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Geng Peng¹, Yafeng Lu¹, Yexin Fang^{2,*}, Song Zeng¹, Junyu Pu¹,
Chongrong Chi¹ and Liu He³

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

² CNPC Engineering Technology R&D Company Limited, Beijing, China

³ Exploration Division, Southwest Oil & Gasfield Company, Petro China, Chengdu, China

INFORMATION

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ABSTRACT

As global conventional oil and gas resources continue to deplete, exploration and development are increasingly targeting deep and ultra-deep reservoirs with high-temperature, high-pressure (HPHT) conditions. This paper presents a comprehensive numerical investigation based on a non-linear finite element model that integrates impact–contact–fluid coupling. The computational framework employs the Cowper–Symonds rate-dependent constitutive law, calibrated via Split Hopkinson Pressure Bar tests, and an Arbitrary Lagrangian–Eulerian (ALE) multiphysics algorithm to handle large-deformation fluid–structure interaction. A systematic parametric study is conducted to quantify the sensitivity of six key design parameters—tubing length, wall thickness, Young’s modulus, yield strength, load amplitude, and internal pressure—on the buckling response. The numerical results establish a clear sensitivity hierarchy: load amplitude > internal pressure > wall thickness > yield strength > length > Young’s modulus. The study further elucidates the underlying mechanisms of stress-wave propagation, energy dissipation, and buckling-mode evolution through a detailed analysis of simulation outputs. An integrated energy–stiffness–pressure mitigation strategy is proposed based on the numerical findings. This work demonstrates the capability of advanced computational modeling to provide both mechanistic understanding and practical design guidance for deep-well tubular systems, highlighting the value of numerical methods in addressing complex engineering challenges where experimental data are limited.

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1 Introduction

With the progressive depletion of conventional oil and gas resources, exploration is shifting to deep and ultra-deep formations characterized by high-temperature, high-pressure (HPHT) conditions [1]. In these extreme environments, downhole tubing strings face not only traditional static loads but also multiphysics-coupled dynamic loads, aggressive corrosion, and elevated-temperature creep [2,3], resulting in strongly nonlinear, time-dependent mechanical responses.

*Correspondence: Yexin Fang (fangyxcnpc@163.com). This is an article distributed under the terms of the Creative Commons BY-NC-SA license

Transient impact loads during tripping and perforation operations are a primary cause of tubing failure. These loads originate from six field scenarios, including perforation shock waves, sucker rod breakage, jarring operations, tripping shocks, high-velocity fluid flow, and water hammer [4–9]. Each scenario generates distinct loading characteristics, complicating the accurate prediction of tubing response.

Numerous studies have addressed dynamic buckling. Canal et al. [10] developed a finite-difference model for perforation shock waves, validated using offshore Brazil data. Sanders et al. [11] revealed the role of tubing mass and stiffness in axial force distribution. Miller et al. [12] studied buckling mode transitions in confined rods. Aarsnes and van de Wouw [13] analyzed distributed drill string dynamics. Blototsky et al. [14] proposed experimental setups for tensile buckling. More recently, Guo [15] examined perforation pressure effects, Ding et al. [16] studied piston force effects, and Benmir and Akroum [17] developed an Euler–Bernoulli-based framework—though nonlinear wellbore factors were omitted.

Despite these contributions, critical gaps remain. Most studies focus on isolated factors without a systematic sensitivity ranking of interacting parameters. The coupled effects of internal pressure—frequently overlooked—are rarely quantified. Few studies address the specific HPHT conditions of ultra-deep wells, where temperature and pressure significantly alter material and fluid behavior. Moreover, no previous work has proposed an integrated mitigation strategy based on comprehensive sensitivity analysis. The underlying buckling mechanisms and mode transition behavior under complex wellbore conditions require more systematic investigation.

This paper presents a numerical investigation of dynamic buckling behavior in deep and ultra-deep well tubing strings under impact loading. A nonlinear finite element model incorporating impact–contact–fluid coupling is developed. The effects of tubing length, wall thickness, Young’s modulus, yield strength, load amplitude, and internal pressure on buckling response are systematically examined. The specific innovations are: (a) systematic quantification of six key parameters with a clear sensitivity hierarchy; (b) identification of the previously overlooked strong stabilizing effect of internal pressure; and (c) proposal of an integrated energy–stiffness–pressure mitigation strategy for optimized design and safety control in extreme conditions.

2 Establishment of Finite Element Model

2.1 Geometric Model

A typical tubing string includes casing, tubing, packers, connectors, valves, and perforating guns. The tubing is the primary load-bearing element. To reduce complexity, the multi-stage assembly is represented as an equivalent isotropic, homogeneous cylindrical tube preserving linear density, bending stiffness, and Poisson’s ratio. Thus, we simplified the original downhole system to a tubing–casing–fluid-coupled model (Fig. 1). Lastly, we set the boundary conditions according to actual downhole conditions: the top packer imposes radial zero-displacement and rotational constraints, we fixed the bottomhole end to simulate the rigid wall constraint, and circumferential contact constraints were imposed by the casing wall.

