamic response to a sudden force, known as impact loading (Fig. 7).

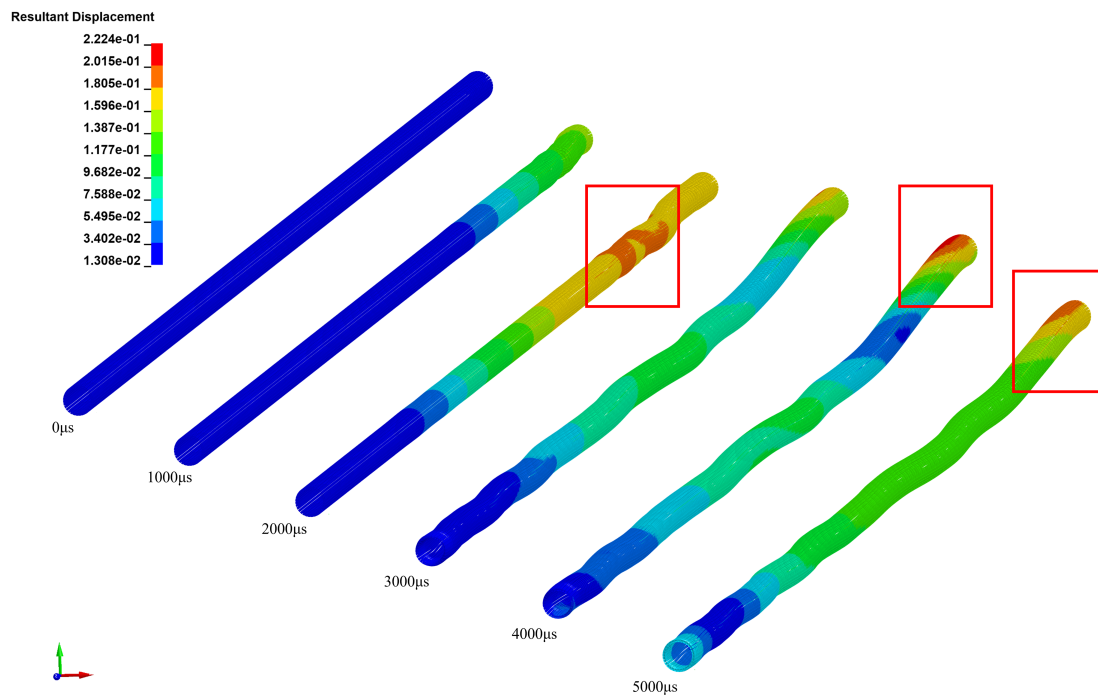


Figure 7: Variation of displacement of tubular string at different time intervals

The predicted deformation pattern correlates with field observations in which buckling is usually localized at the bottom of the string, as shown in Fig. 8. The strong agreement between simulation results and field data demonstrates that the finite element method reliably captures the dynamic buckling behavior under impact loading.

