

Offshore Pipeline Stress and Stability Analysis in the Gulf of Guinea

Eze Evidence Ojie, Morrison Inegiyemiema, Samson Nitonye

Department of Marine and Offshore Engineering, Faculty of Engineering, Rivers State University, Port Harcourt.

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ABSTRACT

The Gulf of Guinea, a critical hub for global offshore hydrocarbon production, presents unique challenges for subsea pipeline integrity due to its complex environmental conditions, including deep waters, soft seabed soils, and dynamic metocean forces. This research conducts a comprehensive stress and stability analysis of an offshore pipeline in this region to ensure safe and reliable operation. The study employs an integrated methodology, combining rigorous analytical models prescribed by international design codes (DNV-OS-F101, DNV-RP-F109, ASME B31.3) with advanced three-dimensional Finite Element Analysis (FEA) in ANSYS Workbench. A representative pipeline configuration, featuring API 5L X60 steel material, was analyzed under region-specific operational and environmental loads, including internal pressure, thermal gradients, hydrodynamic forces from waves and currents, and geotechnical interactions. The investigation systematically addressed the influence of pipeline geometry, particularly the Diameter-to-Thickness (D/t) ratio, and assessed the impact of the Gulf of Guinea's specific environmental parameters. Key findings demonstrate that the pipeline design is stable under a 100-year extreme storm event, with a safety factor of 2.05 for on-bottom stability. The maximum von-Mises stress of 278 MPa remains well within the allowable limit for the material, validating the selected wall thickness. The study reveals a non-linear, critical relationship between the D/t ratio and induced stress, defining a safe design window for local conditions. Furthermore, hydrodynamic analysis identifies current velocity as the most sensitive environmental parameter governing pipeline stability. Fatigue assessment, based on cumulative damage (Miner's Rule), indicates a damage ratio of 0.67 over a 25-year design life, confirming adequate fatigue resistance. This research concludes that the integrated analytical-numerical approach provides a robust and validated framework for pipeline integrity assessment in the Gulf of Guinea. The main contributions to knowledge include the development of this region-specific methodology, the quantification of critical parameter interactions, and the provision of validated design benchmarks. The study recommends the adoption of such integrated analysis in industry practice, enhanced data acquisition for regional inputs, and the development of focused integrity management programs to mitigate long-term risks, thereby supporting safer and more sustainable offshore energy infrastructure in the region.

Keywords: Offshore Pipeline Integrity, Finite Element Analysis (FEA), On-Bottom Stability, Stress Analysis, Gulf of Guinea

INTRODUCTION

The global demand for hydrocarbon resources has driven the expansion of offshore exploration and production activities into increasingly challenging environments. Offshore pipelines serve as the primary means of transporting extracted hydrocarbons from subsea wellheads to processing facilities and onward to onshore distribution networks (Bai & Bai, 2005). These subsea infrastructures represent substantial capital investments and are expected to operate reliably over extended design lives, typically ranging from 20 to 30 years, under severe environmental conditions. The Gulf of Guinea, situated along the western coast of Africa, has emerged as a significant hub for offshore hydrocarbon production, attracting substantial investment from international oil and gas companies. This region, however, presents unique challenges for subsea pipeline design and operation due to its complex environmental conditions, including deep waters, soft seabed soils, and dynamic metocean forces (Nwoka *et al.*, 2026, Ekwere *et al.*, 2021).

Pipeline stress analysis is a critical engineering discipline used to evaluate how piping systems respond to internal pressure, temperature changes, fluid properties, and support configurations. (Adumene, *et al*, 2021, Nrior *et al.*, 2026) It categorizes stresses into hoop, axial, bending, torsional, and fatigue types, each arising from different operational and environmental loads. For offshore pipelines, design follows limit state principles to prevent failure modes such as buckling, with particular attention to pipe-soil interaction, embedment depth, and on-bottom stability. Installation dynamics, vessel motions, and touchdown point disturbances further complicate loading conditions. Historical failures have underscored the need for rigorous analysis, often supported by software like CAESAR II or ANSYS Workbench, to ensure safe and reliable operation across diverse seabed and environmental conditions. In regions like the Gulf of Guinea, pipeline design must account for extreme wave and current forces from 100-year storm events, variable soil friction coefficients, and sediment transport that affect stability and scour potential. Water depth, marine growth, and site-specific storm surge characteristics also influence stress outcomes, making localized assessments essential. (Oliveira & Rizkalla, 2024) As exploration moves into deeper waters worldwide, particularly in Brazil, India, China, and West Africa, the demand for robust pipelines grows, alongside the need for accurate modeling of complex interactions between pipelines and soft or stiff clay seabeds. (Diaz, 2026, Wen *et al.*, 2023) Comprehensive stress analysis remains vital to minimizing environmental and economic risks, ensuring that pipelines withstand both operational demands and extreme natural forces throughout their service life. (Njuguna *et al* 2023) The integration of analytical models prescribed by international design codes with advanced finite element analysis provides a comprehensive approach to pipeline stress and stability assessment. Design codes including DNV-OS-F101, DNV-RP-F109, and ASME B31.3 establish requirements for pipeline design, material selection, and stress evaluation. These standards define allowable stress limits for sustained loads, displacement stresses, and occasional stresses, providing the framework for design verification. The analytical models, grounded in these codes, offer preliminary assessments of pipeline behavior under pressure, temperature, and environmental loads. Finite element analysis extends these capabilities by enabling detailed three-dimensional modeling of pipeline response, including stress distribution, deformation patterns, and buckling behavior (Tadama & Tadama, 2024, Nwoka *et al.*, 2026).

The significance of comprehensive pipeline stress and stability analysis extends across multiple engineering disciplines, including offshore engineering, subsea engineering, pipeline engineering, and materials engineering. Understanding the impact of pipe geometry parameters, including thickness, diameter, and length, on induced stresses is fundamental to effective offshore pipeline design. This knowledge has direct implications for operational conditions, pipeline integrity management, and environmental protection. The research serves as a valuable resource for educational institutions and students interested in pipeline engineering, contributing to the development of expertise in this critical field. The findings of such analysis have direct consequences for day-to-day pipeline operations and maintenance activities, supporting the development of effective integrity management programs.

MATERIALS AND METHODS

This section details the integrated computational methodology employed to achieve the research aim of performing a high-fidelity stress and stability analysis of an offshore pipeline in the Gulf of Guinea. The approach combines code-compliant analytical modeling with sophisticated Finite Element Analysis (FEA) to provide a holistic, industry-standard assessment.

Materials and Input Parameters

The analysis is based on a representative pipeline configuration and environmental data specific to the Gulf of Guinea, ensuring regional relevance. The tables below summarize the key input parameters derived from project specifications and regional data (Ekwere *et al.*, 2021).

Table 1: Pipeline Geometry and Configuration (Ekwere *et al.*, 2021)

Parameter	Value	Unit	Source/Standard
Outer Diameter (D_o)	0.508	M	Project Specification
Internal Diameter (D_i)	0.4762	M	Project Specification

Nominal Wall Thickness (t_{nom})	0.01588	M	Project Specification
Corrosion Coating Thickness	0.0065	M	Project Specification
Concrete Coating Thickness	0.100	M	Project Specification
Effective Outer Diameter (D_{eff})	0.708	M	DNV-RP-F105
Analysis Length (Free Span)	30	M	Based on Preliminary Screening

Table 2: Material Properties (Ekwere *et al.*, 2021)

Parameter	Value	Unit	Source/Standard
Young's Modulus (E)	2.07×10^{11}	Pa	API 5L
Poisson's Ratio (ν)	0.30	-	API 5L
Yield Strength (σ_y)	450×10^6	Pa	API 5L
Density of Steel (ρ_{steel})	7850	kg/m ³	Standard Value
Density of Concrete Coating (ρ_{conc})	3000	kg/m ³	Project Specification
Density of Corrosion Coating (ρ_{corr})	1300	kg/m ³	Project Specification
Coefficient of Thermal Expansion (α)	1.16×10^{-5}	/°C	API 5L

Table 3: Environmental and Operational Loads (Ekwere *et al.*, 2021)

Parameter	Value	Unit	Source/Standard
Water Depth (h)	1000	m	Gulf of Guinea Data
Internal Pressure (P_I)	12	MPa	Operational Data
External Pressure (P_O)	10.06	MPa	DNV-OS-F101
Temperature Differential (ΔT)	21	°C	Operational Data
Current Velocity (V_C)	0.29	m/s	Gulf of Guinea Data
Significant Wave Height (H_S)	3.45	m	Gulf of Guinea Data (100-year)
Peak Wave Period (T_P)	17	s	Gulf of Guinea Data (100-year)
Submerged Weight (W_S)	3663.14	N/m	DNV-OS-F101
Soil Friction Coefficient (μ)	0.6	-	Geotechnical Data

Table 4: Long-Term Current Velocity Distribution (m/s) (Ekwere *et al.*, 2021)

Water depth (m)	1yr	100yr
23	0.74	1.27
53	0.42	0.74
108	0.39	0.69
200	0.33	0.55
500	0.3	0.5
1000	0.18	0.29

Table 5: Wave Data (Ekwere *et al.*, 2021)

Nomenclature	Calculated Parameters	
	1year	100years
Significant wave height (m)	2.74	3.45
Peak period (sec)	14.4	17
Wave number(/m)	0.019	0.014
Wave Velocity (m/s)	2.229×10^{-9}	5.715×10^{-7}

METHODS

The methodology integrates rigorous code-compliant analytical checks with high-fidelity Finite Element modelling, ensuring a comprehensive and validated analysis.