2.2 Constitutive Models and State Equations

Confidence in the finite element solution fundamentally relies on the algorithmic efficiency of the numerical framework and the consistency of spatiotemporal discretization. In the numerical model of this study, we employed an explicit integration scheme by adopting a mesh density sufficient to achieve the required numerical accuracy, along with an appropriate constitutive relation and precisely

defined material property parameters [18,19]. When an axial impact load acts on the tubing, the resulting response simultaneously involves three distinct scales: microscopic dislocation multiplication, mesoscopic grain rotation, and macroscopic buckling instability, exhibiting strong nonlinearity and path-dependent characteristics.

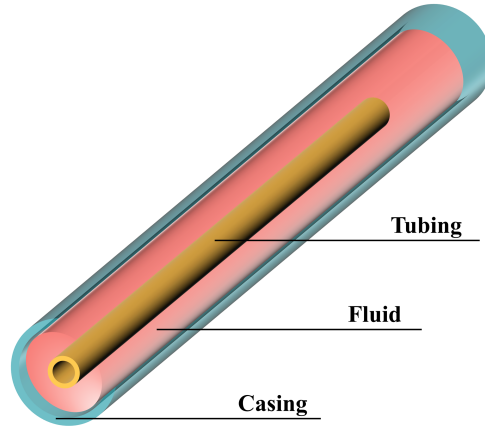


Figure 1: Finite element simulation model of tubing string system

The dynamic plasticity of metal tubing string can be characterized by three experimentally validated rate-dependent mechanisms: (a) As the loading rate increases, the dislocation drag effect becomes significant; (b) The instantaneous stress-strain curve steepens with the strain rate, demonstrating positive strain rate sensitivity; (c) When a step change occurs in the strain rate history, the flow stress requires the re-establishment of a steady state, indicating that the deformation path is modulated by the strain-rate memory effect. The first two mechanisms are collectively defined as the strain-rate effect, which forms the core of dynamics research in solid mechanics [20].

To accurately capture the yield evolution of the tubing and casing under impact loading, the Cowper-Symonds model, calibrated via Split Hopkinson Pressure Bar (SHPB) tests, is adopted in this study. This model is particularly suitable for describing material yield behavior under high-rate loading conditions and is well-suited for dynamic problems involving high strain rates. Within the framework of this model, the yield stress is given by Eqs. (1) and (2) [21].

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

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

where σ_m represents the material's yield stress, which is dependent on the strain rate; ε_m denotes the strain rate of the material; Y_1 and Y_2 are strain rate sensitivity parameters; σ_0 is the reference static yield stress, typically measured under quasi-static loading conditions; E_p is the plastic hardening modulus; and ε_p^{eff} represents the effective plastic strain.

We modelled the annular fluid domain using the MAT_NULL material model, which exhibits no yield strength and simulates materials without shear resistance or inherent stiffness—such as fluids, gases, vacuums, and non-deformable regions. To more accurately capture the state changes of the high-pressure annular fluid under transient impact loading, we adopted the Gruneisen equation of state (Eq. (3)) for describing the annular fluid's physical state.

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

where P_H represents the annular pressure; γ_0 is a constant with a value of 0.35; δ denotes the correction factor; ρ_m is the density of the annular fluid; C_m is the intercept of the wave velocity curve; S_1 , S_2 and S_3 are the slope coefficient of the wave velocity curve; and α_m represents the compressibility coefficient of the annular fluid.

2.3 Model Validation

To establish the reliability of the numerical model, two complementary validation approaches were adopted: a mesh convergence study to ensure numerical consistency (Fig. 2) and a qualitative morphological comparison with a field bending case to assess physical fidelity.

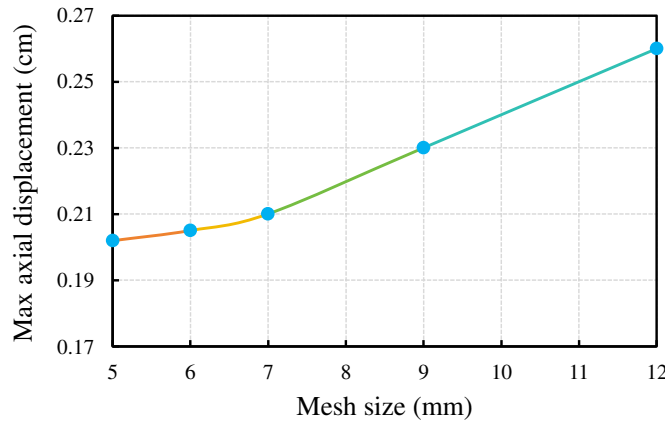


Figure 2: Mesh convergence study results

To validate the discretization strategy and determine the optimal mesh size, a systematic mesh convergence study was performed using five characteristic element sizes: 12, 9, 7, 6, and 5 mm. The maximum axial displacement at the bottom end of the tubing string was selected as the primary convergence criterion because it directly reflects the global dynamic response.

Refining the mesh from 12 to 9 mm and then to 7 mm reduced the displacement from 0.260 to 0.213 cm, but the relative errors remained high (9.62% and 9.36%), indicating that the solution was still sensitive to mesh refinement. When the mesh was refined from 7 to 6 mm, the relative error dropped sharply to 2.35%, which falls below the commonly accepted convergence threshold of 2%–3%. Further refinement from 6 to 5 mm yielded a marginal improvement (error 1.44%) but at a substantially higher computational cost. Therefore, the range of 6–7 mm offers the best trade-off between numerical accuracy (error < 2.5%) and computational efficiency.

