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RISER TECHNOLOGY AND INDUSTRY-UNIVERSITY COOPERATION. PART I - RISER MECHANICS

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ABSTRACT

This is the first of two companion papers intending to summarize and pointing out some of the main engineering achievements obtained through a continuous cooperative research effort, jointly conducted by Industry and University. Part I covers Riser Mechanics and design issues. Part II is dedicated to Riser Loading and Dynamic Positioning.

The work starts with the first SCR (Steel Catenary Riser) concept and engineering analysis that paved the road toward a mature and reliable design which has been applied with success in floating production in Campos Basin, offshore Brazil. Several computer codes, numerical methods and analytical tools were developed and verified based on experimental results.

In Part I, SCR dynamics, touch down point dynamic interaction loading, catenary riser dynamic compression under twist, as well as risers and umbilical cables structural mechanics, are treated and modeled through analytical and numerical methods. SCR design issues and fatigue assessment of flexible pipes are also addressed. In Part II [1] , VIV (Vortex-Induced Vibration), extreme dynamic loading, nonlinear dynamics of FPSOs and turret positioning are presented. Nonlinear control techniques are also addressed aiming to optimize FPSO heading, constrained to a cost function that takes into account risers loading, mooring tension, roll motion and fuel consumption.

INTRODUCTION

Those two companion papers present an overview of the technological achievements resulting from the technical cooperation between the Subsea Engineering Group Exploration and Production and CENPES, Petrobras, and the Departments of Ocean and Mechanical Engineering of the University of São Paulo.

This partnership started in early nineties when Petrobras started to face its first deepwater riser challenge. The first phase of Marlim project included the semi-submersible Petrobras XVIII, which demanded an export riser system for 910 meters water depth. At that time there was no available technology to produce large diameter flexible risers. Petrobras then signed a technical cooperation agreement with a major flexible pipe company to develop such risers and at the same time, started to investigate the SCR concept.

Since then several tools have been developed and feasibility studies have been conducted, focusing on riser design. Among the topics comprised in Riser Mechanics are: dynamic curvature and its variation near the touch down point and its role in fatigue analysis; structural VIV effect in SCR; dynamic tension and non-linear statistics based on random waves; dynamic compression analysis under twist; effects of soil elasticity; development of software to design SCR and fatigue model analysis of flexible pipes. Part I is aimed to summarize such achieved results.

Vortex-Induced Vibration of offshore structures, mostly applied to risers, has been a keen research issue. Nonlinear dynamics of FPSOs and turret positioning concerning riser integrity have also been studied. Nonlinear control techniques have been implemented aiming to optimize FPSO heading, constrained to a cost function that takes into account risers loading, mooring tension, roll motion and fuel consumption. Those are the issues presented in Part II [1] .

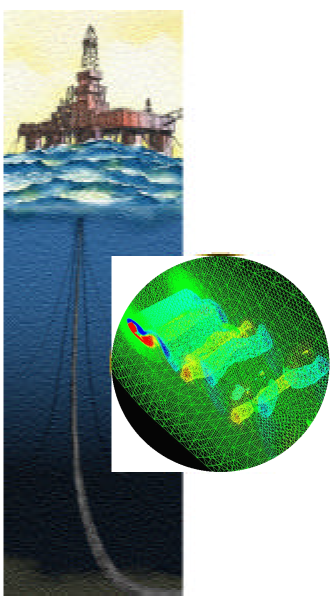
This overview intends to present some insights of these technologies that helped to boost deepwater riser engineering, establishing feasible and cost-effective alternatives.

RISERS

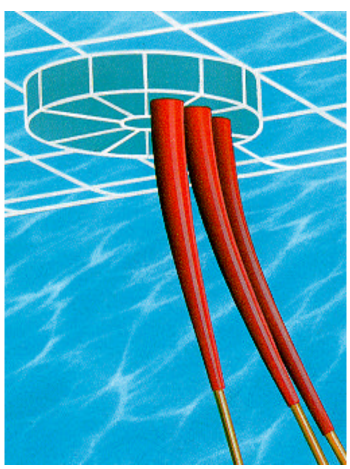
Risers are pipes that connect the floating production structure to the sea floor. The primary goal of these structures is to transport the oil produced at the Christmas tree installed on the wellhead at seabed to the floating unit.