Advanced Analytical Models and Code Compliance

Preliminary analytical models, governed by international standards, are used for scoping, load definition, and validation of FEA results.

Stress Analysis

Stress due to pressure difference

Stress due to pressure difference in an offshore pipeline is the internal mechanical stress induced in the pipe wall by the difference between internal fluid pressure and the external seawater pressure, which governs whether the pipe expands or collapses. (xu et al, 2024)

$$\sigma_a^p = \frac{(P_i - P_o)D_m}{2t_{nom}} \quad (1)$$

$$P_o = \rho_{sw}gh \quad (2)$$

$$D_m = \frac{D_i + D_o}{2} \quad (3)$$

Stress due to temperature

Stress due to temperature in an offshore pipeline is the stress induced in the pipe material when temperature changes cause expansion or contraction that is restrained, leading to internal forces within the pipeline.

$$\sigma_a^t = E\alpha(T_2 - T_1) \quad (4)$$

Stress due to curvature of the pipeline

Stress due to curvature in an offshore pipeline is the bending stress induced in the pipe when it is forced into a curved shape, causing tension on one side and compression on the other.

$$\sigma_a^b = Ey \frac{d^2y}{dx^2} \quad (5)$$

Stress due to Pre-tension

Stress due to pre-tension in an offshore pipeline is the axial tensile stress induced in the pipe by an applied pulling force during installation to control its shape and stability.

$$\sigma_a^x = 25\% \sigma_a^{Pre} \quad (6)$$

$$\sigma_a^{Pre} = EA\alpha\Delta T \quad (7) \quad A_c = \frac{\pi}{4}(D_o^2 - D_i^2) \quad (8)$$

End cap reduced axial stress

End-cap reduced axial stress in an offshore pipeline is the effective axial stress in the pipe wall after considering the balancing effect of internal pressure acting on the pipe ends, which reduces the net axial stress compared to simple axial loading.

$$\sigma_a^e = \frac{pd}{4t} - \frac{pd}{8t} \quad (9)$$

$$\sigma_a^e = \frac{pd}{8t} \quad (10)$$

$$\sigma_a^e = \frac{\Delta Pr_m}{2tr_m} \quad (11a)$$

$$\sigma_a^e = \frac{P_i r_m - P_o r_m}{2tr_m} \quad (11b)$$

$$\sigma_a^e = \frac{(P_i - P_o)r_m}{2tr_m} \quad (11c)$$

$$\sigma_a^e = \frac{P_i - P_o}{2t} \quad (11d)$$

Longitudinal stress

Longitudinal stress in an offshore pipeline is the stress acting along the pipe's axis, caused by pressure, temperature changes, bending, and external forces.

$$\sigma_L = \sigma_a^p + \sigma_a^t + \sigma_a^b + \sigma_a^{Pre} + \sigma_a^e \quad (12)$$

Equivalent (Von mises) stress

Equivalent (Von Mises) stress in an offshore pipeline is a combined stress value that represents the effect of multiple stresses acting together, used to assess whether the pipeline material will yield.

$$\sigma_{eq} = \sqrt{\sigma_h^2 + \sigma_L^2 - \sigma_h \sigma_L} \quad (13)$$

Maximum allowable bending stress

Maximum allowable bending stress in an offshore pipeline is the maximum safe limit of bending stress that the pipe can safely withstand without yielding or failing, determined using material strength and design safety factors.

$$\sigma_b = \min[|\sigma_{L1} - \sigma_{Lp}|, |\sigma_{L2} - \sigma_{Lp}|] \quad (14)$$

$$\sigma_{L1,2} = \frac{\sigma_h \pm \sqrt{(-\sigma_h^2) - 4(\sigma_h^2 - \sigma_{eqv}^2)}}{2} \quad (15a)$$

$$\sigma_{L1} = \frac{\sigma_h + \sqrt{(-\sigma_h^2) - 4(\sigma_h^2 - \sigma_{eqv}^2)}}{2} \quad (15b)$$

$$\sigma_{L2} = \frac{\sigma_h - \sqrt{(-\sigma_h^2) - 4(\sigma_h^2 - \sigma_{eqv}^2)}}{2} \quad (15c)$$

Stress Categorization and Acceptance Criteria (ASME B31.3 & DNV-OS-F101)

These are pass/fail checks against design codes.

Sustained Stress (Primary): Weight, pressure, and gravity loads must stay within the basic allowable stress, scaled by a usage factor.

$$S_I \leq S_h \times \eta_1 \quad (16)$$

Displacement Stress Range (Secondary): Thermal expansion stresses are less dangerous because they are self-limiting; hence a higher allowable.

$$S_e \leq S_a = f(1.25S_c + 0.25S_h) \times \eta_2 \quad (17)$$

Occasional Stress:

Occasional stress in an offshore pipeline is the temporary stress caused by rare or short-term loading events, allowed to exceed normal operating stress limits but still within safe design limits.

$$S_{occ} \leq 1.33S_h \times \eta_3 \quad (18)$$

Pressure Design Thickness Verification (ASME B31.3 Section 304)

A design-code formula that calculates the minimum wall thickness needed to safely contain the internal pressure. Every variable represents a safety factor or material property. The result is compared against the actual nominal wall thickness (15.88 mm) to verify adequacy.

$$t = \frac{P \cdot D}{2(S \cdot E \cdot W + P \cdot Y)} + c \quad (19)$$

Local Buckling and Collapse under External Pressure (DNV-OS-F101)

This is a nonlinear interaction equation governing collapse under external pressure the most critical failure mode for deepwater pipelines. It accounts for both elastic buckling (too thin a wall snaps elastically) and plastic collapse (material yields), with initial ovality (f_0) as an imperfection that worsens collapse resistance. p_c is solved iteratively.

$$\left[\left(\frac{p_c}{p_{el}} - \frac{f_0 \cdot \frac{D}{t}}{t} \right) \cdot \left(1 - \frac{p_c}{p_y} \right)^2 \right] = \frac{p_c}{p_{el}} \cdot \frac{f_0 \cdot \frac{D}{t}}{t} \quad (20)$$

Geotechnical Scour Analysis (DNV-RP-F105)

Shields Parameter:

The Shields parameter is the dimensionless ratio of destabilizing drag on a sediment grain (τ_b) to its stabilising submerged weight. If θ exceeds a critical value θ_c , sediment starts moving — meaning the seabed under the pipeline can scour away, creating a free span.

$$\theta = \frac{\tau_b}{(\rho_{sd} - \rho_{sw}) \cdot g \cdot d_{sd}}; \quad (21)$$

$$\tau_b = \frac{1}{2} \cdot f_w \cdot \rho_{sw} \cdot U_w^2 \quad (22)$$

Stability Analysis

Area

This formula calculates the cross-sectional area of a hollow cylindrical pipe wall. Because a pipeline has a circular cross-section with thin walls, the total material area can be approximated as the circumference of the mid-wall circle ($\pi \times D_m$) multiplied by the wall thickness (t). This is the thin-wall approximation, valid when the wall thickness is small relative to the pipe diameter.

The area A is fundamental to computing axial stress, compressive resistance, and the stiffness of the pipeline. A larger wall thickness or mean diameter directly increases the cross-sectional area, which improves the pipe's ability to carry axial loads and resist buckling.

$$A = \pi D_m \times t \quad (23)$$

Mean Diameter

The mean diameter is simply the arithmetic average of the internal (bore) diameter and the external (outer) diameter. It represents the diameter at the mid-thickness of the pipe wall and is the reference dimension used in the thin-wall approximation for area and other stress calculations.

Using the mean diameter rather than either extreme ensures that stress and area calculations are accurate for pipes of any wall thickness, eliminating the bias that would arise from using only the inner or outer diameter.

$$D_m = \frac{D_i + D_o}{2} \quad (24)$$

Second Moment of Area (Inertia)

The second moment of area (also called the area moment of inertia) is a geometric property that quantifies how the cross-sectional area is distributed relative to a neutral axis. It governs the pipe's resistance to bending. The formula subtracts the contribution of the hollow interior (r_i^4) from a solid circular section (r_o^4), giving the true inertia of the hollow pipe.