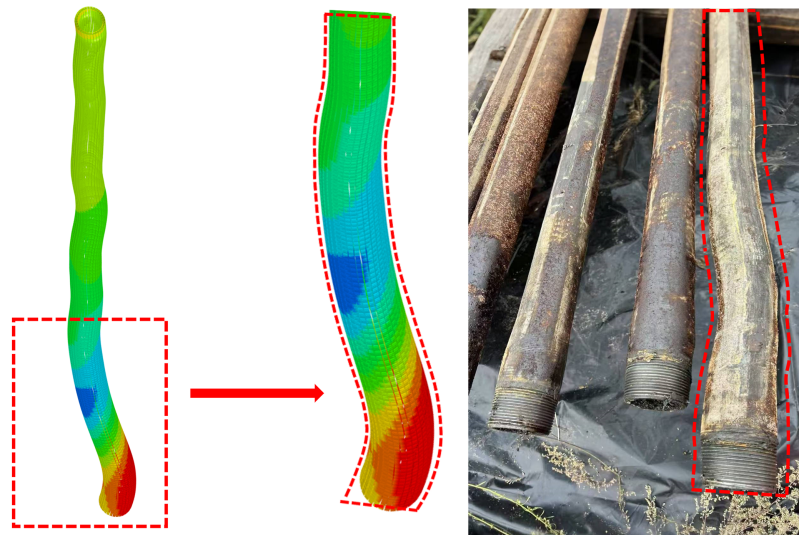


Figure 8: Buckling shape of the tubular string

Fig. 7 reveals pronounced spatiotemporal heterogeneity in the deformation. At $0 \mu\text{s}$, the load has not yet reached the bottom of the string, so the entire string remains undeformed. By $1000 \mu\text{s}$, the impact front arrives at the lower end and produces highly localized deformation; because the load is transient, displacement is concentrated at the impacted cross-section. At $2000 \mu\text{s}$, the stress wave has travelled to the mid-section, producing noticeable deflection in the lower half of the string, whereas the top—restrained by the packer—exhibits negligible movement. At $3000 \mu\text{s}$, the wave continues to propagate upward, distributing energy over a longer region. Reflections become evident at $4000 \mu\text{s}$ and amplify the displacement at the bottom. By $5000 \mu\text{s}$, the mid- and upper sections have undergone further deflection, but the largest deformation still occurs at the bottom, where the incident and reflected waves superpose, and the mechanical stress is highest.

This pattern aligns with field observations (Fig. 8), confirming that the finite-element approach accurately captures the dynamic buckling behavior of a tubular string under impact loading.

To quantify the string's bottom end deformation, displacement histories are extracted from a reference node at the base of the string in three orthogonal directions: radial X, radial Y, and axial Z. The resulting displacement-time curves are given in Fig. 9, where positive and negative values denote displacement relative to the center of the bottom surface, taken as the origin. Positive X-displacement indicates rightward motion, negative leftward; positive Y-displacement signifies forward motion, negative backward. In the Z-direction, positive values correspond to upward axial displacement, and negative values to downward.

Under impact, the string's bottom end moves in all three directions, with the Z-component exhibiting the largest amplitude. This axial dominance reflects both the longitudinal stiffness of the string and the predominantly axial path of the stress wave.

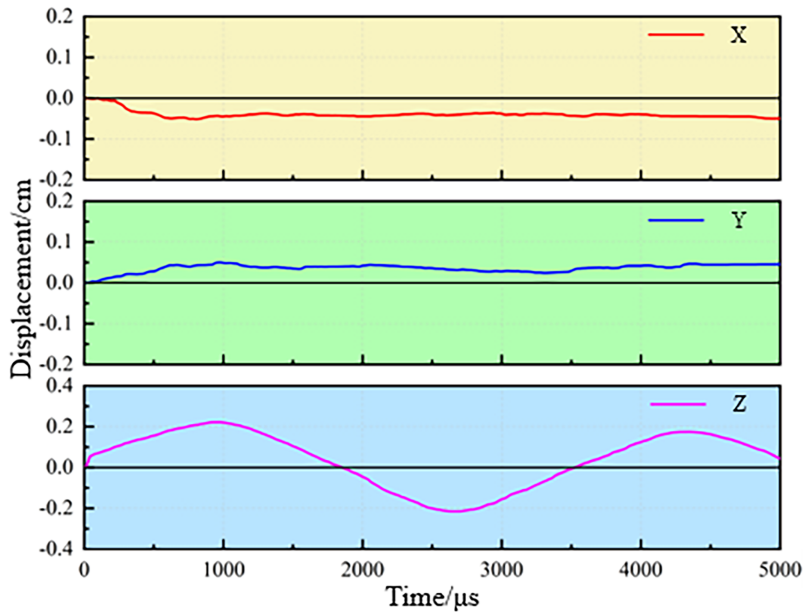


Figure 9: Displacement response of the tubular string bottom in three directions

Radial (X and Y) deformations are essentially shear. X-displacement is strictly negative, implying leftward unilateral motion; Y-displacement remains strictly positive, denoting forward radial motion. Axial (Z) displacement shows a near-sinusoidal waveform. Under transient impact, the axial response bifurcates into two phases: (a) compression, with upward displacement increasing monotonically to a peak of 0.222 cm; (b) tension, where downward displacement grows as the wave propagates, reaching 0.215 cm. The resultant peak-to-peak axial deformation in the first cycle is therefore 0.437 cm. Wave reflections in the second cycle produce renewed compression accompanied by energy dissipation. This hysteretic response evidences strong longitudinal structural coupling and underscores the nonlinear deformation mechanisms under dynamic loading.

