



Structural Sizing for EDP Valve Body for HPHT Wellhead

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The reduction in production from active oil wells in Brazil and the depletion of reservoirs have driven research into deep-sea reservoirs, especially on the Brazilian equatorial margin and in projects such as Sergipe-Águas-Profundas. This work aims to design an EDP valve for intervention in HPHT (High Pressure, High Temperature) offshore wells. API and ASME standards, in addition to technical literature, were used to define the appropriate material and mechanical properties, and to calculate allowable stresses, as well as the stresses acting on the body and its components, using stress criterion to ensure equipment integrity. Stress concentrations were applied in regions with geometric alterations, as these are more critical. Finite element analysis more accurately assessed the equipment's behavior under the established conditions. Both calculation models were within the allowable design limits and demonstrate that the extreme conditions of HPHT wells require a thorough analysis of the design parameters.

Keywords: Oil and Gas, High-Pressure-High-Temperature; Intervention; Emergency Disconnect Package; Structural Analysis.

1.Introduction

Brazil seeks to reduce oil imports and explore deep-water reservoirs, such as the projects of Sergipe Águas Profundas and the equatorial margin, aiming for self-sufficiency, due to their high production potencial [1,2]. Those projects must also align with current ecological demands, ensuring the protection of the ocean environment, amid contemporary environmental and climate challenges [3]. In HPHT deep wells, conditions can reach 103.5 MPa and 177 °C, which require intervention operations to ensure functionality [4].

These operations support equipment repair, maintenance, and fluid injection, extending service life and ensuring integrity, which boosts production and prevent accidents, such as oil leaks, that may harm the environment [5-8]. For this, subsea systems between the

wellhead and intervention riser are used, as shown in figure 1: the Lower Riser Package (LRP) for well control and the Emergency Disconnection Package (EDP) for cutting the intervention string and sealing the well [9–11].

Figure 1. Positioning and Structure of Intervention Systems [11,12]

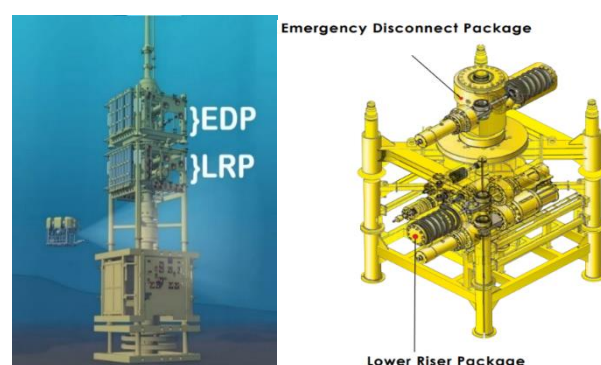
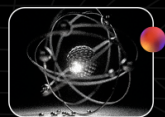


Figure 1 represents these systems, with accumulators and intervention panels; the EDP, located at the top, functions as an additional pressure barrier, while the LRP serves as the primary barrier, containing two main valves for



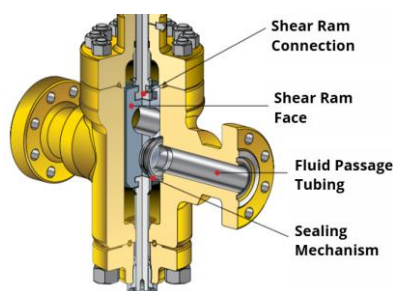
wireline and coiled tubing cutting [12,13]. These systems, exposed to adverse conditions that may lead to failures, are essential for operational safety [14,16]. Efforts focus on improving efficiency and simplifying intervention system design to reduce costs and support deep-water oil production [17]. This study presents the initial phase of a structural sizing project for a forged EDP valve body for offshore HPHT wellheads, covering material selection, stress and dimension calculations, and a preliminary fatigue life analysis and finite element verification.

2.Theoretical Basis

The EDP may adopt various structural configurations, differing in cavity layout, shape, or shearing valve type. Figure 2 shows a linear-actuated valve for well sealing in operations up to 137.9 MPa, suitable for HPHT use. Its forged, unified structure protects internal components and mechanisms. Forging minimizes defects and increases strength, with materials chosen for high mechanical and corrosion resistance [18-20].

Proper sizing is essential for safe interventions and to maintain production under critical conditions [4,13,18,21].