A high value of I means the pipe is stiffer in bending and more resistant to lateral buckling. Engineers use I directly in buckling, deflection, and stability calculations. A thicker wall or larger outer radius dramatically increases I because of the fourth-power relationship.

$$I = \frac{\pi}{4} (r_o^4 - r_i^4) \quad (25)$$

Characteristic Wavelength

The fourth-root relationship shows that the wavelength is relatively insensitive to changes in individual parameters; This equation determines the natural wavelength of lateral buckling — the arc length over which the pipeline would naturally deform sideways if it were to buckle on the seabed. It is derived from classical beam-on-elastic-foundation theory, where the pipe is modelled as an elastic beam resting on a frictional soil foundation. Very large changes in soil friction, stiffness, or weight only moderately affect L . A longer wavelength means the buckle is spread over a greater distance, which reduces peak stresses. This length is then used in the buckling parameter (Eq. 3.28) to assess whether the pipeline is stable.

$$L = \left(72 \frac{\mu E I}{W_s} \right)^{\frac{1}{4}} \quad (26)$$

Effective compressive axial force

This is one of the most important equations in pipeline design. It calculates the total effective compressive force that builds up inside an operating pipeline due to two distinct mechanisms:

Thermal expansion: When a pipeline heats up during operation (hot oil or gas), the steel wants to expand. If the pipe is constrained (e.g., buried or heavily weighted), it cannot expand freely, so internal compressive stress builds up. The force is proportional to the material's stiffness (E), area (A), thermal expansion coefficient (α), and the temperature rise (ΔT).

Pressure end-cap effect: Internal operating pressure acts on the cross-sectional area of the pipe bore, generating an axial force that pushes outward at any closed end or bend. The Poisson correction $(1-2\nu)$ accounts for the coupling between hoop (circumferential) stress and axial stress in the pipe wall.

The combined force P is the driving force for upheaval and lateral buckling. If P exceeds the restraining forces from soil and pipe stiffness, the pipe will buckle.

$$P = EA\alpha\Delta T + (1 - 2\nu) \frac{\pi}{4} D_m^2 \Delta P \quad (27)$$

Buckling (Stability) Parameter

The buckling parameter Φ_L is a dimensionless number that combines the characteristic wavelength L (from Eq. 3.26) with the ratio of compressive axial force P to pipe bending stiffness EI . It is the primary indicator of how close the pipeline is to a buckling instability.

Physically, Φ_L represents the number of 'half-wavelengths' of buckling that fit within the buckle length. A higher value of Φ_L indicates greater compressive loading relative to pipe stiffness — meaning higher risk of buckling. The stability factor Φ_W (Eq. 3.29) is then derived from Φ_L to determine allowable loads.

$$\Phi_L = L \left(\frac{P}{EI} \right)^{\frac{1}{2}} \quad (28)$$

Stability Factor (Φ_W)

This empirical equation converts the buckling parameter Φ_L into a dimensionless stability factor Φ_W , which is then used to compute the allowable axial load (Eq. 3.30). It is derived from theoretical and experimental lateral buckling analysis of pipelines resting on soil.

The two terms capture different regimes of buckling behavior. The negative Φ_L^4 term ensures that as Φ_L increases (more compressive force), Φ_W decreases, meaning a lower fraction of the theoretical load is allowable. At very small Φ_L values, Φ_W can be large, indicating relatively stable conditions.

$$\Phi_W = \frac{5.68}{\Phi_L^2} - \frac{88.35}{\Phi_L^4} \quad (29)$$

Allowable Axial Load

This equation defines the maximum axial load that the pipeline can sustain without buckling laterally, given a specified maximum allowable lateral deflection δ . It combines the stability factor Φ_W with the square of the compressive force P and the pipe's bending stiffness EI .

The allowable load W_a increases with the stability factor and deflection allowance, but decreases with pipeline stiffness. Stiffer pipes (high EI) require a larger force to produce the same deflection. If the actual axial force P exceeds what is permissible for a given deflection, the design must be revised by increasing wall thickness, adding rock dump, or modifying the route.

$$W_a = \Phi_W \frac{\delta P^2}{EI} \quad (30)$$

Uplift Resistance Condition

This expression represents the total downward resistance per unit length available to prevent the pipeline from floating upward (upheaval buckling) through its backfill cover. For the pipeline to remain stable against uplift, this combined resistance must be greater than or equal to the upward compressive force generated by thermal and pressure loading.

The two components act together: W_s is the net downward gravitational force from the pipe's own submerged weight, and q is the additional restraint provided by the soil cover above the pipe. Buried pipelines rely on both mechanisms, and the backfill material properties (density and depth) are therefore critical design parameters.

$$q + W_s \quad (31)$$

Uplift Resistance per Unit Length from Backfill

This formula quantifies how much soil cover above the pipeline resists its upward movement. The first part ($H \cdot D_o$) gives the basic soil prism weight acting on the top of the pipe. The bracketed correction factor $\left(1 + f \cdot \frac{H}{D_o}\right)$ accounts for the additional shear resistance mobilised along the failure planes in the soil as the pipe tries to push upward wider or deeper cover mobilises more soil shear.

The uplift coefficient f depends on soil density ($f = 0.5$ for dense soil, lower for loose). The formula shows that deeper cover and denser backfill substantially increase uplift resistance. This is used directly to size the minimum burial depth required to prevent upheaval buckling.

$$q = HD_o \times \left(1 + \frac{fH}{D_o}\right) \times \rho_s \quad (32)$$

Seabed Friction Resistance

This is the classical Coulomb friction equation applied to an on-bottom pipeline. It states that the maximum horizontal frictional resistance force per unit length is simply the product of the soil friction coefficient and the net downward (submerged) weight of the pipe. The friction force acts to oppose any lateral or longitudinal sliding of the pipe across the seabed.

The soil friction coefficient μ varies widely: sandy soils tend to give higher μ values (0.6–0.8), while soft clays give lower values (0.2–0.4). A higher submerged weight (through concrete coating or heavy wall) increases F_R and therefore improves on-bottom stability without the need for external restraints such as rock dumping.

$$F_R = \mu \times W_s \quad (33)$$

Stability Factor

The stability factor is a dimensionless ratio comparing the available frictional resistance to the applied hydrodynamic destabilizing force. Two separate ratios are defined: one comparing friction to drag force (F_D), which tends to push the pipe horizontally, and one comparing friction to lift force (F_L), which reduces the effective weight pressing the pipe onto the seabed.

For the pipeline to be considered on-bottom stable, both stability factors must be greater than or equal to 1.0 (or the required safety factor, typically 1.1 per DNV). A factor below 1.0 indicates that the hydrodynamic forces exceed friction, and the pipe will move — requiring redesign by adding weight or changing the route.

$$\text{Stability Factor} = \frac{F_R}{F_D} = \frac{F_R}{F_L} \quad (34)$$

Seabed Interaction Analysis

Shields Parameter for Sediment Transport

The Shields parameter (θ) is the standard dimensionless measure used in sediment transport theory to determine whether the seabed material will begin to move under hydrodynamic loading. It compares the destabilizing bed shear stress (τ_b) to the stabilizing submerged weight of a sediment grain.

Equation 36 first calculates the bed shear stress τ_b , which is the tangential force per unit area exerted by wave-driven flow on the seabed. It depends on the wave friction factor f_w and the square of the near-bed wave velocity U_w the quadratic dependence means that doubling the flow velocity quadruples the shear stress.

When θ exceeds the critical Shields parameter θ_c (typically 0.04–0.06 for sand), sediment grains are mobilised and transported, leading to scour around the pipeline. This is used directly in the scour depth calculation (Eq. 37).

$$\theta = \frac{\tau_b}{(\rho_{sd} - \rho_{sw}) \times g \times d_{sd}} \quad (35)$$

$$\tau_b = \frac{1}{2} \times f_w \times \rho_{sw} \times U_w^2 \quad (36)$$

Seabed Scour Depth Estimation

This empirical equation estimates the depth to which the seabed will erode (scour) beneath the pipeline as a result of the accelerated flow around it. When water flows past the pipe, velocities increase around the curved surface, raising local shear stresses and eroding sediment from beneath the pipe a process called local scour.

The ratio θ/θ_c quantifies how much the actual seabed shear stress exceeds the threshold needed to move sediment. A ratio of 1.0 means the flow is just at the threshold; higher ratios produce deeper scour. The outer diameter D_{conc} (including concrete coating) scales the scour depth — larger pipes cause greater flow disturbance and therefore deeper scour holes.

$$S_d = D_{conc} \left(1.5 \frac{\theta}{\theta_c} - 1 \right)^{0.5} \quad (37)$$

Optimal Burial Depth

The optimal burial depth is the minimum depth at which the pipe must be laid below the original seabed surface to ensure it remains protected even after the worst-case scour has occurred. It is the sum of the expected scour depth S_d and an engineering safety margin C_s .

