COMPARATIVE DESIGN AND ANALYSIS FOR CATENARY RIGID RISERS FOR ULTRA DEEP WATER APPLICATION

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Abstract. *The production riser is a critical component of most offshore petroleum production systems. The riser acts as a physical connection between the subsea wells and floating oil and gas production facility. The transference of production and reservoir injection fluids and other wellhead control commands are exchanged between the subsea system and platform thru the riser. In the literature, many different riser systems have been studied and analyzed for use in ultra deep water applications. The Steel Catenary Riser (SCR) is attractive in terms of ultra deep water depth conditions. This system is comprised of a rigid steel pipe installed in a free catenary shape, and the weight of the riser usually limits its use by requiring larger capacity and much more expensive floating production platforms. Catenary shaped risers with lighter material such as aluminum seem to be an alternative which greatly reduces the riser weight thus allowing the use of smaller floating platforms and ships. The main objective of the present paper is to present and to discuss procedures involving analysis of the operation and design of petroleum production riser systems, and make comparison between catenary riser system with steel pipe and other with lighter material. Systems are analyzed to water depth up to 3000 meters. Hydrodynamic loading due to currents and waves along with floating platform motions are included. Parametric results are presented to identify a feasible solution for very deep water depths with focus on a feasible sytems for pre-salt reservoirs.*

Keywords*: Production Risers, Offshore Petroleum, Sea Current, Waves*

1. INTRODUCTION

The ever increase of global demand for petroleum brings the need for new discoveries of petroleum reserves and new technologies to exploit them. Due to the challenges of petroleum reservoirs in ultra deep waters in pre-salt offshore petroleum fields, offshore productions systems have been studied intensively. Particularly, the study of one of the most critical component on these systems which is the riser that is the physical connection between a subsea oil or gas well and the floating production facility. Riser systems come in many configurations, such as top tensioned riser (TTR), self standing hybrid riser (SSHR), and steel catenary riser (SCR).

The SCR configuration is attractive for use in ultra deep water depths conditions. This system is comprised of a rigid pipe installed in a free catenary shape. The first SCR was installed on the Auger TLP in 1994 (Phifer, 1994). Since this time, SCR applications have been used on TLPs in the Gulf of Mexico (Auger, Mars and Ursa) (Carter and Ronalds, 1998). In Brazil, SCRs were proposed for the submarine production schemes of Petrobras-18 and Petrobras-36 semi-submersibles (Sertã *et al.* 2001). All these applications represent a wide range of water depths between 85 meters and 1650 meters.

For ultra deep water, SCR is a technically feasible and economical configuration. If compared with others riser types, SCR shows advantages. A flexible riser is frequently used as production and injection pipes. They usually present a complex set of layers of different materials along the pipe diameter. Due to high hydrostatic pressures the flexible riser of larger diameters tends to buckle and thus rigid riser become more apt under high pressures.

The increase use of SCRs has created the need to understand their dynamic behavior during its installation and operation. The riser behavior when subjected to extreme environmental sea conditions is critical to design of such systems. Forces resulting from current, waves and floating platform motions combined can cause dangerous levels of stress in the riser and lead to catastrophic failure from material fatigue.

Vortex induced vibration (VIV) has an important influence on riser dynamic behavior and the prediction of the VIV is one of the biggest challenges in the offshore industry (Morooka *et al.* 2007). Another dynamic phenomenon, which influence the dynamic behavior of a riser, is the internal flow of two-phase oil and gas mixtures. The internal flow momentum may impose local displacements along riser's length and thus stress to system (Bordalo *et al.* 2007).

According to Pereira *et al.* (2007), one of the causes of the high stresses at the touch down point (TDP), where the catenary riser comes into contact with the seafloor, is the large curvature of the riser at this point in the riser. The floating production platform motions also have an important influence in stress of the riser. The critical regions of a SCR are the top connection and the TDP regions (Gonzalez *et al.* 2005).

The riser weight usually limits its use by requiring lager capacity and more expensive floating production platforms. To reduce the weight and consequently allow the use of smaller platforms, catenary riser shaped with lighter material such as aluminum will be studied. Foyt *et al.* (2007) and Karunakaran *et al.* (2005) show that a variation in riser weight, caused by modifying density of the coating, improves the dynamic behavior of the riser.

In order to provide a feasible riser configuration the main variables of the riser system have been parametrically modified in this work. The results are from an example of catenary riser in a water depth around 3000 meters. The procedure is based on international technical standards such as API (1998). This example considers extreme environmental conditions acting over the riser and the floating platform.