Figure 2. EDP Valve Body Structure for Intervention Operations [17]



For HPHT analysis, API 17TR8 outlines key design and construction guidelines,

addressing corrosion, fatigue, and plastic collapse [22–26]. Table 1 defines corrosion failure criteria, setting minimum H₂S concentrations for sulfide stress corrosion based on pressure [27]. At 103.5 MPa, H₂S above 2.5 ppm classifies the environment as sour, requiring metallurgical evaluation for operational viability [24].

Table 1. Minimum H₂S Concentration for Acidic Environment Classification. [22]

Rated Working Pressure	69 MPa	103 MPa	138 MPa	172 MPa	207 MPa
Concentration of H ₂ S (ppm)	5	3,3	2,5	2	1,7

At high temperatures, steels lose yield strength, requiring reduction factors in the design of bodies, flanges, and bolts [28–31]. Subsea equipment must comply with structural integrity standards [23]. This study applies API 17TR8's flowchart for 103.5–137.9 MPa and ASME VIII Divisions 2 and 3 [22]. Stress analysis follows Lamé's method for thick-walled cylinders [32], with allowable stresses from ASME II-D and VIII-2, and sizing features per API 6A [33-35].

Fatigue is analyzed via stress cycling and fracture mechanics, crucial for HPHT conditions, supported by tests and statistical models [33,36–38]. Plastic collapse considers linear or elasto-plastic behavior, focusing on stress concentrators [33,39-41]. Testing validates designs but is costly, so modeling and simulations are used [19,25,33,42–45].

The Finite Elements Method (FEM), endorsed by API 17TR8, enables accurate, cost-effective simulations of complex geometries [22,26,40,46,47].



The aforementioned standards, despite being written by international organizations, are recognized by Brazilian corporations and agencies, being used and referenced for projects in the national territory[48-50].

3. Methodology

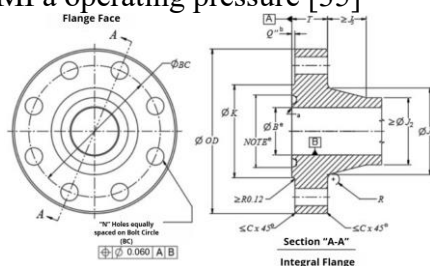
The methodology defined design conditions, developed an analytical model for load, stress, and fatigue life, and applied FEM.

3.1. Design Conditions

Design pressure was 103.5 MPa, operating 51.8 MPa, and temperature 176.7 °C. based on 3000 m HPHT wells in Potiguar and Sergipe-Alagoas basins [1,51–55]. The analyses covered stress, deformation, fatigue, and hydrostatic tests [22,33].

An 11" cylindrical shear valve body with two lateral holes (Figure 2) was sized analytically and numerically. Forged AISI 4145H steel, quenched and tempered, was the chosen material, for its acceptable hardness, nickel content, and corrosion resistance [27,56]. Its ambient yield strength is 862 MPa [57,58], reduced to 90% for high temperature, per API 6MET [28]. An API 6A integral flange (Fig. 3), was selected based on the proposed pressure and dimensions [35,59].

Figure 3. Flange 6A with BX seal for 103.5 MPa operating pressure [35]



The hub's vertical flange nominal diameter was set at 11", and the side flange was 13 5/8", to fit the shear ram. Detailed flange and bolt sizes are shown in Figures 4 and 5 [35,59].

Figure 4. Dimensions for API 6BX Integral Flanges to 103.5 MPa [35]

Nominal size and bore of flange		Basic flange dimensions							
		Maximum bore	Outside diameter of flange			Maximum chamfer	Diameter of raised face	Total thickness of flange	Large diameter of hub
(in)	mm	B	OD	tol.	C	K	T	J ₁	J ₂
						±1.6	$\frac{+3}{-3}$	$\frac{0}{-3}$	
11.	279	280.2	815	± 3	6	454	187.4	584.2	427.0
13 ⁵ / ₈	346	346.9	885	± 3	6	541	204.8	595.3	528.6

Figure 5. Bolt Dimensions for API 6BX Flanges to 103.5 MPa [35]

Nominal size and bore of flange		Bolting dimensions								
		Length of hub	Radius of hub	Diameter of bolt circle	Number of bolts	Diameter of bolts	Diameter of bolt holes	Minimum length of stud bolts	Ring number	
(in)	mm	J ₃	R	BC		(in)	tol. ^a	L ₃₃₀	BX	
11	279	235.7	16	711.2	20	2	54	+ 2.5	490	158
13 3/8	346	114.3	25	771.5	20	2 1/4	61	+ 2.5	540	159

For fasteners, SA-320 Grade L7 material was chosen for subsea use, with required properties in Figure 6 [29,60]. Suitable for bolts up to 2,5" (65 mm), heat treated by quenching and tempering, it has 35 HRC hardness, 724 MPa yield strength, and 861,85 MPa tensile strength. A safety factor of 2.0 was applied, considered acceptable [33,34,60].