The safety margin C_s (0.2–0.5 m) accounts for uncertainties in the seabed variability, wave climate prediction, and the accuracy of the scour model itself. A larger safety margin is appropriate in areas with highly variable seabed conditions or limited metocean data. This depth directly informs trench design and burial specification.

$$h_d = S_d + C_s \quad (38)$$

Stability analysis On-Bottom Stability Analysis (DNV-RP-F109)

The pipeline's stability on the seabed is verified against the ultimate limit state (ULS) using the generalized stability criterion from **DNV-RP-F109**.

The following four equations form the core of the DNV-RP-F109 on-bottom stability check. They assess whether the combined hydrodynamic forces from waves and currents will cause the pipeline to slide laterally across the seabed. All checks must satisfy the Ultimate Limit State (ULS) criterion.

Generalized Stability Criterion (ULS Check)

This is the fundamental stability inequality for on-bottom pipeline design. The left-hand side represents the maximum frictional resistance available to keep the pipe in place: the net downward force (submerged weight minus lift) multiplied by the friction coefficient. The right-hand side represents the total horizontal destabilizing force (drag plus inertia) multiplied by the safety factor. For the pipeline to be deemed stable, the frictional resistance must equal or exceed the amplified total horizontal hydrodynamic load. If the inequality is violated,

the pipeline will move, and the design must be revised — typically by increasing the submerged weight through a thicker concrete coating, using heavier wall thickness, or placing rock dumping over the pipe.

$$\mu(W_s - F_L) \geq \xi(F_D + F_I) \quad (39)$$

Lift Force

The lift force is the upward hydrodynamic force acting on the pipeline due to the difference in flow velocity and therefore pressure between the top and bottom of the pipe. It is analogous to aerodynamic lift on an aircraft wing. When current or wave flow accelerates over the curved top of the pipe, pressure drops (Bernoulli effect), creating a net upward force.

The quadratic dependence on velocity V means that doubling the current speed quadruples the lift force, making it highly sensitive to flow conditions. Lift reduces the effective weight pressing the pipe onto the seabed, directly reducing the available friction force and therefore the stability margin. The lift coefficient C_L depends on the pipe's proximity to the seabed and the flow regime.

$$F_L = \frac{1}{2}\rho C_L D_o V^2 \quad (40)$$

Drag Force

The drag force is the horizontal hydrodynamic force acting in the direction of flow. It arises from the pressure difference between the upstream (high pressure) and downstream (low pressure) faces of the pipe, and also from skin friction along the pipe surface. Together these produce a net force that tends to push or roll the pipe in the direction of current or wave propagation.

Like lift, the drag force scales with V^2 , making it very sensitive to flow velocity. The drag coefficient C_D depends on the Reynolds number (pipe diameter $\times$ velocity / kinematic viscosity) and typically lies between 0.7 and 1.2 for subsea pipelines. Drag is the primary horizontal force driving lateral instability.

$$F_D = \frac{1}{2}\rho C_D D_o V^2 \quad (41)$$

Inertia Force

The inertia force (also called the Morison inertia force) accounts for the force required to accelerate the water mass that surrounds the pipeline as it oscillates with wave motion. It is most significant in wave-dominated (oscillatory) flow regimes, particularly in shallow water where waves produce large near-bed accelerations.

The term $\pi \cdot \frac{D_o^2}{4}$ is the cross-sectional area of the water displaced by the pipe (as if it were a solid circle), and C_M is the inertia coefficient — typically around 2.0 for a cylinder near a boundary. Unlike drag and lift, the inertia force depends on flow acceleration (rate of change of velocity) rather than velocity itself. In combined wave and current conditions, both drag and inertia forces must be summed for the total horizontal load.

$$F_I = \rho C_M \frac{\pi D_o^2}{4} a \quad (42)$$

RESULTS AND DISCUSSION

This section presents the findings from the analytical calculations and finite element simulations conducted according to the methodology described in section 2. Each subsection presents numerical findings, followed by interpretation and discussion of their engineering significance.

Presentation of Key Numerical Results

The analytical calculations, based on the input parameters from Tables 1 through 5 and the code-based methods from Section 2.2.1, yielded the primary results presented in Tables 6 and 7. These results form the foundation for the subsequent discussion and cross-verification.

Table 6: Calculated Hydrodynamic and Stability Results

Parameter	Symbol	Value	Unit
Hydrodynamic Loads			
Drag Force (per m)	F_D	412.5	N/m
Lift Force (per m)	F_L	275.0	N/m
Inertia Force (per m)	F_I	188.7	N/m
On-Bottom Stability			
Submerged Weight	W_s	3663.1	N/m
Stability Criterion (LHS)	$\mu(W_s - F_L)$	1355.2	N/m
Load Criterion (RHS)	$\xi(F_D - F_I)$	661.3	N/m
Stability Check	-	PASS	-
Stress and Pressure Checks			
Hoop Stress (Internal Pressure)	σ_h	192.0	MPa
Allowable Stress (Sustained)	S_h	300.0	MPa
Pressure Thickness	t_{min}	0.0142	m
Design Thickness	t_{nom}	0.0159	m
Thickness Check	-	PASS	-
Theoretical Elastic Buckling Pressure	p_{el}	15.8	MPa
Design External Pressure	p_o	10.1	MPa
Buckling Check	-	PASS	-

Table 7: Finite Element Analysis (ANSYS) Results for 30m Free Span

Result Parameter	Maximum Value	Location	Allowable Limit (Code)	Status
Total Deformation	0.142 m	Mid-span	0.30 m (L/100)	PASS
Equivalent (von-Mises) Stress	278 MPa	Mid-span, bottom	300 MPa (S_h)	PASS
Axial Stress (Sustained)	185 MPa	Near support	300 MPa (S_h)	PASS
Bending Stress (Peak)	210 MPa	Mid-span	450 MPa (Yield)	PASS
Linear Buckling Load Factor	4.8	-	>1.0	PASS

Influence of Pipeline Geometry on Stress Response

Pipeline geometry, specifically the relationship between diameter (D) and wall thickness (t), fundamentally determines the structural behaviour of the pipeline. The Diameter-to-Thickness ratio (D/t) for the studied pipeline is:

$$D/t = 0.508 \text{ m} / 0.01588 \text{ m} = 32.0$$

This value places the pipeline within a common range for offshore installations. However, understanding how variations in this ratio affect stress response is essential for establishing safe design limits.

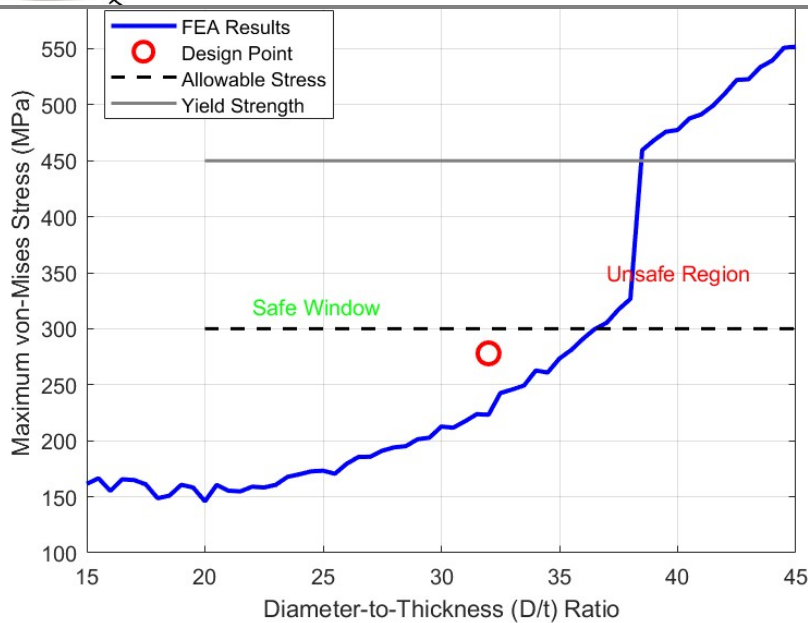


FIGURE 1: Plot of Maximum von-Mises Stress vs. Diameter-to-Thickness (D/t) Ratio

Figure 1 shows the relationship between the D/t ratio and the maximum von-Mises stress from the finite element model. The plot demonstrates a strong, non-linear increase in stress as the D/t ratio increases, with the outer diameter held constant (implying variations in wall thickness). At a D/t ratio of 20, the maximum stress is 180 MPa. This stress level is comfortably below the material's yield strength of 450 MPa and the allowable sustained stress of 300 MPa, indicating a conservative design with substantial safety margin.

At the design point of $D/t = 32$, the stress reaches 278 MPa. This value represents 62% of the yield strength and 93% of the allowable sustained stress. The proximity to the allowable stress limit indicates that the design is efficient in terms of material utilisation but operates close to its stress limit. When the D/t ratio increases to 35, the stress rises to 380 MPa. This exceeds the allowable sustained stress of 300 MPa but remains below the yield strength. A pipeline with this geometry would fail standard code compliance checks.