High-intensity, short-duration impact generates large transient stresses at the string bottom, thereby localizing deformation in the lowermost section. Upward axial impact forces are dominant. Bottom-up displacement grows monotonically with the impact wave until peak compression is reached. Subsequently, reflected waves then induce downward (tensile) displacement. Repeated impact-reflection cycles generate alternating tension and compression, driving fatigue damage. In the radial directions, however, the presence of casing constraints limits displacement, permitting only minor deformation to occur.

3.2 Movement Analysis

As shock-wave energy attenuates along the string, a spatially heterogeneous axial response emerges, producing distinct kinematic signatures at the top, middle, and bottom sections. Velocity histories extracted from these three locations are plotted in Fig. 10.

Fig. 10 shows that during the initial impact phase, the bottom of the string undergoes pronounced velocity oscillations, peaking at +9.07 and −5.16 m/s. This region experiences the strongest impact, and the intense fluctuations are consistent with near-field wave-propagation theory.

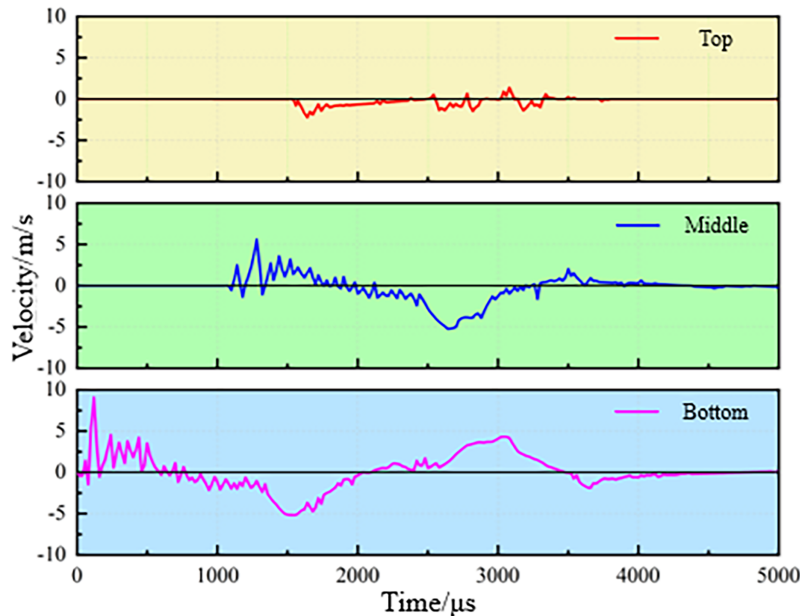


Figure 10: Axial velocity response at different positions of the string

As the stress wave propagates upward, energy attenuation becomes evident in the middle section, where velocity amplitude drops markedly. Peak positive velocity falls to 5.61 m/s ($\approx 38\%$ attenuation), while peak negative velocity is -5.22 m/s. This nonlinear attenuation conforms to viscoelastic wave-dissipation theory, in which frequency dispersion distorts the waveform and increases response complexity.

At the top, velocity is sharply reduced by boundary constraints and further energy attenuation, yielding peaks of $+1.37$ and -2.22 m/s. Constrained-dynamics analysis reveals that packer stiffness suppresses vibration, resulting in a velocity attenuation of approximately 75.6%. This boundary-induced modal shift forces the top section into a forced-vibration regime.

The analysis indicates that the shock wave initially creates a high-energy concentration zone at the bottom, generating a pronounced velocity transient. Upon reaching the middle section, energy dissipation reduces velocity amplitude by 38.2%. By the time the wave reaches the top, the packer-induced constraint strongly modifies the response, leaving minimal velocity variation at the upper node. The intense axial oscillation is identified as the primary driver of instability. Repeated compression-tension cycles can precipitate dynamic buckling, especially in the bottom and middle sections.