Figure 6. Mechanical Properties of Ferritic Materials for High-Alloy Fasteners [61]

Class and Grade, Diameter, in. (mm)	Heat Treatment	Minimum Tempering Temperature, °F (°C)	Tensile Strength, min. ksi (MPa)	Yield Strength, min. ksi (MPa) (0.2 % offset)	Elongation in 2 in. or 50 mm, min. %	Reduction of Area, min. %	Hardness, max
L7, L7A, L7B, L7C, L7D, L7E, L7F, L7G, L7H			125	105	16	50	321 HBW or 35 HRC
2 1/2 (65) and under ^a	quenched and tempered	1100 (593)	(860)	(725)			

3.2. Allowable Stresses and Loads Calculation

Allowable stresses were set as the minimum yield or rupture ratio, with its respective factors, as shown in eq. 1 [32].

$$S_{adm} = \min \left(\frac{S_y}{1.5}; \frac{S_u}{2.4} \right) \quad (1)$$

Loads were estimated from body-fixing bolts and flange forces under pressure and sea current moments, according to eqs. 2 and 3 [33].

$$M_o = abs[(H_D * h_D + H_T * h_T + H_G * h_G) * B_{SC} + M_{oe}) * F_s] \quad (2)$$

$$M_g = W_g * (C + G) * B_{SC} * F_s * 0,5 \quad (3)$$

These body stresses was defined accounting for stress concentrators and secondary stresses, and safety factors [33,34,62]. The stiffness index was calculated per eq. 4 [33].

$$J = \frac{52,14 * V * M}{L * E * K_R * h_o} \quad (4)$$

3.3. Analysis of Lamé and Secondary Stresses

Lamé stress analysis, using defined dimensions and corrosion allowance, computed wall thickness and stresses by eqs. 5–7 [32].

$$\sigma_{rr} = \frac{(p_1 * R_1^2)}{R_2^2 - R_1^2} - \frac{R_1^2 * R_2^2}{r^2 * (R_2^2 - R_1^2)} * (p_1) \quad (5)$$

$$\sigma_{\theta\theta} = \frac{(p_1 * R_1^2)}{R_2^2 - R_1^2} + \frac{R_1^2 * R_2^2}{r^2 * (R_2^2 - R_1^2)} * (p_1) \quad (6)$$

$$\sigma_{zz} = \frac{(p_1 * R_1^2)}{R_2^2 - R_1^2} + \frac{F}{\pi * (R_2^2 - R_1^2)} \quad (7)$$

Equivalent stresses by von Mises criterion (eq. 8) included self-weight, thermal effects, and concentration factors [33,41,61].

$$\sigma'_{eq_max} = \sqrt{\frac{(\sigma'_{\theta\theta} - \sigma'_{rr})^2 + (\sigma'_{rr} - \sigma'_{zz})^2 + (\sigma'_{zz} - \sigma'_{\theta\theta})^2}{2}} \quad (8)$$

3.4. Fatigue Life Analysis

Fatigue life followed the modified Goodman criterion, with resistance limits, factors and stresses, as shown in eq. 9 [40].

$$\frac{\sigma'_m}{s_{ut}} + \frac{\sigma'_a}{s_f} = 1 \quad (9)$$

3.5. Finite Elements Method

FEM model evaluated stress–strain response under complex loading and geometry, complementing analytical results [22,33].

4. Results and Discussions

The EDP was designed per load and dimensional requirements. Figures 7 and 8 shows the main body and a full 3D CAD model, with external structure and an internal sealing section. The design features a vertical bore, for coiled tubing, and a horizontal chamber, with a ram that shears the tube and seals the system via O-rings.

Figure 7. Top (a), Side (b) and Front View with Section Cut (c) of the EDP Body.