In addition to the transport of oil to the surface, there is a need to control subsea valves, inject gas or water in reservoir, transfer oil or gas from one floating system to another, and so on. It means that there are several types of risers, which are designed to meet the specifications of a particularly application. These structures are typically tailor-made or fit for the purpose intended.

A worldwide trend in offshore oil industry is to use the most simple and cost-effective system to connect the offshore production units with the seabed. Such system is a catenary riser. At the same time, development of hydrocarbon in deeper waters made the use of floating systems more common. Therefore, such trend implied in the growth of very large fields for development and initiated research activities on catenary risers, mainly related to the dynamic problem. Next sections will define this riser problem, point out some results achieved and theories developed.



(a)



(b)

Fig. 1 (a) A Floating production System (of the Semi-submersible platform type) and a bundle of catenary risers.
(b) Top end-fits of the bending stiffener type.

THE DYNAMIC PROBLEM OF A CATENARY RISER

A *catenary riser* is a free-hanging riser with no intermediate buoys or floating devices. This kind of ocean system is subject to dynamic loads caused by the action of current, and waves as well as excited by the FPS (Floating Production System) motions, imposed to the top extremity (top end-fitting). This makes a consistent treatment of the problem extremely difficult. Figure 1 shows a view of such a dynamic system together with top end fitting adapted to typical bending stiffeners.

Current induces static and dynamic loads, including high amplitude Vortex-Induced Vibrations. The direct action of sea waves causes oscillatory loads, in a wide range of frequencies, but restricted to a region near the surface. However, motions imposed to the top connection of the riser by FPS cause the most important loads. The vessel oscillates in two different frequency ranges. The first one is the same range that characterizes the surface waves. The second one is associated to the well-known slow-drift motions. These two actions make the riser to move, displacing dynamically the Touch Down Point (TDP) on the seabed.

There are at least three different time-scales in the catenary riser problem; see, e.g. Triantafyllou et al. (1985) [2] . The first and shortest one is dominated by axial rigidity giving rise to relatively small periods of oscillation. The second one is related to the geometric or catenary rigidity. The third one is of a local nature and is due to the local flexural rigidity effects.

Such a diversity of time-scales can lead to serious limitations concerning numerical integration methods by rendering dynamic equations mathematically stiff. On the other hand, it enables asymptotic approaches to be rather effective. As an example, the flexural rigidity effect is confined and dominant just inside small regions close to the ends, the TDP and the upper end-fitting, enabling to treat the global dynamic problem under the ideal cable equation in the frequency-domain, by standard perturbation techniques.

The local non-linear effects, due to geometric non-linear boundary conditions in the TDP region, can then be included by means of a standard boundary-layer technique, giving a consistent non-linear analytical solution for the curvature problem; Aranha et al. (1997) [3] [1]. This type of results have been experimentally confirmed; Pesce (1997) [5] ; see also Pesce et al. (1998) [6] . The boundary-layer technique can be applied as well for the local solution in the upper connection region, or else to incorporate the effect of soil elasticity in the TDP region. Another kind of asymptotic solution can be constructed, for the dynamic

[1] Similar problems, concerning stresses in cables, have been treated under the same approach; see, e.g. Irvine (1992) [4] .

tension response, making use of the disparity between the first and the second time-scales. In this latter analytical solution the dynamic tension response to a harmonic excitation applied to the top, in the presence of waves and current, has been determined and confirmed experimentally for a full range of exciting frequencies; Aranha et al. (1993) [7] , Aranha & Pinto (2000) [8] .