3 Finite Element Analysis of Tubing String Dynamic Response

3.1 Geometric Parameter Effects on Impact Buckling

We modelled the tubing and casing using the following specifications, as shown in Table 1.

Table 1: Specifications of tubing and casing

Parameter	Tubing	Casing
Outer diameter	73.02 mm	244.40 mm
Wall thickness	5.51 mm	11.95 mm
Density	7850 kg/m ³	7850 kg/m ³
Young's modulus	210 GPa	210 GPa
Poisson's ratio	0.3	0.25
Yield strength	552 MPa	450 MPa

Tubing lengths of 10, 20, 30, 40, and 50 m were simulated, with axial displacement data extracted from the bottom-end node of each configuration. The displacement responses of tubing strings with varying lengths are presented in Fig. 3.

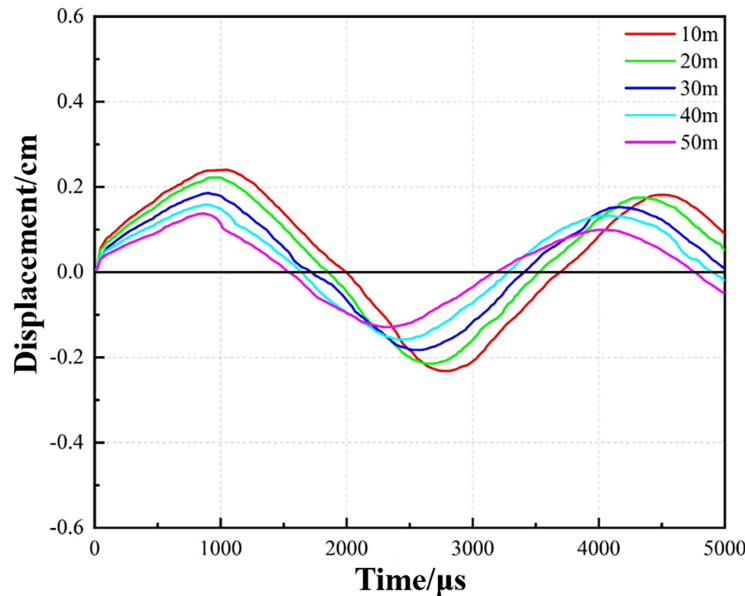


Figure 3: Axial displacement response of the tubing with varying lengths

Based on the wave superposition effect observed from the top-end reflection, the stress wave velocity and one-way travel time to the top can be derived as follows:

$$c = \sqrt{\frac{E_g}{\rho}} = 5172 \text{ m/s} \quad (4)$$

$$t_{10} = \frac{L_g}{c} = 1933 \text{ } \mu\text{s} \quad (5)$$

Within the 5000 μs load duration, the stress wave completed approximately 2.6 one-way trips along the 10 m tubing. As shown in Fig. 3, the motion trajectories of tubing strings with different lengths

follow generally consistent patterns. As the tubing length increased from 10 to 50 m, the maximum axial deformations during the first cycle were 0.472, 0.437, 0.368, 0.316, and 0.266 cm, respectively.

During the initial stage of impact loading, the axial displacement at the tubing's lower end increases rapidly, exhibiting a sharp transient rise. This behavior results from the instantaneous application of the impact load at the tubing's lower end, inducing localized deformation. In shorter tubing strings (e.g., 10 m), the axial displacement exhibits larger amplitude fluctuations over a prolonged duration, as the impact load cannot be rapidly dispersed through the tubing string, leading to pronounced displacement oscillations at the bottom-end node. In contrast, longer tubing (e.g., 50 m) distributes the impact load more effectively through stress superposition and inertial dispersion, resulting in reduced amplitude variations in axial displacement at the bottom end. Over time, the axial displacement gradually stabilizes, with a diminished fluctuation amplitude.

Consequently, shorter tubing strings experience more concentrated impact effects, manifested in larger fluctuations in axial displacement at the bottom end, slower energy attenuation, and prolonged stabilization periods. In comparison, longer tubing strings exhibit superior load distribution, resulting in smoother axial displacement, faster stabilization, and improved buckling resistance under impact. [Table 2](#) summarizes the maximum equivalent stress values corresponding to each tubing length configuration. The rapid increase in subsequent peak equivalent stress exceeding 552 MPa ([Table 2](#)), together with the bottom-end axial displacement, satisfies our buckling initiation criteria.

Table 2: Maximum equivalent stress of the tubing with varying lengths

Tubing length (m)	Maximum equivalent stress (MPa)
10	642.61
20	519.52
30	473.68
40	403.87
50	342.67

Specifically, the peak equivalent stress at the top of the 10 m tubing reaches 642.61 MPa, exceeding the material's yield strength of 552 MPa and indicating that the stress has exceeded the material's elastic limit. Under impact loading, shorter tubing strings lack sufficient length to effectively attenuate the impact energy, resulting in localized stress concentrations and increased buckling susceptibility. Put simply, the limited length impairs effective load dissemination, thereby inducing more significant stress concentrations.