Triaxial acceleration histories extracted from a bottom-end mesh block are plotted in Fig. 11.

As illustrated in Fig. 11, the peak accelerations in the X (148.73 m/s²) and Y (185.40 m/s²) directions are significantly lower than in the Z direction (460.44 m/s²), indicating a pronounced axial response. The low radial peaks are attributed to the geometric and boundary constraints of the tubular string: the outer casing provides an annular confinement that dissipates radial vibration energy and reduces acceleration amplitudes. In contrast, the markedly higher axial peak acceleration reflects direct vertical transmission of the impact load. Moreover, geometric constraints such as the top packer

cause multiple reflections and superposition of axial stress waves, significantly amplifying the axial acceleration.

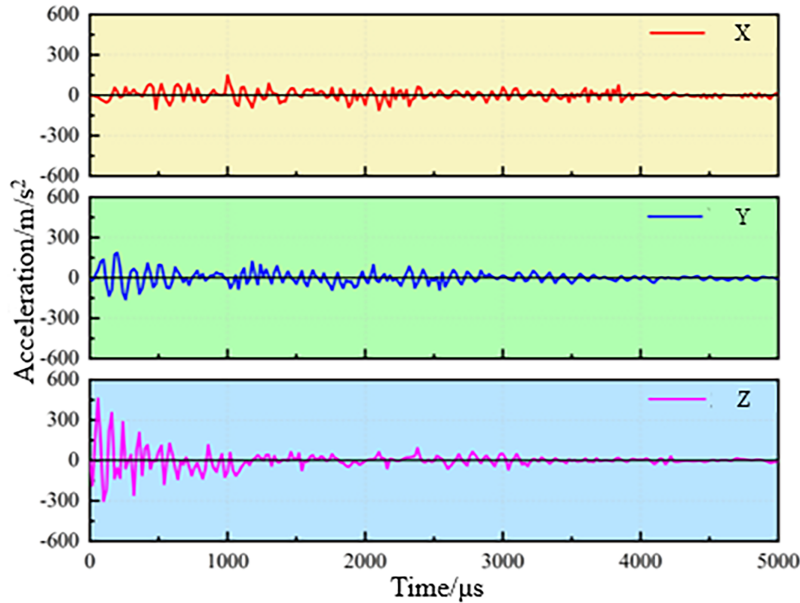


Figure 11: Acceleration response of string in three directions

Furthermore, under impact loading, the triaxial acceleration time-histories exhibit oscillatory decay. This nonlinear vibration arises from multimodal coupling induced by reflection, transmission, and mode conversion of stress waves at material interfaces. Over time, acceleration amplitudes attenuate, and oscillations stabilize, indicating progressive energy absorption and dissipation within the tubular system.

3.3 Strength Analysis

By analyzing string deformation and motion, the displacement, velocity, and acceleration responses to impact loading are clearly revealed. To further quantify the structural strength under impact, time-varying von Mises stress contour plots ($\times 10^5$ MPa) are presented in Fig. 12.

Fig. 12 highlights the strong temporal variation of the von Mises stress field within the tubular string. At 0 μ s, the system is in static equilibrium with zero axial stress throughout. By 2000 μ s, the stress wave has propagated rapidly, producing a pronounced concentration at the string's bottom end. By 3000 μ s, the load has travelled upward along the string, and von Mises stress increases in both the middle and top sections. At 4000 μ s, the wave reaches the top (packer) and reflects, causing a rapid rise in stress; the fixed constraint further amplifies stress. By 5000 μ s, the reflected wave has traveled back toward the bottom, superposing in the mid-lower section.

Von Mises stress histories extracted from structural blocks along the string are plotted in Fig. 13. Owing to the shock wave, the von Mises stress at the bottom of the string rapidly reaches 434.35 MPa. The middle section initially experiences lower stress, but axial propagation of the disturbance raises it to 349.75 MPa. At the top, the packer constraint limits deformation, yet reflection causes the stress to jump to 519.52 MPa.