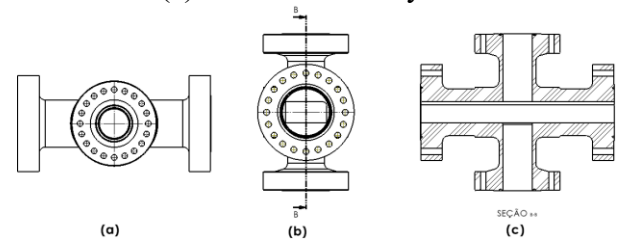
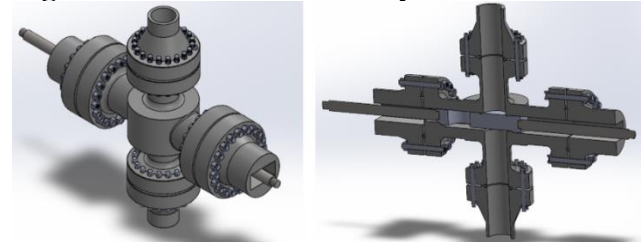


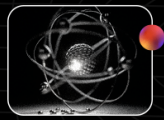
Figure 8. EDP CAD Assembly



To verify structural integrity, stresses in the nozzle opening regions were analyzed. Allowable stresses for flanges, bolts, and the EDP body at design (177 °C) and sealing (ambient) temperatures are presented in Table 2, with body and flange stresses being identical due to its material equivalence. The stress reduction due to temperature is responsible for the difference in stresses between the two temperature conditions.

Table 2. Allowable Stress for Flanges and Bolts

Equipment Component	Allowable Stress at Design Temperature (MPa)	Allowable Stress at Sealing Temperature (MPa)
Body and Flanges	415,50	458,33
Flange Bolts	337,55	358,33



Flange load calculations considered the bending moments induced by sea current forces, which increase stresses acting on the body, as calculated previously. Table 3 presents the resulting loads for each designed flange. Stress concentrators were then accounted for, and flange integrity was assessed.

Table 3. Bending moment at flanges caused by current force.

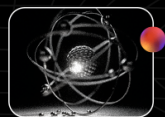
Internal Diameter (In.)	Normal Force (N)	Shear Force (N)	Bending Moment (J)
11''	650	1166,8	1,4E+5
13 5/8''	650	1798,3	2,2E+5

Table 4 lists the safety factors for operating, sealing, hydrostatic test, and combined

stress conditions for both flanges, calculated by dividing the material's allowable stress by the acting stresses, following ASME VII-2 criteria, also shown in Table 4. This standard considers the unit as the minimum value to ensure equipment integrity, applying safety parameters as constants in the calculations, being a usual practice at the oil and gas sector. Therefore, all safety factors calculated below were acceptable, above the recommended minimum limit of 1, indicating that the equipment, with the calculated dimensions and chose materials, can withstand the forces and stresses acting on it [33].

Table 4. Safety Factors for EDP Flange Design

Component	Analyzed Condition	Calculation Method	Safety Coefficient
Axial Flange	Operation (With concentrator)	$\frac{1,5 * \sigma_{adm_op}}{\sigma_{axial_op} * K'_t}$	7,72
	Sealing (With concentrator)	$\frac{1,5 * \sigma_{adm_sg}}{\sigma_{axial_sg} * K'_t}$	5,80
	Hydrostatic Test (With concentrator)	$\frac{0,9 * S_{y_Aço_4145}}{\frac{P_{TH}}{P_i} * \sigma_{axial_op} * K'_t}$	6,45
	Combination of Sealing Stresses (Axial + Tangential)	$\frac{2 * \sigma_{adm_sg}}{\sigma_{axial_sg} + \sigma_{tan_sg}}$	8,57
	Combination of Sealing Stresses (Axial + Radial)	$\frac{2 * \sigma_{adm_sg}}{\sigma_{axial_sg} + \sigma_{radial_sg}}$	4,75
	Combination of Stresses in Operation (Axial + Tangential)	$\frac{2 * \sigma_{adm_op}}{\sigma_{axial_op} + \sigma_{tan_op}}$	11,40
	Combination of Stresses in Operation (Axial + Radial)	$\frac{2 * \sigma_{adm_op}}{\sigma_{axial_op} + \sigma_{radial_op}}$	6,22
Side Flange	Operation (With concentrator)	$\frac{1,5 * \sigma_{adm_op}}{\sigma_{axial_op} * K'_t}$	9,03
	Sealing (With concentrator)	$\frac{1,5 * \sigma_{adm_sg}}{\sigma_{axial_sg} * K'_t}$	9,52
	Hydrostatic Test (With concentrator)	$\frac{0,9 * S_{y_Aço_4145}}{\frac{P_{TH}}{P_i} * \sigma_{axial_op} * K'_t}$	7,55
	Combination of Sealing Stresses (Axial + Tangential)	$\frac{2 * \sigma_{adm_sg}}{\sigma_{axial_sg} + \sigma_{tan_sg}}$	14,58
	Combination of Sealing Stresses (Axial + Radial)	$\frac{2 * \sigma_{adm_sg}}{\sigma_{axial_sg} + \sigma_{radial_sg}}$	5,56
	Combination of Stresses in Operation (Axial + Tangential)	$\frac{2 * \sigma_{adm_op}}{\sigma_{axial_op} + \sigma_{tan_op}}$	13,83