In contrast, the peak equivalent stress at the upper end of the 50 m tubing is 342.67 MPa, remaining well below the material's yield strength. This demonstrates that longer tubing strings effectively distribute impact loads via stress wave dispersion, leading to more uniform stress along the string, and reducing local stress concentrations and buckling susceptibility. Compared to shorter configurations, longer tubing strings exhibit superior buckling resistance and enhanced capacity to withstand impact loading.

We conducted a parametric numerical study to evaluate how wall thickness influences the dynamic response. Models were established for tubing strings with wall thicknesses of 5.51, 7.01, 8.64, 9.96, and 11.18 mm, with all other parameters held constant—including outer diameter, material properties, peak impact load, and load duration—to isolate the influence of wall thickness. We performed

numerical simulations for each configuration, with axial displacement data extracted from the bottom-end node. The displacement responses for different wall thicknesses are shown in Fig. 4.

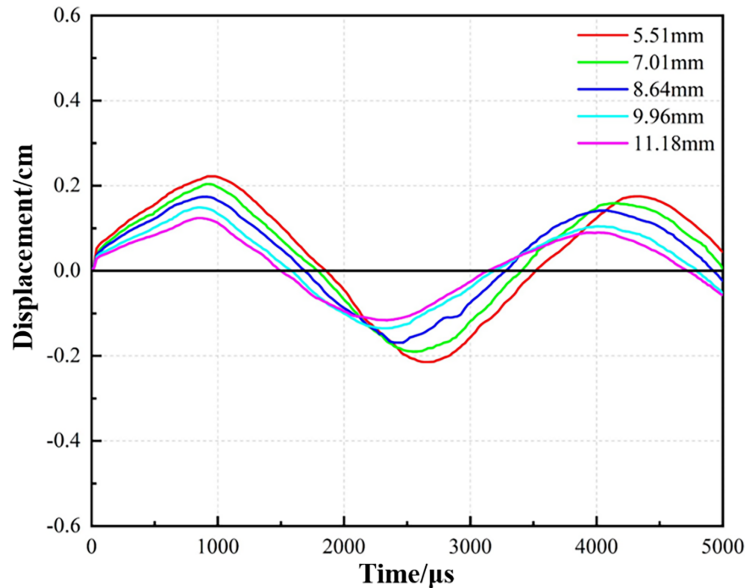


Figure 4: Axial displacement response of the tubing with varying thicknesses

As shown in Fig. 4, the axial displacement behavior varies significantly across different wall thicknesses due to differences in bending stiffness. As the wall thickness increases from 5.51 to 11.18 mm, the maximum axial deformations recorded in the first cycle measure 0.437, 0.393, 0.343, 0.284, and 0.240 cm, respectively.

Upon initial application of the impact load, all tubing configurations exhibit a rapid increase in axial displacement. Under sustained impact loading, the displacement response shows periodic oscillations. As time elapses, impact-induced deformation gradually dissipates, and the axial displacement subsequently stabilizes. Thin-walled tubing strings (e.g., 5.51 mm wall thickness) exhibit a smaller cross-sectional moment of inertia and consequently lower bending stiffness, resulting in less effective dispersion of impact energy. This results in greater amplitude fluctuations in axial displacement at the bottom-end node over an extended period. In contrast, thick-walled tubing strings (e.g., 11.18 mm wall thickness), with their enhanced structural stiffness, distribute impact loads more effectively throughout the tubing string. Such mechanical behavior results in a more gradual increase in axial displacement at the bottom end and reduced overall displacement amplitude.

Increasing wall thickness lowers buckling susceptibility under impact. Thicker walls spread the impulsive load more evenly, mitigate local stress concentrations, and raise overall stability, markedly mitigating the risk of impact-induced buckling. We listed the peak values in Table 3.

Specifically, for a wall thickness of 5.51 mm, the peak equivalent stress reaches 519.52 MPa, approaching the material's yield strength of 552 MPa. A reduced wall thickness is associated with a smaller cross-sectional moment of inertia and lower bending stiffness, which limits effective distribution of the impact load. This reduced structural stiffness produces localized stress concentrations, markedly increasing buckling susceptibility.

Table 3: Maximum equivalent stress of the tubing with varying thicknesses

Tubing length (m)	Maximum equivalent stress (MPa)
5.51	519.52
7.01	461.23
8.64	389.08
9.96	338.71
11.18	292.90

At medium wall thicknesses of 7.01 and 8.64 mm, the peak equivalent stresses under impact loading are 461.23 and 389.08 MPa, respectively. These values are substantially lower than those observed in the thinner-walled configuration (5.51 mm), indicating that increased wall thickness enhances structural stiffness, improves impact load distribution, and mitigates stress concentrations. When we further increased the wall thickness to 9.96 and 11.18 mm, the peak equivalent stress at the top-end node decreased significantly to below 340 MPa. The enhanced stiffness of thick-walled tubing strings facilitates more effective load distribution and facilitates more uniform stress transmission along the length of the tubing string.

In field applications, the optimal wall thickness is crucial for ensuring the stability of tubing strings. Overly thin tubing strings are prone to failure under impact and susceptible to buckling; overly thick tubing strings lead to overdesign, additional costs, and increased energy consumption due to their greater weight. Wall thickness should therefore be optimized according to service conditions, taking into account the expected impact amplitude and duration.