The present work describes the development of catenary shaped riser for ultra deep water conditions with different riser materials aiming petroleum production from pre-salt reservoirs. The Figure 1 show a schematic view of the riser system considered.

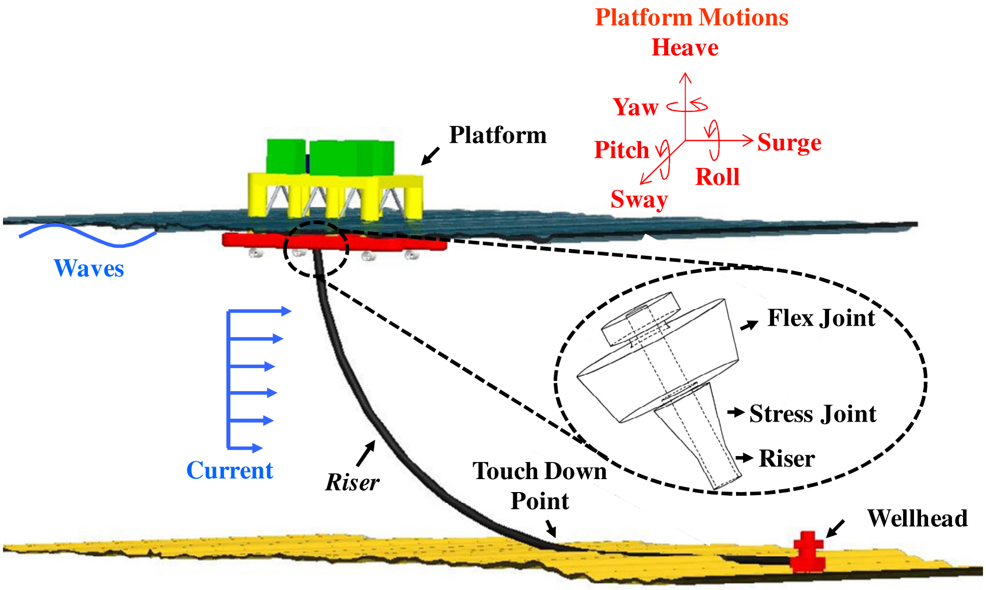

Figure 1. Schematic view of the riser system in the study

2. ANALYSIS PROCEDURE

In the design of a catenary riser it is essential to examine whether the applied design analysis procedure is adequate and acceptable. Different methods have been developed such as the nonlinear or linearized time domain methods and the frequency domain method (Wang *et al.* 2005).

The present work considers applies the nonlinear time domain method. The riser is considered as a co-rotated beam element under lateral loads and the effect of hydrostatic pressures due to internal and external fluids. According to Mourelle *et al.* (1995), the co-rotated beam finite element is the most adequate choice to take into account bending effects and it is a powerful tool nonlinear analysis of the systems under large deformation with good convergence characteristic. The effects of VIV and internal flow were not considered in this work, only the effects of currents, waves and floating production platform motions are considered.

The presented analysis consists of a static and dynamic analysis. Loads acting on the riser are classified as static or dynamics ones. Static loads include dead weight, buoyancy, current and the offset from the floating platform due to current and wind. Only the wave effects and first order motion of the floating platform are considered dynamic loads on riser. For the static behavior of the riser, the governing equations for structure are solved using the Newton-Raphson method. The stiffness matrix is updated at each time step in the case of dynamic analysis and every iteration of the incremental loading in the static analysis (Mourelle *et al.* 1995).

The dynamic analysis consisted in a nonlinear time domain analysis. The Eq. (1) presents the dynamic equilibrium equation which governs the dynamic behavior of the riser.

$$[M]\{\ddot{d}\}+[B]\{\dot{d}\}+[K]\{d\}=\{f\} \tag{1}$$

where, $[M]$ is the mass matrix, $[B]$ is the structural damping matrix, $[K]$ is the riser stiffness matrix, $\{\ddot{d}\}$, $\{\dot{d}\}$ and $\{d\}$ is the acceleration, velocity and displacement vector, respectively, and $\{f\}$ is the load vector.

Modified Newmark-Raphson method is used to solve Eq. (1). This method of solution involves the integration of the Eq. (1) throughout discrete intervals of time (Mourelle *et al.* 1995).

In order to understand the behavior of the riser system in ultra deep water for production of pre-salt reservoirs, analysis of the main variables was conducted using the method already describe. The aim is to obtain a feasible structure when the system is submit to environmental loads, such as current, waves and floating production platform motions.