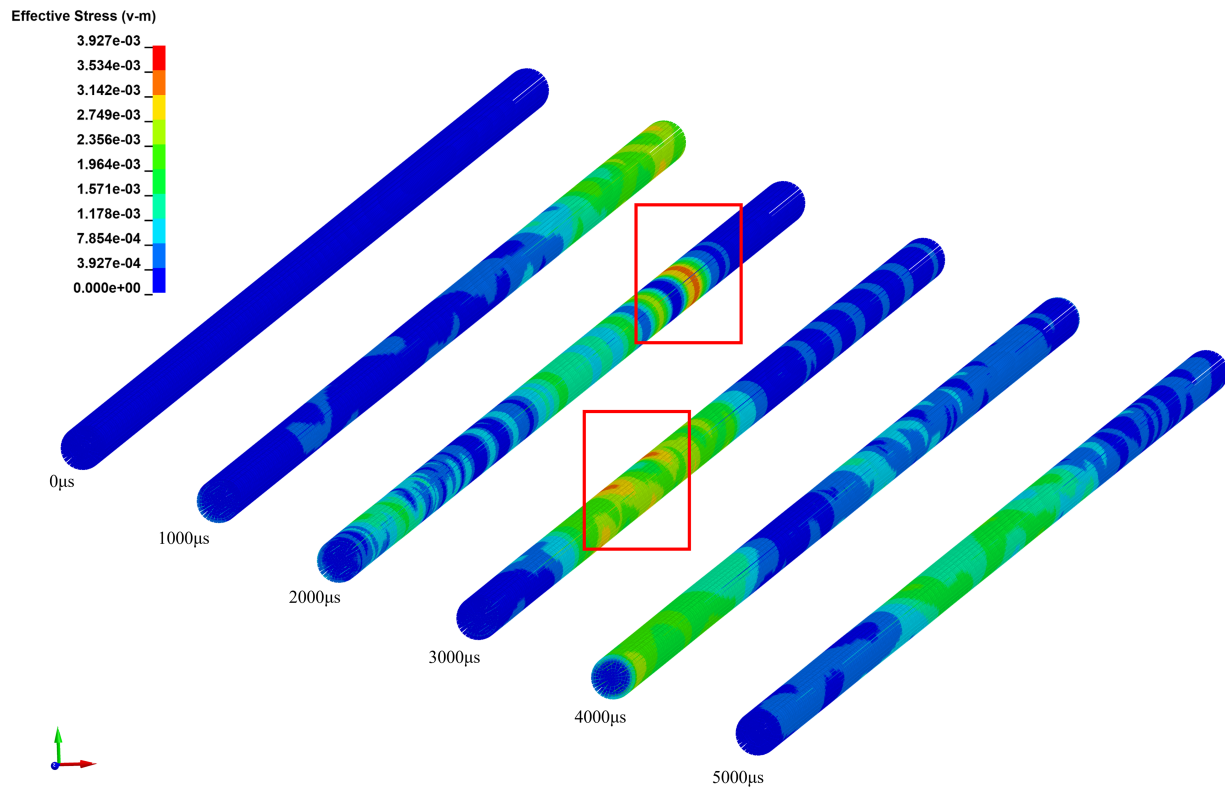


Figure 12: Equivalent stress response of the tubular string

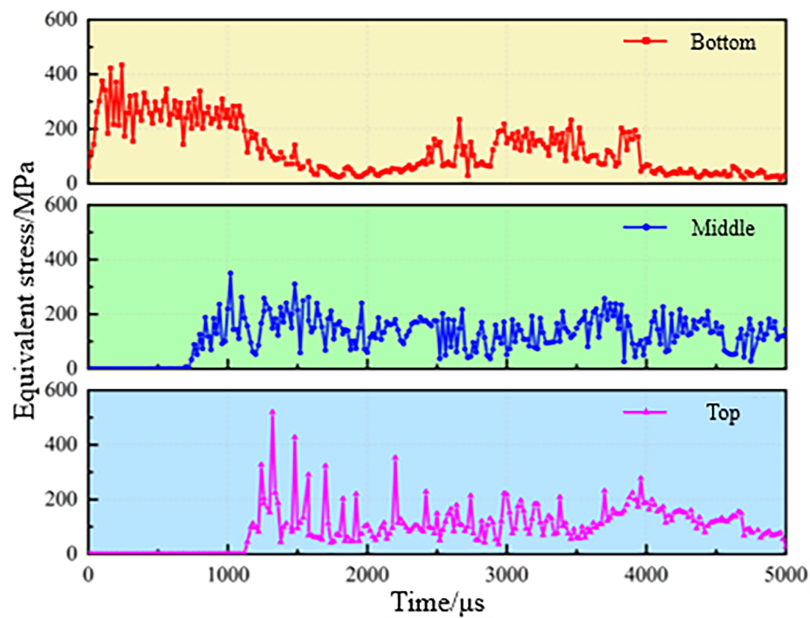


Figure 13: Variation of equivalent stress at different positions of the tubular string

The von Mises stress at the bottom, middle, and top of the string exhibits oscillatory periodicity. Upon bottom reflection, the wave re-interacts with the string, generating new stress peaks. Reflection and superposition produce extreme stresses at different locations, completing one cycle. The packer yields the highest stress peak, demonstrating its role in stress amplification. Thus, under impact, stress first peaks at the bottom, then at the middle, and subsequently at the upper end. Wave reflection and superposition are the primary mechanisms of instability, leading to local buckling and eventual fatigue failure.

3.4 Discussion

The simulation results reveal that impact loading induces pronounced spatiotemporal heterogeneity in the deformation, motion, and stress distribution of tubular strings. Axial displacement dominates the dynamic response, with a peak-to-peak deformation of 0.437 cm in the first cycle, accompanied by hysteretic energy dissipation. This finding aligns with the theoretical framework established by Lepik [7–9], who emphasized that axial inertia and stress wave propagation are key drivers of nonlinear deformation in elasto-plastic structural members under impact. However, the present study further quantifies the cyclic alternation between compression and tension—attributed to the superposition of incident and reflected waves—a mechanism that was not explicitly characterized in previous dynamic buckling analyses of tubular strings. The localized deformation at the string bottom, observed in both simulation results and field observations (Fig. 8), confirms that the impact initiation site is the primary region for early-stage buckling, consistent with the failure modes reported in industry case studies.