3.2 Material Property Effects on Impact Buckling

Young's modulus of a material characterizes its resistance to elastic deformation. We developed numerical models for tubing strings featuring Young's modulus of 190, 195, 200, 205, and 210 GPa. To isolate the influence of Young's modulus on the buckling response, other parameters—including tubing length, outer diameter, wall thickness, peak impact load, and load duration—were held constant. We conducted simulations for each Young's modulus, with axial displacement data extracted from the bottom-end node of the tubing string. The variation in axial displacement with Young's modulus is shown in Fig. 5.

As shown in Fig. 5, upon initial impact loading, the axial displacement at the lower end of the tubing string increases rapidly, followed by periodic fluctuations. As Young's modulus increases from 190 to 210 GPa, the maximum axial deformations recorded in the first cycle are 0.646, 0.600, 0.552, 0.480, and 0.437 cm, respectively.

Tubing strings with a lower Young's modulus (e.g., 190 GPa) exhibit larger fluctuations in axial displacement at the bottom end over an extended duration and undergo more substantial deformation. In contrast, tubing strings with a higher Young's modulus (e.g., 210 GPa) exhibit reduced axial displacement fluctuation amplitudes and demonstrate greater resistance to impact loading, resulting in superior buckling resistance and stability. Table 4 summarizes the maximum equivalent stress values for different Young's modulus values.

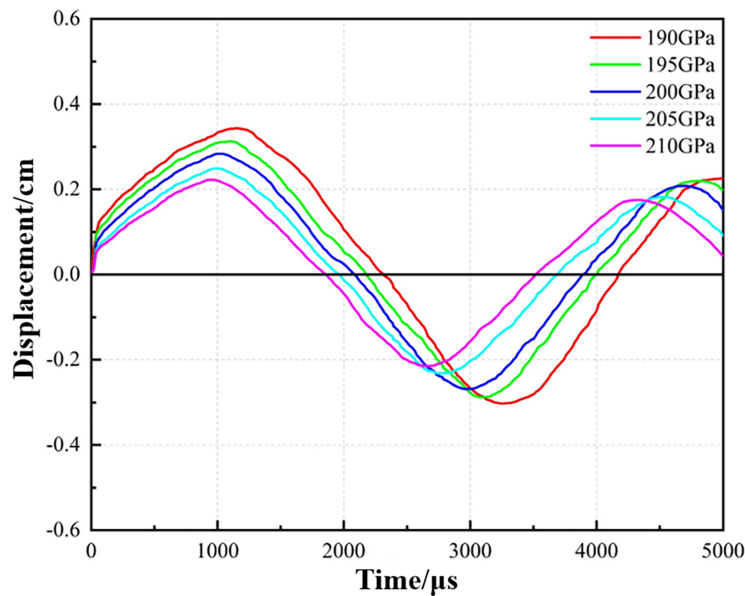


Figure 5: Displacement response of the tubing with varying young's modulus

Table 4: Maximum equivalent stress of the tubing with varying young's modulus

Young's modulus (GPa)	Maximum equivalent stress (MPa)
190	753.25
195	679.12
200	639.22
205	574.14
210	519.52

As illustrated in Table 4, at a Young's modulus of 190 GPa, the peak equivalent stress at the top-end of the tubing string attains 753.25 MPa, surpassing the material's yield strength of 552 MPa. This indicates that the upper end of the tubing string is prone to significant stress concentrations. A lower Young's modulus results in reduced stiffness, rendering the tubing string more susceptible to bending deformation and local buckling under impact loading. For Young's modulus values of 195, 200, and 205 GPa, the peak equivalent stresses at the top-end are 679.12, 639.22, and 574.14 MPa, respectively. Increasing Young's modulus enhances the tubing string's stiffness, reduces deformation at the top section, and consequently lowers buckling susceptibility. For the tubing string with a Young's modulus of 210 GPa, the peak equivalent stress is 519.52 MPa, which remains below the material's yield strength of 552 MPa. This observation suggests that, under the condition of constant density, a higher Young's modulus facilitates faster stress wave propagation (since $c = \sqrt{E/\rho}$), enabling a more rapid global mechanical response to impact loading. This enhanced wave transmission helps distribute and absorb impact energy more effectively, reducing local stress concentrations and favoring the development of global buckling modes over localized failure.

Therefore, in practical engineering applications, increasing Young's modulus helps reduce both the magnitude and duration of axial displacement fluctuations. Tubing strings with a higher Young's modulus respond more rapidly to impact loads, mitigate local stress concentrations, and effectively enhance structural stability in dynamic environments.

Among material properties, yield strength exerts a pronounced influence on the tubing string's response to impact loading. Consequently, we conducted a complementary numerical investigation to assess the impact of yield strength on the tubing string's dynamic buckling response.

We established numerical models for tubing strings with yield strengths of 552, 655, 758, 965, and 1034 MPa. To isolate the influence of yield strength on buckling behavior, other parameters—including tubing length, outer diameter, wall thickness, peak impact load, and load duration—were held constant. We conducted simulations for each yield strength value, with axial displacement data extracted from the bottom-end node of the tubing string. The variation in axial displacement with yield strength is plotted in Fig. 6.