A flange stiffness index analysis also confirmed compliance, with values below 1 for both sealing and operating conditions.

In the Lamé stress analysis with the Von Mises criterion, considering secondary stresses, it was found that wall holes created stress concentrators exceeding design limits. To mitigate this, lateral projections were added to the hole region (Figure 8), sized so that their external surface extended beyond the stress concentration

zone, preventing overload on the necks between projections and flanges.

The local thickness increase preserved structural integrity while reducing material and financial costs compared to increasing the overall body thickness. Table 5 presents the safety coefficients for regions with and without stress concentration, calculated as the ratio between the material strength and the acting equivalent stresses.

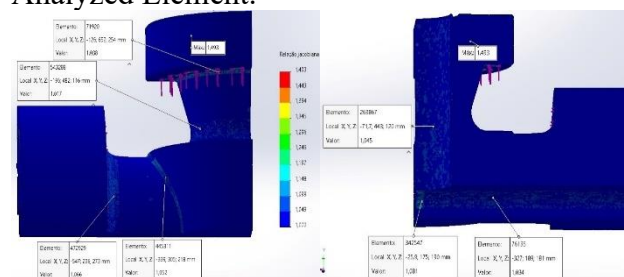
Table 5. Safety Coefficients for Equipment Pressure Conditions.

Load Condition	Radial Stress (MPa)	Circumferential Stress (Mpa)	Axial Stress (MPa)	Von Mises Stress (MPa)	Calculation Method	Safety Coef.
Operation, away from the concentrator	103,4	238,6	76,9	296,4	$\frac{Y_f * S_y}{\sigma_{eq}}$	2,17
Hydrostatic Test away from the concentrator	155,1	330,0	87,4	420,1	$\frac{0,9 * S_y}{\sigma_{eq}}$	1,47
Operation with concentrator	103,4	476,3	21,1	528,6	$\frac{Y_f * S_y}{\sigma_{eq}}$	1,41
Hydrostatic Test with concentrator	155,1	609,4	3,4	698,8	$\frac{0,9 * S_y}{\sigma_{eq}}$	1,11

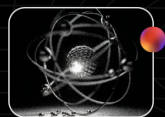
Structural modifications ensured the equipment body met acceptance criteria under internal pressure and secondary loads, being suitable for the proposed operating conditions.

Simplified finite element analysis using 1/8 symmetry and three mesh sizes (20 mm, 5 mm, 2 mm) showed satisfactory mesh quality (Figure 9). The Jacobian criterion, ideally near unity, ranged from 1.0 to 1.16 in critical regions like the internal nozzle surface, nozzle necks, and ram sliding nozzle concentrator, being within the 20% tolerance [63] and validating the mesh quality.

Figure 9. Division and Mesh Quality in the Analyzed Element.



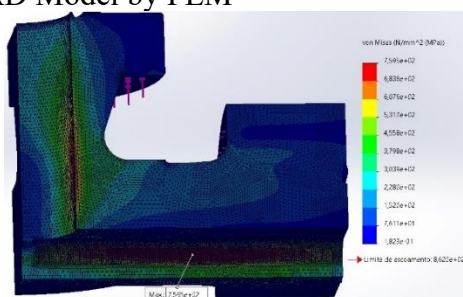
The FEM analysis identified the highest stress concentrations in the thinnest regions and areas with geometric changes, confirming these as critical structural points. Figure 10 presents the maximum stress values under the most



demanding load case, the Hydrostatic Test, which remain below the material's yield strength.

The analysis considered torque and bolt stress values recommended by API 6A for the respective types of flanges and bolts used in the equipment design [35]. Maximum stress zones extended along the flange necks, highlighting significant bending effects. Increasing thickness in these regions is a possible design optimization, to reduce those stresses and improve stiffness.