Another interesting issue is the eigenvalue problem for a catenary riser, particularly pertinent for VIV analysis. An analytical and closed form JWKB approximate solution, previously derived by Triantafyllou (1984) [9] , was applied to the catenary riser problem, given the tension function along the length; Pesce et al. (1999) [10] The asymptotic solution is found to compare well with numerical results obtained with *PoliFlex*, a code in frequency-domain, developed using a standard Finite Element method.

Another fundamental analytical result concerns dynamic compression of risers under curvature (Aranha et al, 2001) [11] , including twist Ramos (2001) [12] and Ramos and Pesce (2001) [13] , and is also addressed in this paper.

These results have been efficiently implemented in practical design codes. Restricted to riser mechanics issue this first present paper aims to provide an overview on this methodology, highlighting some major results obtained from the analytical methods.

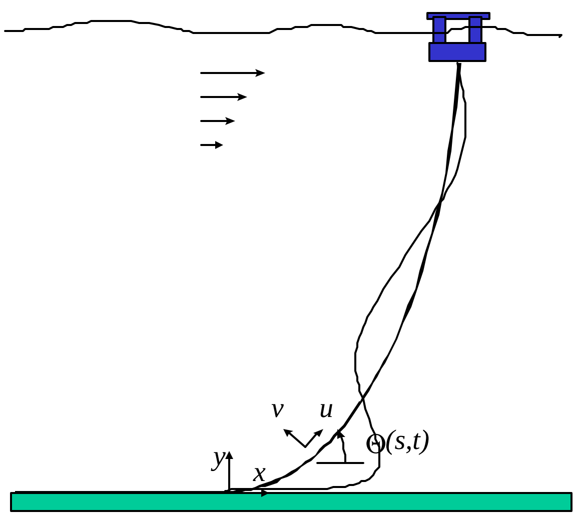


Fig. 2 The two-dimensional 'Catenary" Riser problem.

LOCAL ANALYTICAL SOLUTIONS

The two-dimensional *catenary riser* problem is considered herein, dynamically formulated, under the usual small perturbation theory, around the static equilibrium configuration. The riser is suspended from a floating unit and hangs freely down to the seabed, under the action of current and waves. Boundary conditions at the upper end are supposed to be specified properly, considering the offset and motions of the floating unit and, possibly, bending stiffener effects. At TDP the boundary condition is of the one-side type. The soil is usually supposed to be infinitely rigid. Nevertheless, the effect of finite rigidity could be easily incorporated, Pesce et al., 1998 [14] .

Sub-critical dynamic boundary-layer solution in the TDP region. Rigid Soil.

Under the ideal cable assumption, it was shown by Triantafyllou et al. (1985) [2] , and after, reinterpreted by Aranha et al. (1997) [3] that a shock condition in which the cable impacts the soil will hold whenever the local 'Mach' number $\mathrm{M} = \dot{x}_0 / c_0 \geq 1$. We observe that $x_0(t)$ is the function describing the instantaneous position of the TDP, $c_0 = \sqrt{T_0/(m+m_a)}$ is the local cable wave celerity and T_0 is the local static tension. This assertion can be physically interpreted as the lack of time for the cable to adjust its geometric form properly. Conversely, the condition $\mathrm{M} < 1$ is said to be a *sub-critical dynamic regime.* For this latter situation, it has also been shown that *the inertia term can be locally disregarded*, with an error of the order of $\mathrm{M}^2 << 1$. Moreover, the boundary condition for the global sub-critical quasi-linear equation can be taken as a simple hinge, placed at the cable static TDP with an error of order $O\left(a_0 c_0\right)^2$, where a_0 is the typical amplitude of $x_0(t)$, $c_0 = q/T_0$ is the local curvature, and q is the immersed weight per unit length.