Two critical regions were focused on, the TDP and the connection with platform. The TDP is critical due to the vessel motions that may cause high levels of stress (Karunakaran *et al.* 2005), and depending on the heave amplitude of the platform compressive force can occur. At the riser top, high stresses may occur due to the riser weight. An increase of the thickness and/or an increase of the length of the stress joint and the presence of a flexible joint may reduce the stress of this region. To study a possible solution aiming to reduce these local stresses, three different configuration of catenary shaped riser with different materials were considered as depicted in the Fig. 2.

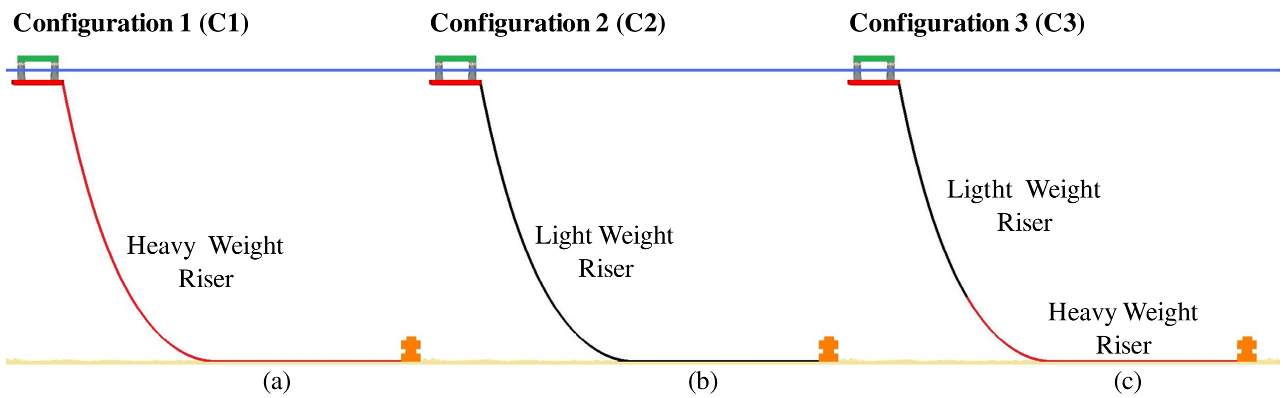

Figure 2. Scheme of catenary shaped riser system with different materials

Typical metaoceanic conditions present of the coast of Brazil have been taken into account. Motions and offset of the floating platform, variation of the current profile and its direction along the water depth were also considered. Six degrees of freedom are assumed for the floating platform motions. A transfer function was used to relate the platform motions to the waves. Table 1 presents the main parameters for the analysis of the riser schematically showed in Fig. 1.

The determination of the riser top angle, which is the angle between the vertical line and the axial direction of the riser, is one of the most important issues for catenary shaped riser design. For the catenary riser, the riser top angle, as well as the weight, defines the riser's curvature at the TDP. The curvature radius for rigid steel riser should not be less than 70 meters in any section of the riser (Pereira *et al.* 2007).

The flexible joint acts as a physical connection between the riser and the platform and it has the function to reduce the stress caused by bending and torsion. Flexible riser is comprised by several plastic and metallic materials layers that allow flexibility while maintaining structural integrity. To avoid stress concentration in this connection, a stress joint is applied. It is a connection between the flexible joint and the riser and has a varying external diameter, ranging between the external riser diameter to the flexible joint diameter (API RP 2RD, 1998). The length of the stress joint can vary depending on the particular application.

Table 1. Main parameters of the riser system

Parameter		Value
Water Depth		3000 m
External Diameter		0.2106 m
Internal Diameter		0.1520 m
Platform Offset		140 m
Current Velocity (Top - Bottom)		$1.25\ m\ s^{-1}$ - $0.0\ m \cdot s^{-1}$
Wave (Height - Period)		11.2 m - 12.7 s
Flex Joint	X, Y and Z stiffness	$2.89 \times 10^{4}\ kN \cdot m^{-1}$
	RX, RY and RZ stiffness	$9.6\ (kN \cdot m^{-1}) \cdot degree^{-1}$

The riser's thickness is another very important design parameter and its value has influence in the static and dynamic behavior. An increase in the thickness implies higher weights and consequently induces a higher tension and stress at the top end of the system. According to API RP 2RD (1998), the lower limit of the thickness is defined by the hydrostatic collapse stress, which considers the difference between the external pressure and internal pressure. Equation (2) shows the criteria.

$$P_a \leq D_f \cdot P_c \tag{2}$$

where, P_a is the external pressure, P_c is the resistant pressure, which depends on the riser geometry, and D_f is the design factor. Depending on the riser manufacturing process D_f is 0.60 or 0.75.