Figure 10. Von Mises Stress (Hydrostatic Test) on CAD Model by FEM



Based on the FEM results, safety coefficients were calculated for each analyzed scenario using the corresponding Von Mises stress values, as shown in Table 6. Differences between the numerical and analytical model results are attributed to FEM's consideration of complex factors excluded in the pressure vessel calculations, such as detailed geometric features, like angles, surfaces and cuts for ram movement.

Table 6. Safety Coefficients Based on Stresses Obtained by FEM

Load Condition	Von Mises Stress (MPa)	Safety Coefficient
Operation	255,4	2,94
Project	507,6	1,54
Hydrostatic Test	759,5	1,02

Despite increased stress during the hydrostatic test, all safety coefficients remained

within acceptable limits, corroborating the analytical results and indicating the equipment's suitability for the intended operations. However, both analytical and numerical results approached critical values under hydrostatic conditions, suggesting the need for structural optimization, particularly increasing wall thickness at the nozzle necks.

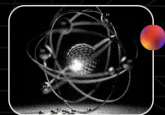
Additionally, membrane stresses on the equipment wall were analysed and compared between analytical and numerical methods using Von Mises stress at average wall thickness. Table 7 presents these stress values and corresponding safety coefficients, calculated as the ratio of membrane stress to the material's allowable stress.

Table 7. Safety Coefficients Based on Membrane Stresses Obtained in FEM.

Type of Analysis	Load Condition	Membrane Stress (MPa)	Safety Coefficient
Analytical	Project	239,5	1,72
	Hydrostatic Test	353,3	1,30
Numerical (FEM)	Project	240,2	1,72
	Hydrostatic Test	363,2	1,27

The stress values obtained for this criterion were within acceptable limits and showed close between the analytical and numerical methods.

Thermal stresses acting on the body under design conditions were also evaluated, combined with pressure effects, to assess combined stresses via FEM. Results, presented in Table 8, remained within acceptance limits, with higher maximum



stresses compared to the pressure-only case due to the additional thermal loading.

Table 8. Combined Stresses Analysis by FEM

Combined Stresses Case	Stress Value (MPa)	Safety Coef.
Von Mises Stresses	549,1	1,43
Maximum Stresses	571,3	1,43

The fatigue life analysis, yielded a safety factor of approximately 0.4, indicating that the equipment has a finite service life, operating up to 42756 cycles. This indicates less operating time than usual for those equipments and a need for earlier replacement. This is a topic that requires further analysis and refinement in later stages of this project, considering a change in materials or the changes in well pressure levels at different stages of its life cycle, since production declines are natural in these structures, and these reduce pressure levels during operation [64].

The design assumptions and simplifications applied in the analyses effectively optimized computational resources and reduced calculation complexity without compromising result quality. The outcomes were consistent with each other and aligned with the theoretical framework, adequately representing the functionality of the modelled mechanisms and the structural integrity of the components.

4. Conclusions

The equipment was sized according to manufacturing and dimensional specifications, utilizing 11" and 13 5/8" API 6BX flanges made from AISI 4145H steel. The objectives were successfully met, with both analytical and finite

element analyses showing compliance within acceptable design limits. In the hydrostatic test, safety coefficients values are close to 1, indicating the need to increase wall thickness, in order to ensure integrity under critical conditions.

As expected, stresses during the hydrostatic test were notably higher than design stresses, due to increased pressure, and stress concentrators significantly amplified these stresses, which justifies careful consideration of this aspect in future design phases.

Fatigue life analysis indicated that the designed EDP valve operates over a finite life cycle, which is not recommended for this type of equipment. Therefore, a more detailed analysis, considering well pressure variations throughout its service life and material optimization, should be performed to achieve more appropriate results.

The development of such equipment is vital for advancing the oil and gas industry, ensuring well integrity and sustained production, particularly at the national level.

Results highlight the importance of accurately determining operating conditions to assess structural integrity, especially for subsea HPHT equipment where pressure is a critical factor influencing stresses.

Another potential improvement includes refining the CAD model with structural optimization and running FEM analysis in other software, as Ansys or Abaqus, to verify and confirm the results.

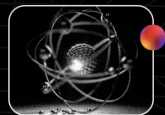


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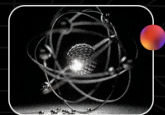
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