Velocity and acceleration analyses highlight the critical role of boundary constraints and energy attenuation in regulating the dynamic response. The bottom section exhibits intense velocity oscillations (peak: +9.07 m/s), while the packer-induced constraint attenuates the top velocity by 75.6%—consistent with Aarsnes and van de Wouw’s conclusion that distributed constraints modulate drill string dynamic behavior [13]. Notably, the axial acceleration (peak 460.44 m/s²) far exceeds the radial components, a phenomenon attributed to the annular confinement of the casing, which limits the dissipation of radial vibration energy. This observation supplements the findings of Han [15], who focused on the effects of wellbore inclination but overlooked the dominant contribution of axial dynamics to instability. The intense axial oscillations identified herein underscore that repeated compression–tension cycles are the primary trigger for dynamic buckling, particularly in the bottom and middle sections of the string.

Stress distribution analysis identifies two critical vulnerability zones: the bottom (local stress hotspot: 434.35 MPa) and the top (maximum equivalent stress: 519.52 MPa). The bottom stress concentration arises from direct impact, while the top stress peak is induced by wave-reflection superposition at the fixed packer boundary. This dual-hotspot phenomenon extends the understanding of Karagiozova and Jones’s [10] research on cylindrical shell buckling, which emphasized the sensitivity of buckling paths to boundary conditions but did not address stress superposition effects in tubular string systems. The stress values observed in this study indicate that dynamic impact can push the material close to its yield limit, thereby increasing the risk of fatigue damage and buckling failure during prolonged service—an important consideration that is largely overlooked by conventional quasi-static design frameworks [1–3].