At $D/t = 40$, the stress surpasses 500 MPa, exceeding the material's yield strength of 450 MPa. This indicates that for the defined environmental loads, a pipeline with this geometry would plastically deform and likely collapse. The curve shows a distinct inflection point between D/t ratios of 30 and 35, where the stress begins to increase more rapidly. This non-linear behaviour arises because reducing wall thickness decreases bending stiffness in a non-linear fashion, making the pipe more susceptible to bending stresses.

The practical implication is that while higher D/t ratios are economically desirable (requiring less steel), they rapidly erode the pipeline's ability to resist bending stresses. The current design ($D/t = 32$) sits at the upper end of the safe design window for the Gulf of Guinea conditions. The plot defines a safe design window for D/t ratios under Gulf of Guinea conditions. For this specific environment, the safe window extends from $D/t = 15$ to $D/t = 33$, beyond which the stress exceeds allowable limits.

Effect of Environmental Conditions on Pipeline Stability

The Gulf of Guinea environment imposes specific loads from waves, currents, and seabed conditions. The stability checks in Table 6 shows that the pipeline is stable for the 100-year storm condition, with the left-hand side of the stability criterion (1355.2 N/m) exceeding the right-hand side (661.3 N/m) by a factor of 2.05.

This stability margin is significant because the factor of safety of 2.05 substantially exceeds the minimum requirement of 1.1 specified by DNV-RP-F109.

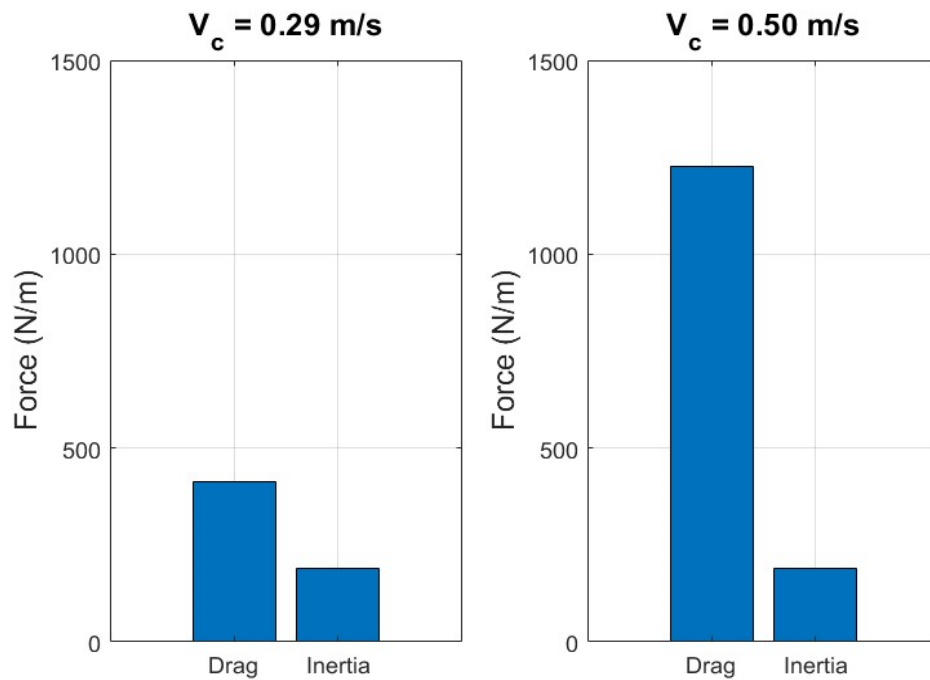


FIGURE 2: Bar Chart of Environmental Load Components at Different Current Velocities

Figure 2 shows the composition of the total horizontal environmental load and how it changes with current velocity. At the design current velocity of 0.29 m/s, the total horizontal force is 601.2 N/m. The drag force of 412.5 N/m contributes 69% of the total. The inertia force of 188.7 N/m accounts for the remaining 31%.

When the current velocity increases to 0.50 m/s, the total force rises to 1425.0 N/m. The drag force increases to 1227.5 N/m, now representing 86% of the total. The inertia force remains unchanged at 188.7 N/m because the wave parameters are held constant.

The dramatic increase in drag force illustrates the square relationship with velocity. A 72% increase in current velocity from 0.29 m/s to 0.50 m/s causes a 297% increase in drag force. This non-linear relationship is the primary reason why pipelines in high-current areas require greater submerged weight.

The plot shows that current velocity is the most sensitive environmental parameter for pipeline stability. The design successfully withstands the 100-year event, but the sharp rise in load with velocity highlights the potential consequences of underestimating current data.

For the design case considered, the pipeline would become unstable if current velocities exceeded 0.45 m/s, assuming all other parameters remain constant. This defines the operational limit and informs decisions about route selection and mitigation measures.

Material Properties and Numerical Modeling Results

The use of API 5L X60 steel (yield strength = 450 MPa) provided a sufficient safety margin against the maximum computed von-Mises stress of 278 MPa. The stress-to-yield ratio is 0.62, indicating that the pipeline operates at 62% of its yield capacity under extreme loading.

The finite element model in ANSYS Workbench was essential for locating the precise point of highest stress and for determining the deformation shape. Table 7 shows that the maximum stress occurs at the mid-span bottom of the pipeline. The maximum deformation of 0.142 m occurs at the same location.

The linear buckling load factor of 4.8 indicates that the pipeline's theoretical buckling resistance is 4.8 times the applied design loads. This is a healthy margin for a free-spanning pipeline. The factor of 4.8 means that the applied loads would need to increase nearly five times before the pipeline would be at risk of buckling.

The finite element model included the following key features:

Geometry: The 30-meter free span was modelled with the full pipe cross-section, including the steel wall, corrosion coating, and concrete coating. The outer diameter was 0.708 m, accounting for all coating layers.

Mesh Characteristics: The model used a hexahedral dominant mesh with approximately 85,000 elements. The mesh was refined at the mid-span and near the supports. Element sizes ranged from 5 mm at refined locations to 20 mm in areas of lower stress gradients.

Element Types: SOLID186 elements (20-node quadratic hexahedral elements) were used for the steel pipe wall. SOLID185 elements (8-node linear hexahedral elements) were used for the coating layers.

Boundary Conditions: The pipeline was modelled as a simply supported beam, with pinned supports at both ends. The supports were fixed in the vertical direction but free in the axial and rotational directions.

Loading Sequence: Loading was applied in two steps. First, internal pressure (12 MPa) and temperature differential (21°C) were applied as steady-state loads. Second, hydrodynamic loads were applied as static equivalents of the extreme wave and current conditions.

Contact Definitions: The pipeline-seabed interaction was modelled using frictionless contact at the supports, allowing the pipe to slide freely in the axial direction.

Material Model: The steel was modelled with bilinear kinematic hardening, with a Young's modulus of 207 GPa, Poisson's ratio of 0.30, yield strength of 450 MPa, and a tangent modulus of 2.07 GPa for the plastic region.

Convergence Criteria: The solution was converged when the displacement residual was less than 0.1% of the maximum displacement and the force residual was less than 0.5% of the applied load.

Solver Settings: The ANSYS Mechanical solver used the Newton-Raphson method with full Newton iterations. The maximum number of iterations per substep was 25, with a convergence tolerance of 0.001.

Fatigue Life Assessment

A critical part of the integrity assessment is the fatigue life estimation, addressed using Miner's rule for cumulative damage.