The design criteria adopted in this work is the von Mises stress as in Eq. (3). The von Mises stress maximum along the entire riser should not exceed the allowable stress, defined by the right hand side of Eq. (4) (API RP 2RD, 1998).

$$\sigma_e = \frac{1}{\sqrt{2}}\sqrt{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_1-\sigma_3)^2} \tag{3}$$

$$\sigma_e \leq C_f \cdot C_a \cdot \sigma_y \tag{4}$$

where, σ_e is the von Mises stress, σ_1 , σ_2 and σ_3 are the principal stresses, C_f is the design case factor, which is equal to 1.2 to extreme conditions, C_a is the allowable stress factor equal to 2/3 and σ_y is the yield strength of the material.

3. RESULTS AND DISCUSSION

The results are based on a parametric analysis of a typical catenary riser as showed in Fig. 1. Configuration 1 (C1) and 2 (C2), as showed in Fig. 2 are studied with the main objective of reduction of the tension of the top connection with the platform and the compression at the TDP region and the stress of both critical regions. The main parameters of the configurations are presented in Tab. 1. After the results and analysis of these two configurations, a practical example is performed comparing the three configurations present in Fig. 2.

The properties of these different materials were present in Tab. 2. This study utilizes a typical Brazilian sea conditions. The current direction is 0° with the heading of the riser and the wave is 45°. The length of the risers was present in Tab. 3. The riser direction heading is equal to 0°. Numerical software has been used for calculations (Mourelle *et al.* 1995).

Table 2. Material properties for the catenary riser

Riser's Material	Young Modulus [GPa]	Specific Weight [kN.m^{-3}]	Yield Strength [MPa]	Allowable Stress [MPa]
Material A (Heavy Weight Riser)	208	77.0	414	331.2
Material B (Light Weight Riser)	70	27.7	260	208.0

Table 3. Total length of the riser for each configuration

Configuration	Length		
	Material A [m]	Material B [m]	Total [m]
1	9600	0	
2	0	9600	9600
3	2200	7400	

The length of the stress joint was also investigated, assuming values between 1 to 5 meters with increments of 0.5 meters. Values larger than 2.5 meters are not reported due to the lack of great improvement for stresses reduction for these larger lengths. Therefore, the results ahead are considering a stress joint with 2.5 meters.

The hydrostatic collapse, considering the hoop stress analysis, pointed that the 0.058 meters wall thickness, for both materials, meets resistance criteria, when the riser of external diameter equal to 0.2106 meters.

Six different analyses were performed, assuming angles of 11° to 21° with the increments of 2° and for two configurations (Fig. 2). The tension at the riser top is composed by a vertical and horizontal force. Results for different riser top angles, for different configurations are shown in Fig. 3. In Fig. 3a, von Mises stress at the TDP region is shown. Lowest stress is achieved with riser top angle equal 15° and 11°, for C1 and C2, respectively. Figure 3b show the results at the riser top region. Due to the light material, von Mises stress at the riser top region was not critical. However, the configuration with heavy material present von Mises Stress at the top of the riser higher than the allowable stress. The practical example presented considers the riser top angle equal 21°. Even though the preliminary von Mises stress results are above allowable stress. Based on these results, the axial force is predominate in the von Mises stress calculation.