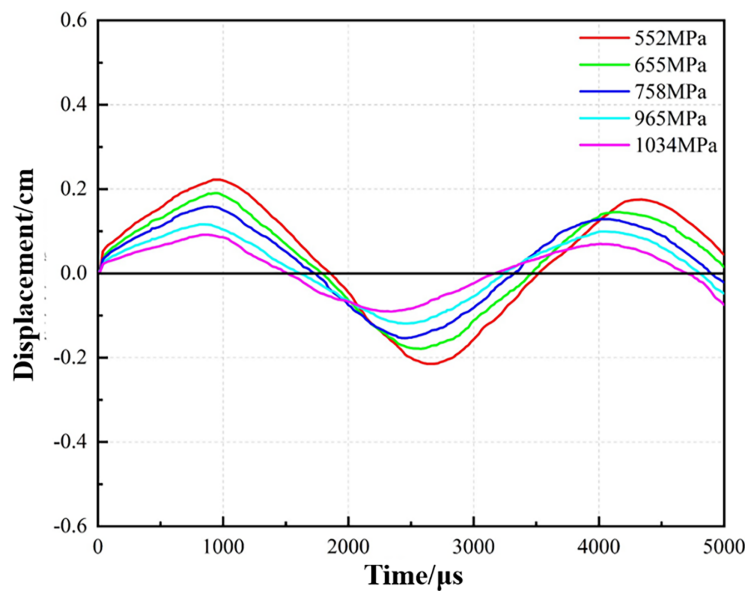


Figure 6: Axial displacement response of the tubing with varying yield strengths

As shown in Fig. 6, upon initial impact loading, the axial displacement at the lower end of the tubing string increases rapidly. As the yield strength increases from 552 to 1034 MPa, the maximum axial deformations recorded in the first cycle are 0.437, 0.369, 0.312, 0.235, and 0.182 cm, respectively.

Tubing strings with lower yield strength (e.g., 552 MPa) exhibit reduced resistance to deformation, as the impact energy cannot be effectively dissipated, thereby inducing more pronounced fluctuations in axial displacement. In contrast, tubing strings with higher yield strength (e.g., 965, 1034 MPa), owing to enhanced material resistance to plastic deformation, demonstrate smaller amplitude and shorter duration of axial displacement variations within a single cycle, thereby reducing both deformation magnitude and buckling susceptibility. Table 5 presents the peak equivalent stress values for different yield strengths.

Table 5: Maximum equivalent stress of the tubing with varying yield strengths

Yield strength (MPa)	Maximum equivalent stress (MPa)
552	519.52
655	450.20
758	398.13
965	336.45
1034	271.65

As depicted in [Table 5](#), at a yield strength of 552 MPa, the peak equivalent stress attains 519.52 MPa—nearing the tubing’s specified yield strength of 552 MPa and signaling a substantial risk of failure under impact loading. For tubing strings with yield strengths of 655 and 758 MPa, the peak equivalent stresses are 450.20 and 398.13 MPa, respectively, indicating a tendency toward localized buckling.

For tubing strings with higher yield strengths of 965 and 1034 MPa, the peak equivalent stresses at the top-end are substantially lower, at 336.45 and 271.65 MPa, respectively, which are well below their corresponding yield strengths. This finding demonstrates that tubing strings with higher yield strength exhibit superior buckling resistance, effectively distributing impact loads and reducing the risk of tubing string failure. As yield strength increases, deformation at the upper end of the tubing string decreases, enabling a more uniform distribution of impact energy. Consequently, tubing strings with higher yield strength demonstrate enhanced buckling resistance, rendering them particularly suitable for resisting dynamic impact loading in demanding engineering applications.

3.3 Loading Condition Effects on Impact Buckling

The amplitude of the impact load directly governs the magnitude of the impact force and the consequent stress state within the tubing string. We established numerical models with impact load amplitudes of 50, 100, 150, 200, and 250 MPa. All other parameters—including tubing length, outer diameter, and inner diameter, and load duration—were held constant to isolate the effect of load amplitude on the buckling response. We conducted simulations for each load amplitude, with axial displacement data extracted from the bottom-end node of the tubing string. The variation in axial displacement with different impact load amplitudes is presented in [Fig. 7](#).

As shown in [Fig. 7](#), upon initial impact loading, the axial displacement of all tubing configurations increases rapidly. Higher load amplitudes induce more pronounced fluctuations in bottom-end displacement. As the impact load amplitude increases from 50 to 250 MPa, the maximum axial deformations recorded in the first cycle are 0.259, 0.347, 0.429, 0.539, and 0.618 cm, respectively.

For lower impact load amplitudes (e.g., 50 MPa), the tubing string can effectively withstand impact loading, with its axial displacement variations being relatively smooth and featuring reduced fluctuation amplitudes as well as shorter durations within each cycle. In contrast, higher load amplitudes (e.g., 250 MPa) induce severe fluctuations in axial displacement at the bottom end and may trigger either local or global buckling. [Table 6](#) summarizes the maximum equivalent stress values for each impact load amplitude.