Considering the global quasi-linear dynamic solution for a cable, where $t(s,t)$ is the dynamic tension, and recalling (Aranha et al., 1993) [7] that the dynamic tension is weakly varying along the arc length, at least close to the TDP, a boundary-layer solution is found, Aranha et al. (1997), with an error of order $O(\mathrm{M}^2 << 1)$, as

$$\frac{c(s,t;\, l)}{c_0} = \begin{cases} \dfrac{1}{1+t(t)/T_0}\left(1-\exp\left(-b(s,t;\, l)\right)\right); \ b(s,t;\, l) > 0 \\ 0; \ b(s,t;\, l) \leq 0 \end{cases} \tag{1}$$

$$b(s,t;\, l) = \frac{\sqrt{1+t(t)/T_0}}{l}\left(s - s_f(t)\right) = \sqrt{1+\frac{t(t)}{T_0}}\left(\frac{s}{l} - \frac{x_0(t)}{l}\right) + 1$$

In equation (1) $l = \sqrt{EI/T_0}$ is the *local flexural length scale*, gauging the bending stiffness (*EI*) boundary layer. All global static information, including the global effect of current, is contained in the static parameter T_0, so in $c_0 = q/T_0$, which approximates the local static curvature. All global dynamic information is locally represented by the functions $t(t)$ *and* $x_0(t)$. In equation (1), $s_f(t) = x_0(t) - l\big/\sqrt{1+t(t)/T_0}$ is the actual TDP instantaneous position, including flexural rigidity effects and measured from the static 'ideal cable' TDP, which has been conveniently chosen as the origin for the coordinate

s. The non-linear boundary-layer solution does not depend on the particular form taken by the oscillatory (global) functions $t(t)$ and $x_0(t)$ and can be easily applied to the riser's response analysis under random excitation.

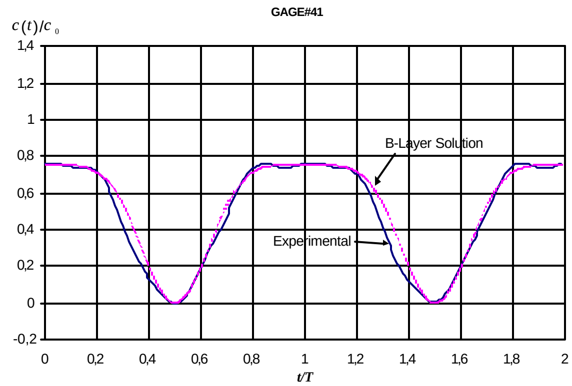


Fig. 3 Nondimensional curvature $c(t)/c_0$. "Critical section": (Pesce et al., 1998 [6]). $c_0 = q/T_0 = 0{,}0409 m^{-1}$; $a_0/l = 1{,}1619$; $s/l = 0{,}0435$; $t_0/T_0 = 0{,}2004$; $j = p$; $M = 0{,}055$.

The analytical boundary-layer solution for the curvature was verified experimentally, as well. A rod, properly scaling the boundary layer region of a SCR, was instrumented with an array of 100 strain-gages. Figures 3 and 4, [6] , show examples comparing asymptotic solutions with experimental results. Figure 3 presents curvature as function of time for the critical section, where the curvature variation attains a maximum value. Figure 4 shows RMS values of curvature along the boundary layer. The agreement is rather satisfactory.

Figures 5 and 6, [3] , show the comparison between the TDP boundary-layer solution with the results obtained with a fully non-linear code, for a 16" diameter SCR, in 575m deep water. Only a circular motion was imposed to the top end, with amplitude of 0.2m. The period of excitation was 7.8s. This condition corresponds to the motion of a semi-submersible platform in a typical sea-state in Campos Basin. The agreement is very good, indeed. A *clockwise* or a *counter-clockwise* motion led to different results, particularly for the RMS curve. The cause for this is the variation occurring in the dynamic tension $t(t)$ and in the TDP excursion $x_0(t)$, depending on the sense of rotation imposed to the top.

Note that the quasi-linear global dynamic solution is iteratively solved in frequency-domain due to the quadratic viscous damping which is the only source of non-linearities in the quasi-linear global dynamic equation.