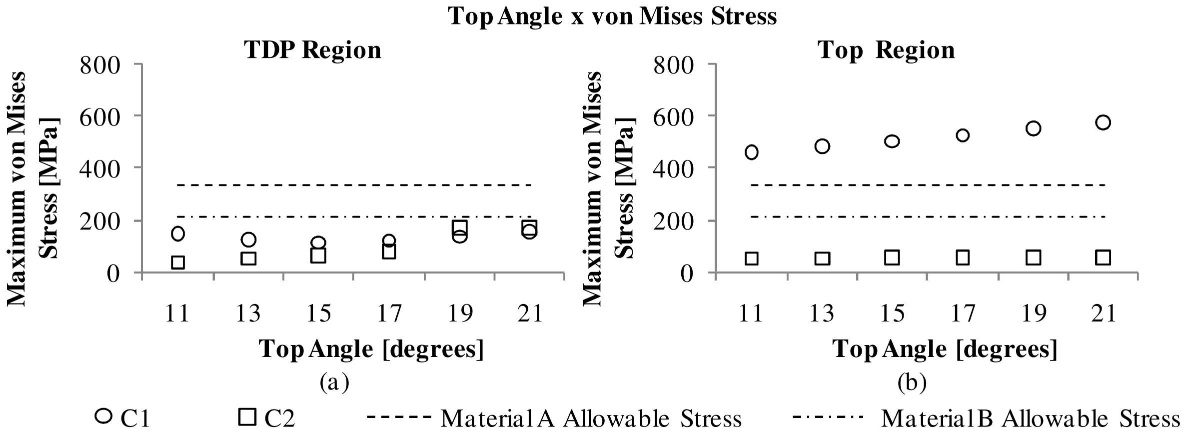

Figure 3. (a) Riser Top Angle vs. von Mises Stress for the TDP Region, and (b) Riser Top Angle vs. von Mises Stress for the Riser Top Region

In order to determine the influence of the platform motion on the dynamic behavior of the riser, a parametric study varying the amplitude of the motion was conducted. The most critical motion is the heave platform motion. This movement induces higher stress and compression of the TDP region. Figure 4 show the results varying the amplitude of the platform heave motion, ranging from zero to five meters for the amplitude. Figure 4a presents the maximum von Mises stress at the TDP region and Fig. 4b presents the axial force. For low values of amplitudes, less than 1.5 meters, the whole riser of C2 is tensioned, even the TDP region, and values less than 4.5 meters for riser of C1.

Compression forces can cause buckling on the riser and thus, these forces must be avoided in riser design. From Fig. 4a, it is clear that increase the heave platform amplitude the von Mises stress at the TDP region increase and the magnitude of this stress was near of the allowable limits.

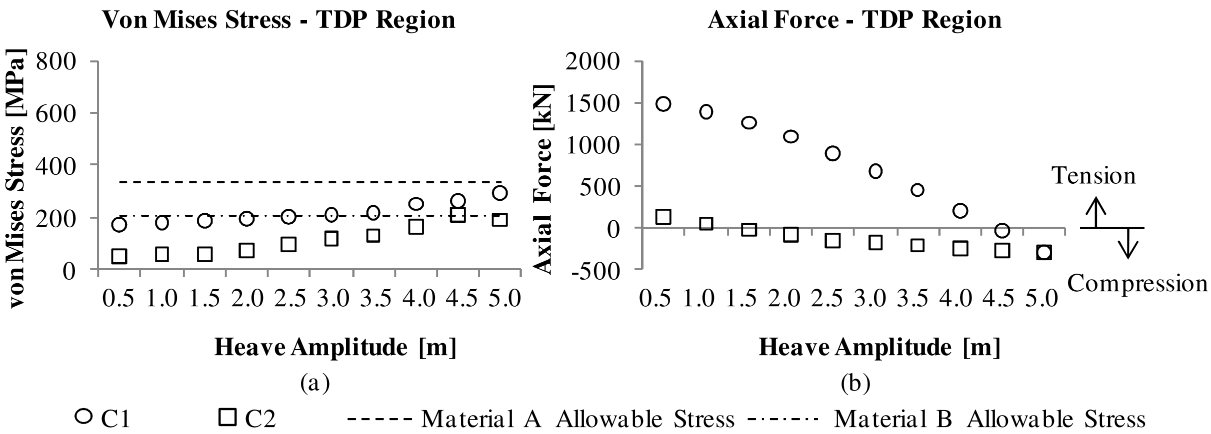

Figure 4. Effects of platform heave motion amplitude on (a) riser's stress and (b) axial force at the TDP region

The effects of the heave motion frequency on the dynamic behavior of the riser were present in Fig. 5. A parametric study varying the frequency of the heave motion was conducted and the results were focused on TDP region. The amplitude of the heave motion for each frequency was present in Tab. 4. Figure 5a presents von Mises stress when the platform heave motion is varying and Fig. 5b present the axial force results. It can be observed that the frequency equal a 0.04 Hz present the most critical combination of frequency and amplitude.

Since the riser top and the TDP are the critical regions for the catenary riser, a new configuration was proposed (Configuration 3), as show in Fig. 2. This configuration tried to eliminate the high tension at the top of the system and the high stress at the TDP. To reduce the tension at the riser top, a light riser material was used for most of the suspended length. A heavy material with higher resistance (yield strength) was utilized near the TDP region. The results of the three configurations are then compared.