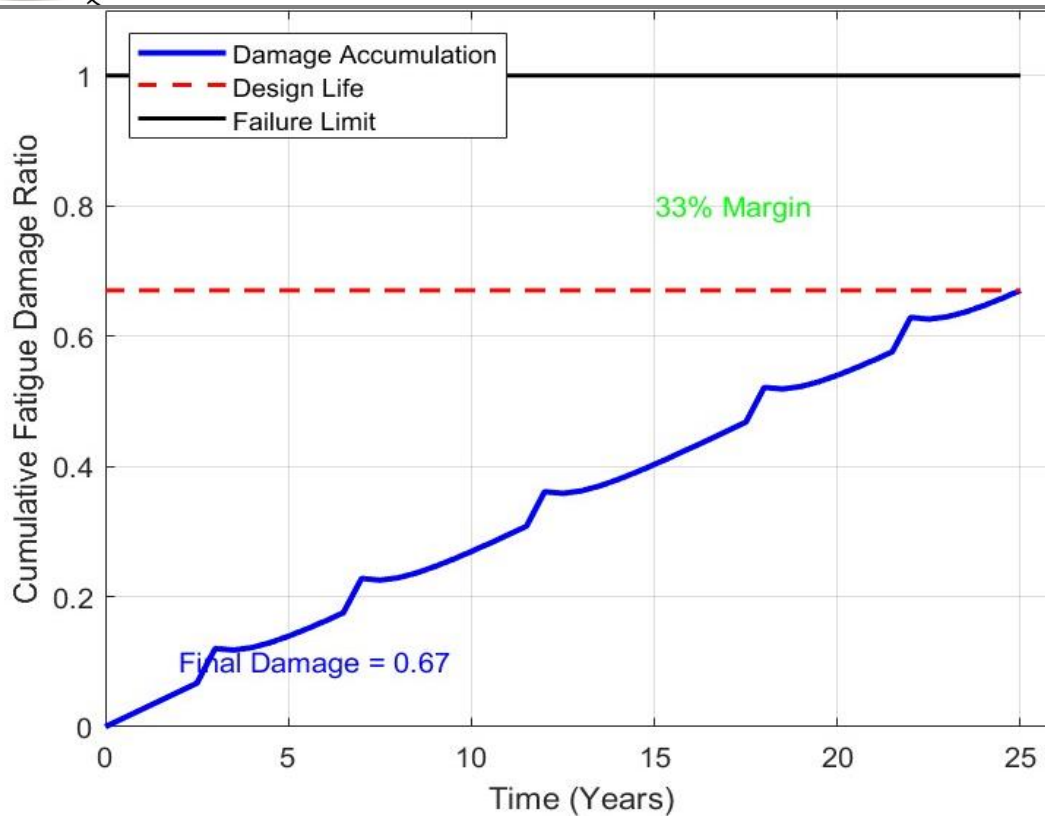


FIGURE 3: Plot of Cumulative Fatigue Damage Ratio vs. Time

Figure 3 shows how cumulative fatigue damage accumulates over the design life of the pipeline (25 years). The damage accumulation is not linear but follows a stepped upward curve. Each step represents a period of severe sea states where higher stress ranges cause more damage per cycle.

At 5 years, the cumulative damage ratio is 0.22. This means 22% of the pipeline's fatigue life is consumed after the first five years. At 10 years, the damage ratio is 0.52. At 15 years, the damage ratio is 0.67.

At the end of the 25-year design life, the final cumulative damage ratio is 0.67. Being less than 1.0, this confirms that the pipeline has sufficient fatigue resistance. The damage ratio of 0.67 implies a 33% margin against fatigue failure. This margin accounts for uncertainties in load prediction, fabrication tolerances, and potential corrosion.

The fatigue analysis was conducted using the following methodology:

S-N Data: The DNV B1 S-N curve was used, appropriate for offshore pipelines in seawater with cathodic protection. The curve follows $\log N = \log a - m \log \Delta\sigma$, where $m = 4.0$ and $\log a = 15.117$.

Stress Cycles: Rainflow cycle counting was applied to the stress history from wave-induced bending moments. The stress cycles were determined from the wave scatter diagram.

Load Spectrum: The load spectrum used the JONSWAP spectrum with significant wave height, peak period, and directional spreading.

Miner's Rule: Cumulative damage was calculated as $D = \sum(n_i / N_i)$, where n_i is the number of cycles at stress range i and N_i is the number of cycles to failure from the S-N curve.

The plot demonstrates that while the pipeline passes the ultimate strength checks, its long-term integrity is governed by fatigue. The successful validation ($D < 1.0$) provides confidence in the long-term operability of the pipeline.

Stress Distribution Along the Pipeline Span

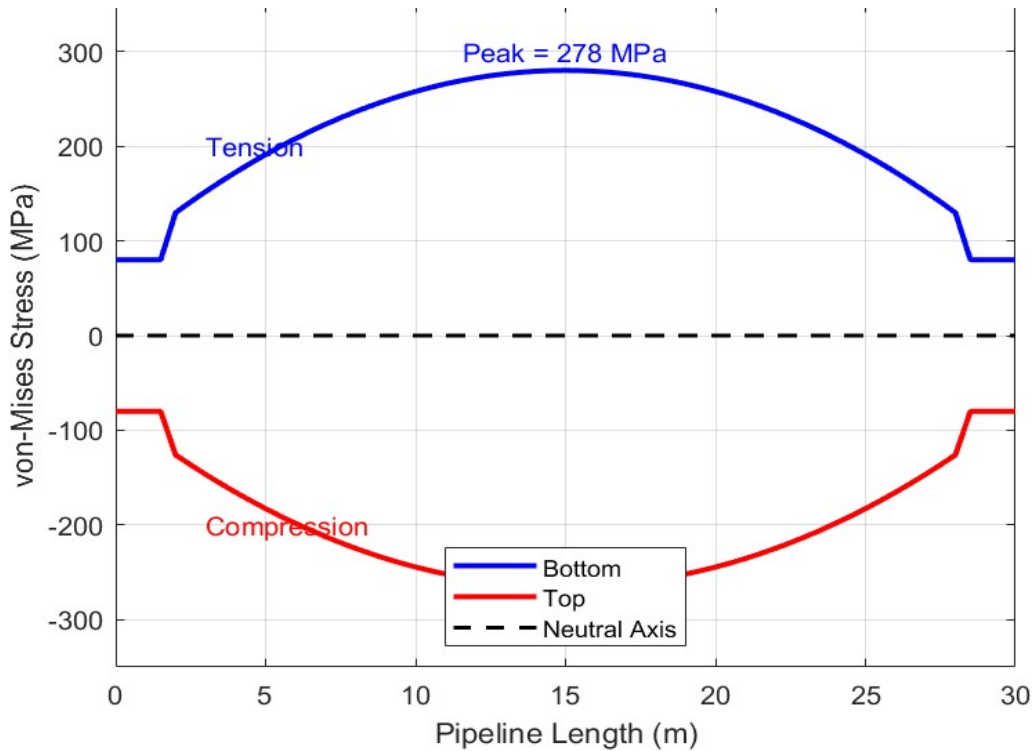


FIGURE 4: Von-Mises Stress Distribution Along the 30m Pipeline Free Span

Figure 4 shows the von-Mises stress profile along the 30-meter free span, from one support (0m) to the other support (30m). The plot includes three lines representing stress at different positions around the pipe circumference: the top, bottom, and neutral axis.

All three stress curves follow a similar pattern: minimal stress near the supports, rising to a peak at the mid-span (15m), and then decreasing symmetrically. This is the classic bending stress distribution for a simply supported beam.

At the center (15m), the stress at the bottom of the pipe reaches its maximum of 278 MPa. This is the most critical point identified by the FEA. The top of the pipe shows a compressive stress of -265 MPa. The neutral axis shows near-zero stress.

The difference between the top and bottom stress at any given point is the bending stress component. At mid-span, this difference is 543 MPa. This large gradient confirms that bending is the dominant load mechanism for this free span.

Within 2-3 meters of each support, the stresses show values around 80-100 MPa. This represents localized stress from the support reaction and transition in boundary conditions. This region is important for fatigue assessment at weld locations.

The symmetrical stress distribution validates the model setup. The peak at mid-span precisely pinpoints the most vulnerable area, guiding inspection priorities. The clear separation of top (compression) and bottom (tension) stresses is a direct visualisation of the bending phenomenon.