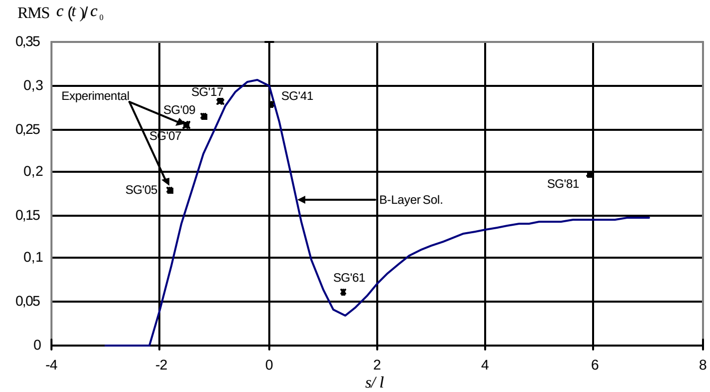


Fig. 4 RMS values for the nondimensional curvature $c(t)/c_0$ along the rod; (Pesce et al., 1998 [6]).
$c_0 = q/T_0 = 0{,}0409 m^{-1}$; $a_0/l = 1{,}1619$;
$t_0/T_0 = 0{,}2004$; $j = p$; $M = 0{,}055$

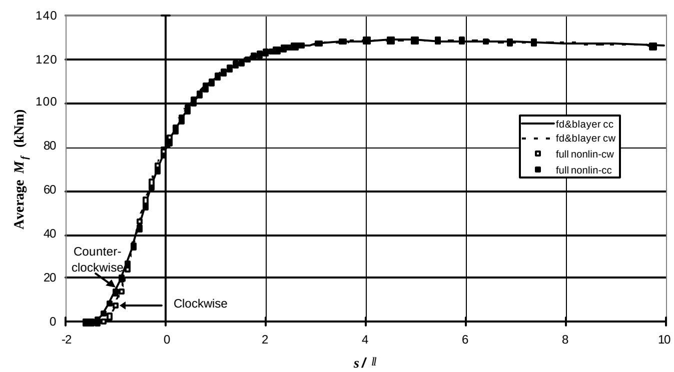


Fig. 5 Average value of the moment in the vicinity of the touchdown point for a 16" SCR. Legend: fd: frequency domain; blayer: boundary-layer solution; full nonlin: fully nonlinear time-domain simulation. (Aranha et al. 1997 [3])

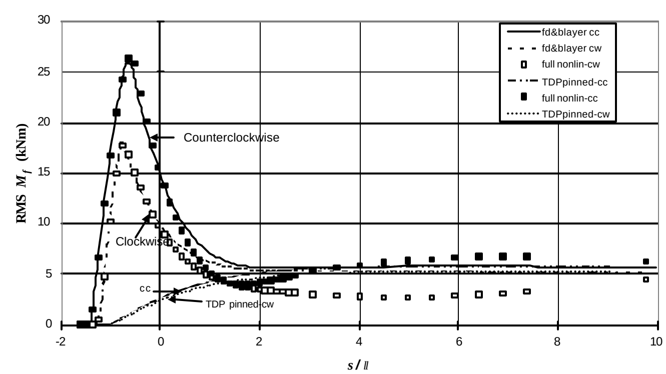


Fig. 6 RMS value of the moment in the vicinity of the touchdown point for a 16" SCR. Legend: fd: frequency domain; blayer: boundary-layer solution; full nonlin: fully nonlinear time-domain simulation. (Aranha et al., 1997 [3])

The boundary-layer solution at the top end

Using a similar reasoning, Pesce (1997) [5] , Martins (2000) [15] , applied to the top end lead to the following boundary-layer solution for the curvature at the upper region,

$$\boldsymbol{c}(s,t) = \boldsymbol{c}^{c}(s,t) -$$

$$-\frac{1}{\boldsymbol{l}_L}\left(\boldsymbol{c}_L^{c}(t)\,\boldsymbol{l}_L + \frac{k_F\,\boldsymbol{l}_L}{B}\left(\boldsymbol{Q}_L(t) - \boldsymbol{F}(t)\right)\right) e^{\left(\sqrt{1+\frac{\boldsymbol{t}(t)}{T_L}}\frac{s-L}{\boldsymbol{l}_L}\right)} \qquad (2).$$