Table 4. Heave motion amplitude for each heave motion frequency

Frequency [Hz]	Amplitude [m]
0.03	6.6
0.04	8.3
0.05	2.4
0.06	3.5
0.07	3.5
0.08	3.5

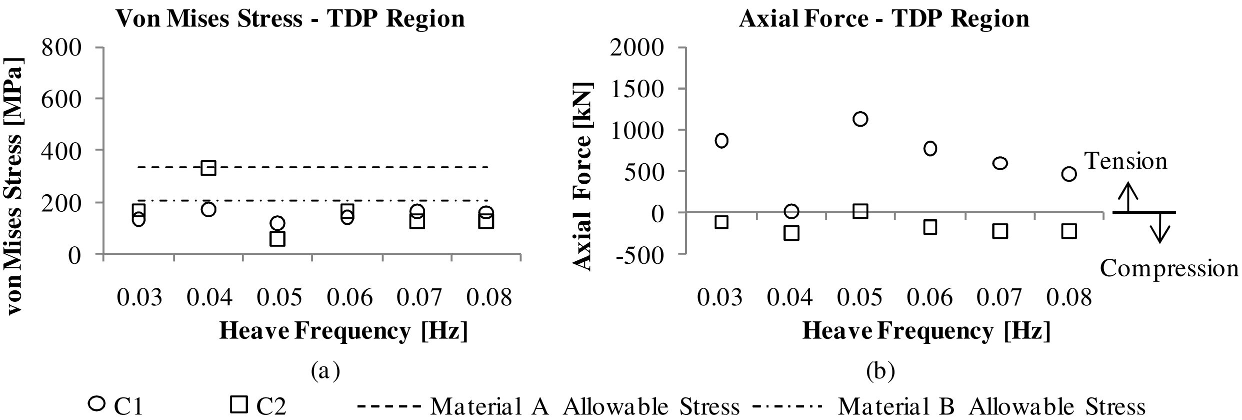

Figure 5. Effects of platform heave motion frequency on (a) riser's stress and (b) axial force at the TDP region