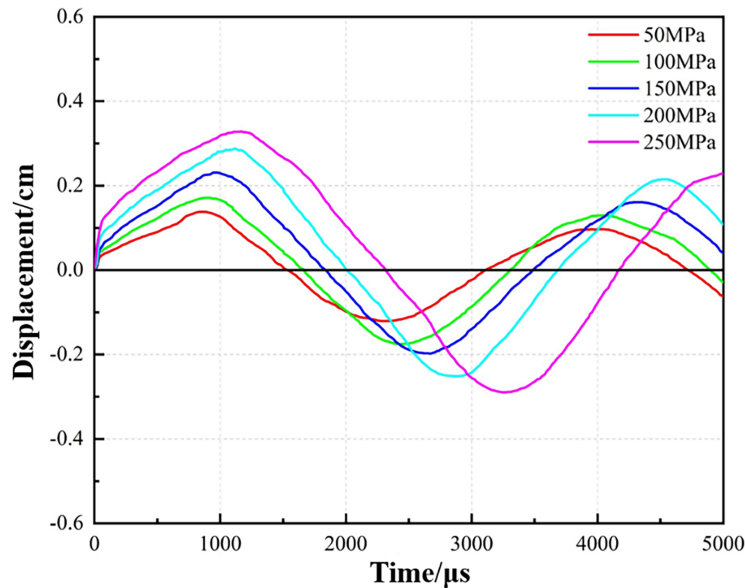


Figure 7: Axial displacement response of the tubing to varying load amplitudes

Table 6: Maximum equivalent stress of the tubing under varying load amplitude

Load amplitudes (MPa)	Maximum equivalent stress (MPa)
50	367.74
100	437.42
150	499.43
200	574.70
250	657.33

As depicted in Table 6, for an impact load amplitude of 50 MPa, the peak equivalent stress at the tubing string's top-end reaches 367.74 MPa, which is relatively low and corresponds to negligible buckling susceptibility, thereby demonstrating the tubing string's capacity to withstand impact loading within the elastic regime. Under load amplitudes of 100 and 150 MPa, the stress at the top increases but remains below the yield strength, signaling a low risk of localized buckling. However, at load amplitudes of 200 and 250 MPa, the peak equivalent stress rises markedly, reaching 657.33 MPa under 250 MPa loading—a value exceeding the material's yield strength of 552 MPa. This finding demonstrates that higher impact load amplitudes subject the tubing string to instantaneous high stress, causing significant stress concentrations at the top-end and thereby elevating buckling susceptibility (both local and global).

Furthermore, forces acting upon the tubing string comprise not only external loads but also internal pressures induced by the enclosed fluid. To investigate the effect of internal pressure, we established numerical models with internal pressures of 0, 15, 25, 35, and 45 MPa. The 0 MPa case corresponds to air-filled tubing strings, while the other pressure cases simulate various fluid-filled conditions. All other tubing string parameters—including length, geometric dimensions, and loading conditions—were held constant to isolate the influence of internal pressure on the buckling response.

We conducted simulations for each internal pressure condition, with axial displacement data extracted from the bottom-end node of the tubing string. The resulting variations in axial displacement with different internal pressures are presented in Fig. 8. Table 7 summarizes the maximum equivalent stress values of the tubing string for different internal pressure conditions.

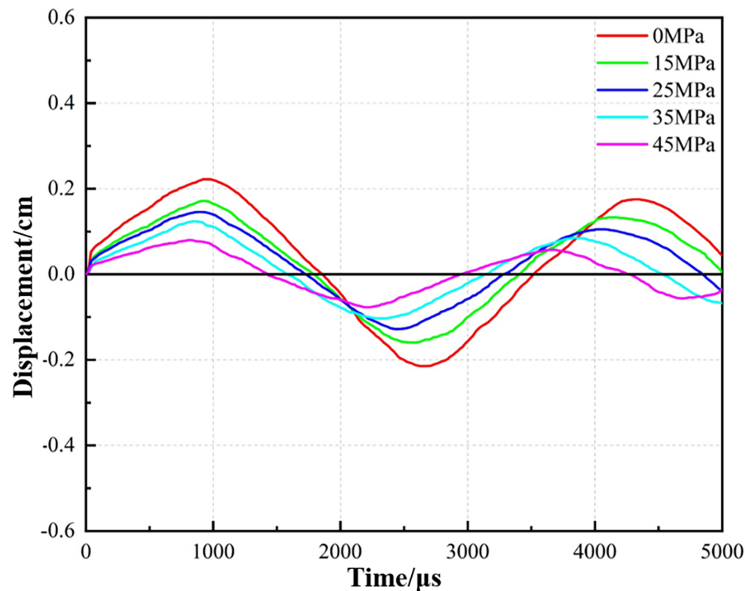


Figure 8: Axial displacement response of the tubing under varying internal pressures

Table 7: Maximum equivalent stress of the tubing under varying load amplitude

Internal pressures (MPa)	Maximum equivalent stress (MPa)
0	519.52
15	403.51
25	357.95
35	296.77
45	254.19

As illustrated in Table 7, the maximum equivalent stress in the tubing string decreases with increasing internal pressure, confirming that internal pressure exerts a significant stabilizing effect on the tubing string's stiffness and structural stability. Under atmospheric conditions (0 MPa internal pressure), the peak equivalent stress at the top-end reaches 519.52 MPa. In this case, the lack of internal support subjects the tubing string to primary exposure to external impact, resulting in substantial deformation and pronounced stress concentrations.