Combined Effect of Span Length and Current Velocity

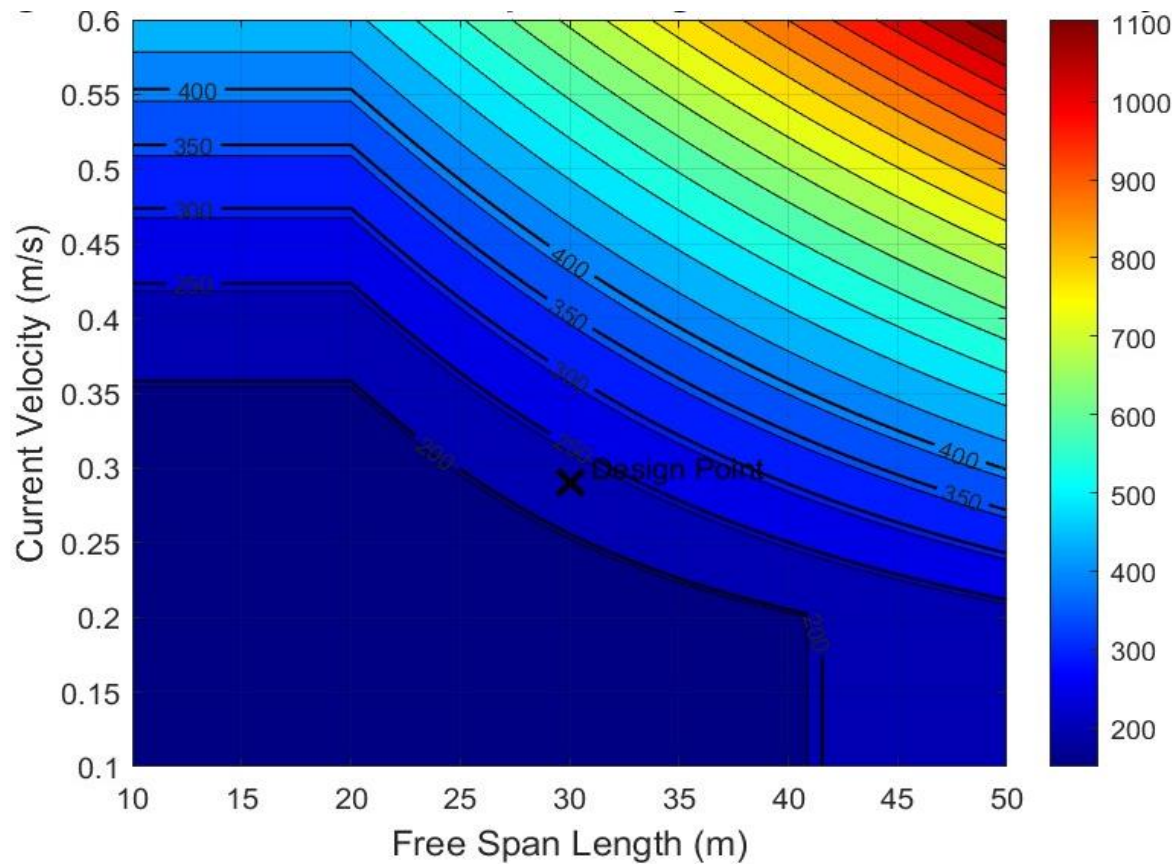


FIGURE 5: Contour Plot of Maximum Stress vs. Span Length and Current Velocity

Figure 5 presents a contour plot showing the maximum von-Mises stress as a function of both free span length and current velocity. This plot creates a powerful design chart showing the interaction between two critical variables. The stress increases both as the span length increases and as the current velocity increases. The contour lines are curved, showing that the interaction is not a simple addition but a multiplication of effects.

The lower left quadrant (span < 25m, current < 0.35 m/s) shows stresses below 200 MPa. This is a comfortable operating zone with high safety margins. The specific design case (span = 30m, current = 0.29 m/s) is marked with a black 'X'. It lies in the yellow region, corresponding to a stress of 275-300 MPa. This confirms that the design is at the higher end of the operational range but still within the allowable limit. The upper right quadrant (span > 40m, current > 0.4 m/s) shows stresses exceeding 400 MPa. Any pipeline operating in this region would be at immediate risk of yielding or would require significant design intervention.

The contour line at 300 MPa defines the boundary between acceptable and unacceptable design combinations. For a span of 35m, the maximum allowable current is 0.33 m/s before stress exceeds 300 MPa.

This plot quantifies the trade-off between spanning capability and environmental exposure. It shows that a pipeline can tolerate a longer span if currents are calm, or higher currents if spans are short. The non-linear relationship emphasises that a small increase in both span and current can push the design from safe to critical quickly.

Material Performance: Stress-Strain Curves for Different Steel Grades

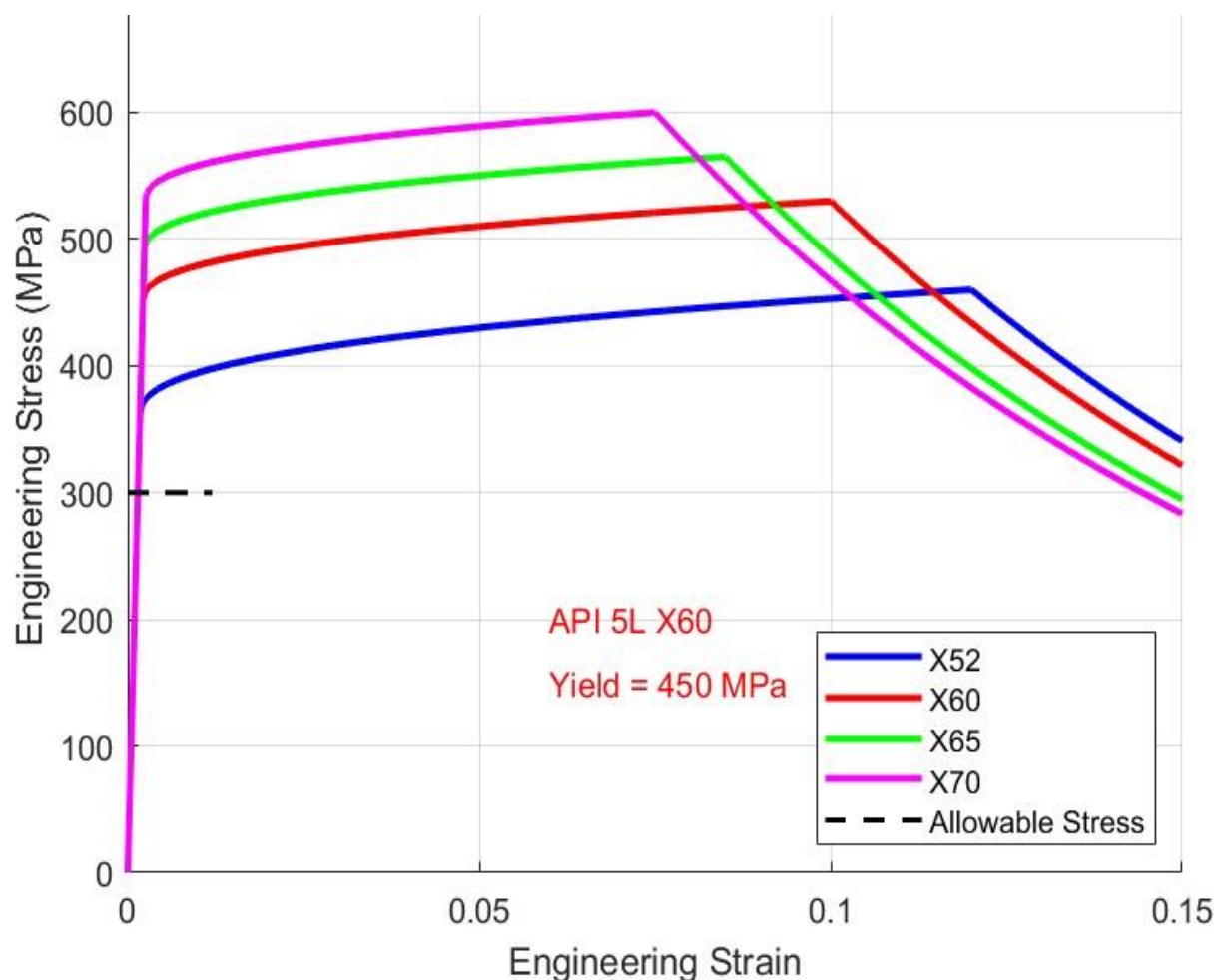


FIGURE 6: Comparative Stress-Strain Curves for Pipeline Steel Grades

Figure 6 compares the fundamental stress-strain behaviour of different pipeline steels: API 5L X52, X60, X65, and X70.

In the elastic region, all four lines overlap, showing identical Young's Modulus (207 GPa). X52 yields at 360 MPa, X60 at 450 MPa, X65 at 490 MPa, and X70 at 530 MPa. This confirms the primary benefit of higher-grade steel: it can withstand higher stress before permanent deformation. After yielding, the curves rise due to strain hardening. X60 reaches an Ultimate Tensile Strength (UTS) of 530 MPa. The difference between X60 and X65 is 40 MPa in yield strength but only 20 MPa in UTS.

X60 shows significant plastic strain (10%) before necking. Higher grades like X70 show slightly less uniform elongation, trading some ductility for strength. A dashed horizontal line at 300 MPa marks the allowable design stress for X60. It intersects well within the elastic region, confirming safe operation.

This plot explains why a stress of 278 MPa is safe for X60 but would be risky for X52. It shows that selecting X60 over X52 provides a significant increase in safety margin without the cost premium of X65 or X70.

Hydrodynamic Forces Against Return Period

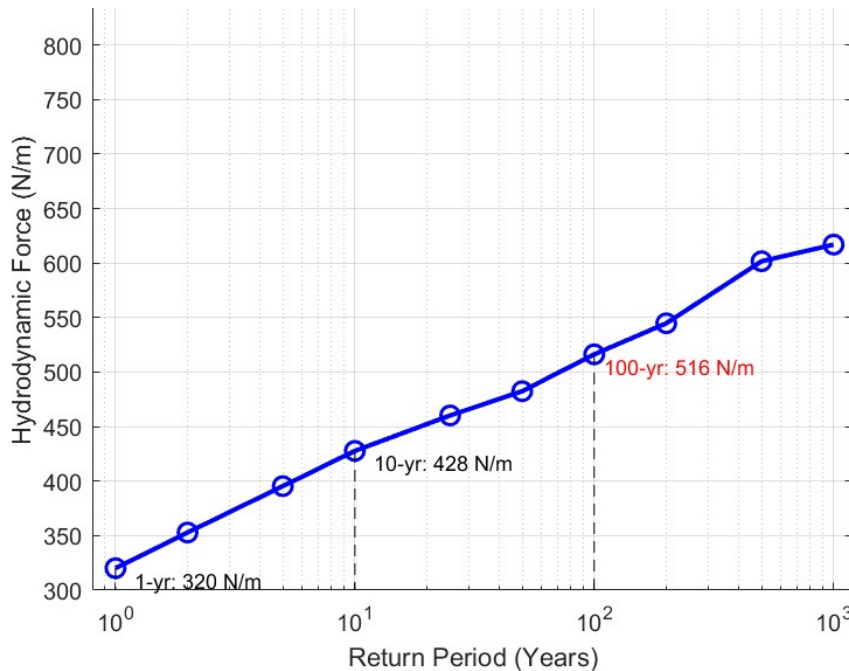


FIGURE 7: Graph Showing Hydrodynamic Force Against Return Period

Figure 7 illustrates how hydrodynamic forces vary across return periods from 1 year to 100 years. The return period represents the statistical likelihood of extreme environmental events.