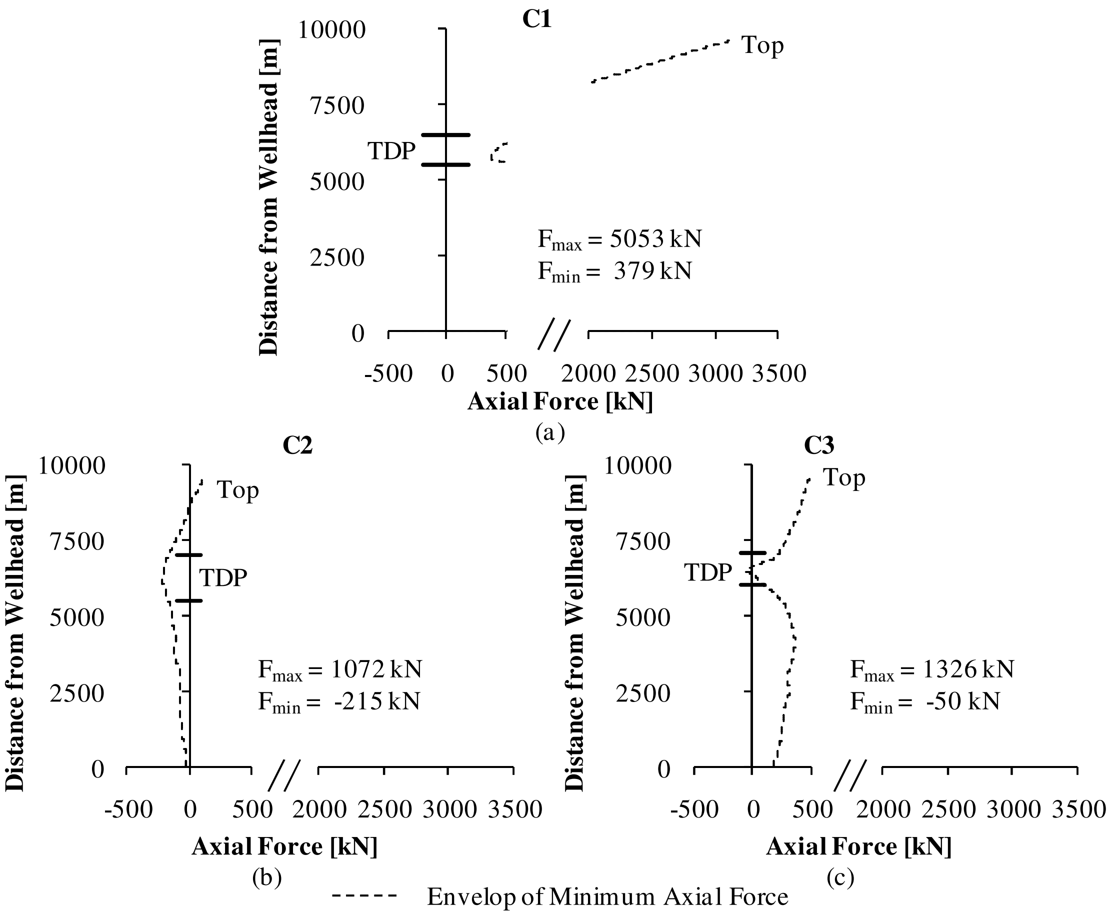

Figure 6. Envelop of minimum axial force of different configurations

In Figure 6 envelop of minimum axial force for each configuration is shown and the values of the maximum (tension at the riser top) and minimum (compression at the TDP) are indicated. The maximum tension force occurs at the connection, in the top end of the riser. As expected, it can be observed that the tension at the riser top reduces significantly. However, C2 present compression in a long section of the riser. C3 seems to be a good alternative, take into account only the axial force. It can be observed that C3 reduce the tension at the riser top if comparing with C1 and the compression reduces compared with C2.

Figure 7 present the results of von Mises stress for the three configurations. It shows that modifying the weight of the riser has great influence on von Mises stress. It is clear, from Fig. 7a that top region of the system is most critical for C1 and do not present high von Mises stress at the TDP region. Figure 7b and 7c show that C2 and C3, due to the lower weight, seem not to be critical at the riser top. C3 present the highest von Mises stress, however, it is less than the material A allowable von Mises stress.

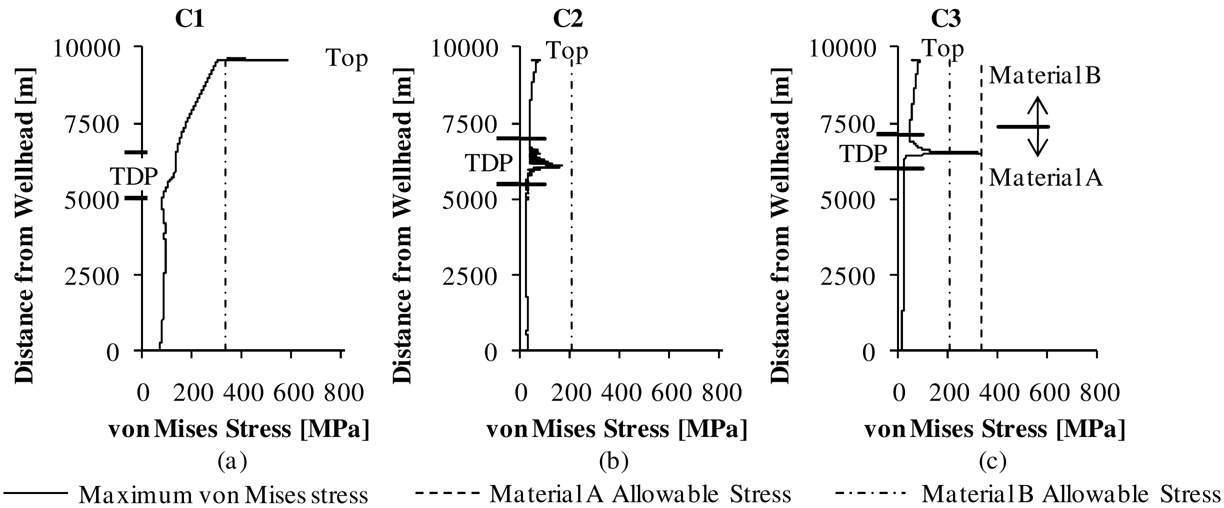

Figure 7. Maximum von Mises stress of different configurations

Results from parametric analysis have been applied to determining the riser configuration, as previously described. This work presents the results for only one direction of incidence of current and waves. A complete analysis should take into account eight directions of incident of current and wave and then combining all the direction.