As internal pressure increases under fluid-filled conditions, the circumferential confinement exerted by the internal fluid imposes a confining effect upon the tubing string's wall. This pressurized environment effectively enhances the tubing string's structural stiffness, thereby reducing both deformation amplitude and equivalent stress. At internal pressures of 35 and 45 MPa, the peak equivalent stresses at the top-end decrease to 296.77 and 254.19 MPa, respectively. Adequate fluid support

significantly mitigates the impact effects, markedly enhancing buckling resistance and resulting in stress levels far lower than those under low-pressure conditions.

In summary, under transient impact loading conditions: (a) Load amplitude exerts the most pronounced influence on buckling risk and should be prioritized in design control measures; (b) Greater wall thickness significantly enhances bending stiffness and buckling resistance; (c) Increased tubing length facilitates the dispersion of impact energy through stress wave propagation, reducing buckling susceptibility; (d) Internal pressure suppresses buckling via circumferential stress and wall confinement; (e) Higher yield strength significantly delays plastic hinge formation and propagation while reducing permanent deformation; (f) Young's modulus exhibits a relatively minor influence on the dynamic buckling response compared to other parameters.

4 Discussions

Among all factors, impact load amplitude is the dominant control variable. Increasing it from 50 to 250 MPa raised peak equivalent stress from 367.74 to 657.33 MPa, exceeding yield strength and triggering irreversible plasticity. Geometric parameters act primarily as mitigating factors. Increasing length from 10 to 50 m reduced max deformation by 43.6% and peak stress by 46.7% via inertial dispersion and wave attenuation, consistent with field observations by Sanders et al. [11]. Similarly, increasing wall thickness from 5.51 to 11.18 mm enhanced bending stiffness and reduced deformation and stress by over 40%.

Material properties play distinct roles. Yield strength significantly affects post-buckling: 1034 MPa material showed 58.3% less deformation and 47.7% lower peak stress than 552 MPa steel, due to delayed plastic hinge formation. Young's modulus had a weaker effect: increasing from 190 to 210 GPa reduced deformation by 32.4% and stress by 31%, indicating elastic stiffness modulates wave speed but not energy dissipation capacity.

Internal pressure, often overlooked, shows strong stabilization: raising it from 0 to 45 MPa reduced deformation by 64.1% and stress by 51.1%. This effect relies on assumptions: uniform quasi-static pressure, incompressible fluid, ideal seal, vertical well, and short loading duration. Validity may decrease with poor end constraints, deviated wells, or longer loading. Field application also faces operational limits (pumping capacity, wellhead pressure).

The sensitivity ranking is conditional on the investigated ranges (length 10–50 m, thickness 5.51–11.18 mm, Young's modulus 190–210 GPa, yield 552–1034 MPa, pressure 0–45 MPa, amplitude 50–250 MPa) and idealized boundaries (fixed top, rigid bottom). Readers should apply cautiously and conduct case-specific checks. Compared with Canal et al. [10] and Miller et al. [12], our model integrates rate sensitivity, fluid coupling, and transient impact. It extends Aarsnes and van de Wouw [13] by including contact nonlinearity and fluid confinement.

Limitations: the equivalent isotropic model omits joint-induced stiffness variations—suitable for global buckling but not local fatigue. The proposed hierarchy and mitigation strategy are numerical insights, not yet experimentally validated. Some trends align with field experience, but quantitative confirmation is needed. Boundary conditions are idealized; real packers and bottom supports are nonlinear/elastic. Nonetheless, they provide a conservative baseline.

Overall, this study quantifies coupled effects of impact, geometry, material, and pressure, offering a practical energy–stiffness–pressure framework for improving tubing reliability in ultra-deep wells.

5 Conclusion

This study systematically investigates the dynamic buckling behavior of tubing strings in ultra-deep wells under impact loading. The main conclusions are:

- a. Geometric parameters: Longer tubing strings (10 to 50 m) promote axial dispersion and dissipation of impact energy, reducing peak deformation by 43.6% and stress by 46.7%. Greater wall thickness (5.51 to 11.18 mm) enhances bending stiffness and suppresses local buckling modes.
- b. Material properties: Yield strength is the dominant material factor; increasing it from 552 to 1034 MPa reduces deformation by 58.3% and peak stress by 47.7% by delaying plastic hinge formation. Young's modulus has a minor effect (deformation reduction 32.4%), confirming that elastic stiffness modulates wave speed but not energy dissipation.
- c. Loading conditions: Impact amplitude is the most influential parameter; a fivefold increase raises stress from 367 to 657 MPa (exceeding yield), driving severe buckling. Internal pressure (0 to 45 MPa) provides significant stabilization (deformation reduction 64.1%, stress reduction 51.1%) via circumferential confinement and hoop-axial energy conversion.

The sensitivity hierarchy is: load amplitude > internal pressure > wall thickness > yield strength > length > Young's modulus, offering clear design prioritization.

Through numerical investigation, this study elucidates the mechanisms governing dynamic buckling and proposes an integrated energy–stiffness–pressure mitigation strategy. The developed model and findings provide a theoretical and practical basis for optimizing design, assessing structural integrity, and formulating risk mitigation strategies for ultra-deep well tubing strings under impact loads.

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