The constitutive modeling and numerical methodology adopted in this study offer several key advantages over previous approaches. The Cowper–Symonds model effectively captures the material’s strain-rate sensitivity, a critical factor for accurately predicting yield behavior under high-rate impact loading. The ALE algorithm overcomes the limitations of pure Lagrangian and Eulerian methods,

enabling reliable simulation of large deformations and fluid-structure interaction without excessive mesh distortion. This methodological advancement addresses the shortcomings of earlier numerical studies that either neglected strain-rate effects or failed to fully couple fluid and structural dynamics—leading to less accurate predictions of critical buckling conditions.

Despite these contributions, this study has certain limitations that warrant future investigation. First, the model simplifies the tubular string as a homogeneous structure, overlooking the influence of threaded connections and material inhomogeneities—factors that may introduce local stress concentrations under impact. Second, the simulation assumes a single impact pulse, whereas downhole operations often involve multiple consecutive impacts (e.g., during fishing or perforating operations). Third, the effect of temperature variations, which are significant in deep reservoirs, was not incorporated into the constitutive model. Future research should integrate these critical factors to develop more comprehensive and robust predictive tools.

This study delivers practical insights for the design optimization and safety enhancement of tubing strings. The identification of critical vulnerability zones—namely, the bottom and top sections—implies that localized reinforcement measures, such as increasing wall thickness or employing high-strength alloys in these regions, could effectively reduce buckling risk. Furthermore, the quantification of dynamic response parameters (e.g., displacement, velocity, and stress) provides a basis for designing impact-absorbing devices or real-time monitoring systems capable of detecting incipient instability. For example, the observed 75.6% velocity attenuation induced by the packer suggests that optimizing packer stiffness represents a promising strategy for vibration suppression.

In summary, this study elucidates the dynamic buckling evolution of downhole tubing strings under impact loading via an integrated numerical framework. The resulting insights into spatiotemporal deformation, stress distribution, and critical vulnerability zones provide a foundation for enhancing the design and operational safety of strings in deep, complex reservoirs. These findings address key knowledge gaps and deliver actionable engineering guidance, ultimately supporting the sustainable extraction of deep oil and gas resources.

4 Conclusion

This study employs numerical simulation to investigate the dynamic mechanical behavior of downhole tubular strings under impact loading and to track the evolution of dynamic buckling and associated mechanical states. The results show that the bottom of the string, as the first region of impact, undergoes the most pronounced deformation and largest displacement. Triaxial displacement analysis further reveals that the largest displacement component is in the axial direction Z . This arises from the initial upward impact force at the bottom, which induces positive (i.e., compressive) axial displacement. As the stress wave propagates along the string, the axial deformation accumulates, with the most severe compression remaining at the bottom because of its rapid response.

The motion data reveal that the initial impact generates a high-energy concentration zone at the string's bottom end, producing a sharp velocity transient. Owing to boundary constraints and energy attenuation, the stress wave undergoes nonlinear velocity decay as it travels toward the wellhead. Throughout this process, intense oscillations compromise string stability, and repeated compression–tension cycles trigger dynamic buckling, especially in the bottom and middle sections.

Triaxial motion analysis shows that casing-induced annular confinement severely restricts radial movement and dissipates much of the radial vibrational energy. In contrast, multiple reflections and superposition within the confined annulus amplify the axial stress wave, yielding axial acceleration

amplitudes that exceed their radial counterparts. It follows that prolonged impact will induce fatigue damage, while the strong axial dynamic response renders the string susceptible to dynamic buckling.

Nodal stress data reveal local von Mises stress concentration at the bottom under impact, increasing the risk of buckling or local instability. The peak stress appears during initial wave impact, and subsequent cyclic deformation repeatedly compromises structural integrity. Although the stress wave propagates upward relatively slowly, the fixed packer boundary conditions lead to a global von Mises maximum at the top, especially under reflected-wave.

In conclusion, the bottom and top sections of the tubular string constitute the critical vulnerable zones. The bottom undergoes concentrated displacement, acceleration, and stress under direct impact, rendering it prone to local buckling. The packer-constrained top region exhibits significant stress concentration—particularly under reflected waves—attaining the global maximum and potentially inducing buckling or deformation. Therefore, axial dynamic response at these two locations should be prioritized in both analysis and design.

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