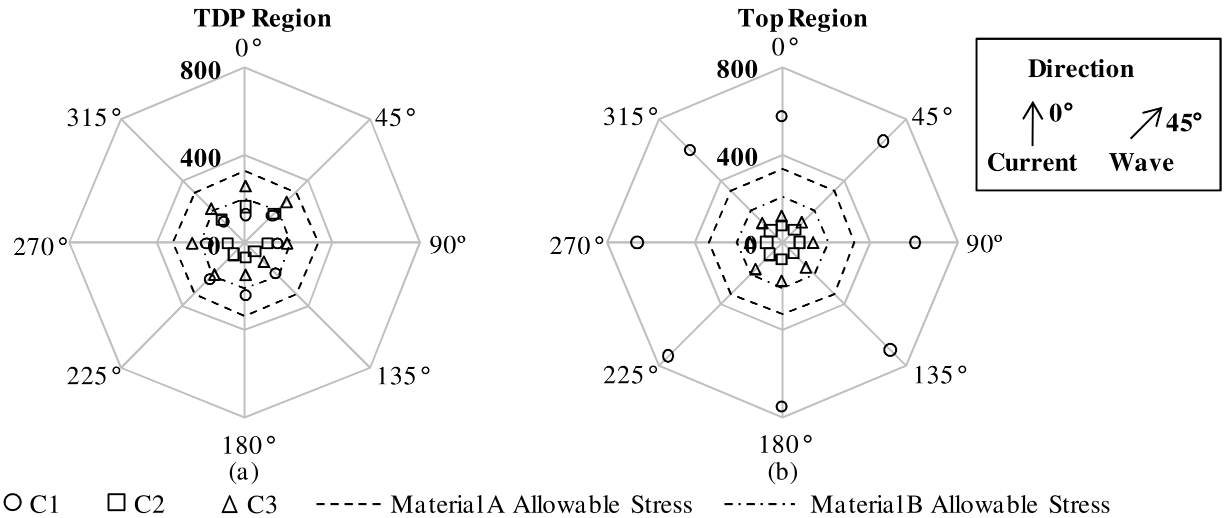

Figure 8. Maximum von Mises stress for different heading direction

Eight distinct headings were considered and the results are shown in terms of maximum von Mises stress for each riser heading direction is present in Fig. 8. Figure 8a shows maximum von Mises stress at the TDP region and it can be observed that for this region all the cases seem to be under the allowable stress limit. However, from Fig. 8b, von Mises stress for C1 was over the allowable. The direction of the current is 0° and the direction of the wave is 45°. This combination of direction, not aligned, is one of the most critical combinations in terms of von Mises stress and compression force in TDP region.

Due to the platform offset, the risers assume near, mean or far configuration. Risers in near configuration have lower curvature radius at the TDP region, favoring the occurrence of compressive forces. Figure 9 present the compression force. It is important to notice that the case here analyzed the direction 0° was in a near configuration and thus 180° was in a far configuration. It can be observed that risers 0°, 45° and 315° present the highest compression force for C2 and C3.

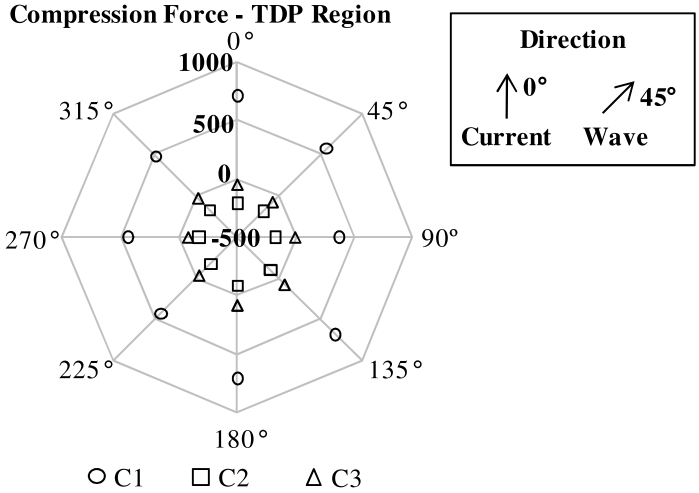

Figure 9. Compression force for different heading direction

4. CONCLUSIONS

The present work presents a comparative analysis results for operation and design of a petroleum riser system. The study has been focused on the feasibility of the system for different riser geometries and materials, aiming the operation in offshore ultra deep water depth for pre-salt reservoirs petroleum production. Parametric analyses were conducted varying some of the main parameters of the catenary riser system.

From the analysis, the hydrostatic collapse stress seems not to be a critical for rigid riser. An optimal combination of riser top angle and some parameter of stress joint can provide lower stress at the top of the system. The heave motion of the platform can cause in the TDP higher stresses and compression of the riser and depending on the heave amplitude the compression disappears and the stress reduces to allowable levels. Therefore, floating platform with low vertical heave motion is desirable for catenary riser application.

A combination of different material along the riser seems to be attractive. This combination provides a reduction of the tension at the top and the von Mises stress appears not to be critical.

For a complete analysis a fatigue analysis should be conducted in near future to obtain the service life of the riser. This is a very important analysis for the design of the riser.

5. ACKNOWLEDGEMENTS

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