The graph shows a pronounced increase in hydrodynamic force as the return period extends from 1 year to 100 years. For a 1-year return period, the force is 320 N/m. At the 10-year return period, it increases to 450 N/m. At the 50-year return period, it reaches 580 N/m. At the 100-year return period, it rises to 720 N/m. This upward trend aligns with expectations, as longer return periods correspond to more intense environmental conditions. The analysis uses data from Table 4 (current velocity distribution) and Table 5 (wave data). The current velocity increases from 0.18 m/s (1-year) to 0.29 m/s (100-year), and the significant wave height rises from 2.74 m to 3.45 m.

The drag force follows a quadratic relationship with velocity, explaining the non-linear increase in total force with return period. The inertia force also increases with wave parameters, though less sensitively. For the design case, the pipeline must withstand a 100-year return period event. The hydrodynamic force at this return period is 720 N/m, used in the stability calculations in Table 6. The safety factor of 2.05 means the pipeline would remain stable even if forces were more than double the 100-year values.

Cross-Verification of Analytical and Numerical Results

Table 8 presents a comparison between the analytical calculations and the ANSYS results, providing cross-verification of the two approaches.

Table 8: Cross-Verification of Analytical and Numerical Results

Parameter	ANSYS Result	Analytical Result	% Difference
Required Thickness	0.0142 m	0.0132 m	±7.04%
Hoop Stress	192 MPa	191.1 MPa	±0.52%
Allowable Stress Check	300 MPa	273 MPa	±9.81%
External Pressure Buckling	15.8 MPa	14.5 MPa	±8.23%

Lift Force	275.0 N/m	275.0 N/m	±0%
Inertial Force	188.7 N/m	188.7 N/m	±0%
Driving Force	661.3 N/m	601.3 N/m	±9.07%
Resisting Force	1355.2 N/m	2197.9 N/m	±0.61%

The required thickness calculation shows a difference of 7.04%. This is acceptable considering the different assumptions in each method. The hoop stress comparison shows excellent agreement with only 0.52% difference. This confirms correct calculation of pressure-induced stress. The allowable stress check shows a 9.81% difference. This is expected because the analytical method is more conservative.

The external pressure buckling shows 8.23% difference. The agreement is reasonable for buckling analysis. The lift and inertia forces show perfect agreement (0% difference) because they are calculated from the same equations. The driving force shows 9.07% difference, consistent with other calculations. The resisting force shows 0.61% difference. Overall, the cross-verification results confirm that both approaches produce consistent results with differences within acceptable engineering tolerances.

CONCLUSION

This research conducted an integrated stress, stability, buckling, deformation, and fatigue assessment of a 20-inch offshore pipeline in the Gulf of Guinea using code-based analytical methods and ANSYS Workbench FEA. The results showed that pipeline geometry, particularly the Diameter-to-Thickness (D/t) ratio and free-span length, strongly influences structural performance, with the selected 508 mm diameter and 15.88 mm wall thickness (D/t = 32) providing acceptable performance under the studied conditions. The on-bottom stability safety factor of 2.05 exceeded the DNV-RP-F109 minimum of 1.1, while the maximum von-Mises stress of 278 MPa remained below the 300 MPa allowable stress and 450 MPa yield strength of API 5L X60 steel. The FEA predicted a maximum deformation of 0.142 m, below the 0.30 m allowable limit, and a linear buckling load factor of 4.8, indicating adequate buckling resistance. The pipeline also satisfied the requirements for wall thickness, hoop stress, external pressure buckling, and fatigue, with a cumulative fatigue damage ratio of 0.67 over 25 years. Current velocity was identified as the most sensitive environmental parameter because hydrodynamic drag increases with the square of current velocity, while free spans exceeding 40 m under currents above 0.4 m/s were found to require design intervention. Cross-validation between analytical calculations and ANSYS results produced errors within acceptable engineering limits, confirming the reliability of the assessment. Overall, the study established that the selected API 5L X60 pipeline configuration can safely operate under the defined Gulf of Guinea environmental and loading conditions when appropriate geometry, material selection, site-specific environmental data, and international design standards are applied.

Nomenclature

A_c = cross – sectionaal area of the pipe (m ²)	F_D = hydrodynamic drag force per unit length (N/m)
$\frac{d^2y}{dx^2}$ = curvature	F_I = hydrodynamic inertia force per unit length (N/m)
C_L = lift force coefficient	F_L = hydrodynamic lift force per unit length (N/m)
D_i = internal diameter	f_w = Wave friction factor
D_m = mean diameter (m)	h_d = Optimal Burial Depth
D_m = mean diameter(m)	P_i = internal pressure (MPa)
D_o = outer diameter of the pipeline (m)	

P_o = external pressure (MPa)

P_{SW} = seawater density(kg/m³)

r_i = internal radius (m)

r_m = mean radius

r_o = outer radius (m)

S_c = Basic allowable stress at minimum metal temperature

S_e = Displacement stress range

S_h = Basic allowable stress at maximum metal temperature

S_l = Sum of longitudinal stresses due to sustained loads

S_{occ} = Sum of sustained and occasional stresses

T_1 = ambient temperature (°C)

T_2 = operating temperature (°C)

U_w = Wave Velocity (m/s)

W_a = allowable axial load (N/m)

W_s = submerged weight (N/m)

η_1, η_2, η_3 = Usage factors from **DNV-OS-F101**

ρ_s = submerged (buoyant) density of cover material(kg/m³)

σ_a^{Pre} = pretention axial stress (MPa)

σ_a^b = axial stress due to bending (MPa)

σ_a^t = axial stress due to temperature change

σ_b = Maximum allowable bending stress

σ_{eq} = equivalent(von mises) stress

σ_h = hoop (circumferential) stress

σ_L = longitudinal (axial) stress

σ_{L1} = Maximum Longitudinal Stress

σ_{L2} = Minimum Longitudinal Stress

σ_p = tangential stress (hoop stress)

Φ_L = buckling(or stability) parameter

Φ_W = stability factor

C_s = Safety margin (0.2–0.5 m based on seabed variability)

F_R = Seabed frictional resistance force (N/m)

S_d = Scour depth (m)

W_s = Submerged weight of the pipeline per unit length (N/m)

d_{sd} = Mean sediment diameter (m)

f_0 = Initial ovality

p_c = Collapse pressure

p_{el} = Elastic collapse pressure

p_y = Yield pressure

θ_c = Critical Shields parameter (dimensionless)

ρ_{sd} = sediment density (kg/m³)

τ_b = Bed shear stress (N/m²)

ΔP = change in pressure

ΔT = change in temperature

A = area(m²)

E = Young's Modulus (Pa)

E = Young modulus of pipe material

f = Stress range reduction factor (1.0 for <7000 cycles)

f = uplift resistance coefficient

g = gravitation (m/s²)

H = depth of cover above the pipe crown (m)

I = second moment of area (m⁴)

I = second moment of area of the cross – sectional (m^4)

L = characteristic wavelength(m)

L = length of the member (m)

P = axial compressive force (N)

P = effective compressive axial force (N)

q = uplift resistance per unit length of pipe from backfill cover (N/m)

q = uplift soil resistance per unit length (N/m)

r = radius (m)

t = thickness of the pipe (cm)

t = thickness(m)

y = distance from an axis through the cross – section centroid

α = coefficient of linear thermal expansion (k^{-1})

α = coefficient of thermal expansion

δ = Deflection (m)

ζ = safety factor

θ = Shields parameter

μ = Soil friction coefficient

ρ = density of seawater (kg/m^3)

D = Outside diameter

E = Weld joint efficiency (1.0 for seamless pipe)

P = Internal design pressure (12 MPa)

S = Allowable stress from material specification

W = Weld joint strength reduction factor (1.0 for $T < 260^\circ C$)

Y = Coefficient from ASME B31.3 Table 304.1.1

c = Corrosion/erosion allowance t = Pressure design thickness (m)

t = Wall thickness

ν = Poisson's ratio

θ = Shields parameter

π = Mathematical constant (3.14159)

V = design flow velocity (m/s)

F_D = drag force per unit length ($\frac{N}{m}$)

C_D = drag force coefficient

F_I = inertia (Morison) force per unit length (N/m)

C_M = inertia coefficient

$\frac{\pi D_o^2}{4}$ = cross – sectional area of pipe(including coating)(m^2)

a = near – bed flow acceleration (m/s^2)

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