Where,

$$\boldsymbol{Q}_L(t) = \frac{\boldsymbol{Q}_L^{c}(t) + \frac{1}{\sqrt{1+\boldsymbol{t}(t)/T_L}}\left(\frac{k_F\,\boldsymbol{l}_L}{B}\boldsymbol{F}(t) - \boldsymbol{c}_L^{c}(t)\,\boldsymbol{l}_L\right)}{1 + \frac{k_F\,\boldsymbol{l}_L}{B}\frac{1}{\sqrt{1+\boldsymbol{t}(t)/T_L}}} \qquad (3).$$

In equations (3)-(4): $\boldsymbol{l}_L = \sqrt{B/T_L}$ is the local 'flexural length'; T_L is the static tension; $\boldsymbol{q}_L$ is the static angle at the upper end; $c^{c}(s,t)$ is the dynamic ideal cable curvature; $\boldsymbol{F}(t)$ is the angle, with respect to the horizontal, imposed by the floating unit to the local cable axis; and k_F is the flexural rigidity for a unitary angle rotation corresponding to a *linearly* flexible top connection.

We should expect some non-linear behaviour only for large dynamic tension amplitude values. The static solution can be readily determined from the above equations, as a particular case. Also, important considerations, concerning structural and flex-joint design, can be promptly derived; see Pesce (1997) [5] .

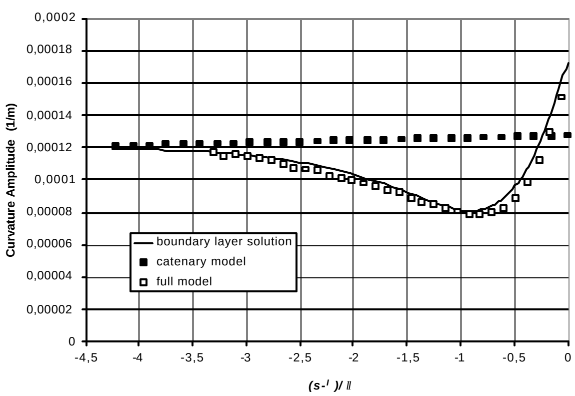


Fig. 7 Dynamic amplitude of curvature at top, for a SCR $k_f = 10\text{kN/grau};\ \boldsymbol{l}_L = 4.60\,\text{m};\ L = 970\text{m}$. Asymptotics compared to a full nonlinear code (ORCAFLEX); Martins et al. (2000) [16] .

Figure 7 presents the dynamic amplitude of curvature, comparing the boundary-layer analytical solution with a full non-linear numerical solution obtained with a numerical code. The agreement is very good, indeed. For reference sake the curve corresponding to the ideal catenary is also presented. It is clear that the flexural boundary layer has a length of about $4\,l_L$, 18.4 meters in this case.

THE CATENARY RISER EIGEN-VALUE PROBLEM

Another very interesting and important problem that can be treated under asymptotic methods is the eigen-value problem, which has been addressed in the riser scenario by Pesce et al. (1999) [10] , following the work by Triantafyllou (1984). Let *L* be the suspended length. Under the inextensible hypothesis, a JWKB solution for the catenary eigen-value problem can be deduced and given in non-dimensional form as,

$$\boldsymbol{j}(\boldsymbol{z}) \cong F^{-1/4}(\boldsymbol{z})\left[C_1 \sin\left(\boldsymbol{L}\int^{z} F^{-1/2}(u)du\right) + C_2\cos\left(\boldsymbol{L}\int^{z} F^{-1/2}(u)du\right)\right] \quad (4)$$

where $\boldsymbol{j}(\boldsymbol{z})$ is the transverse displacement[2], for each particular natural frequency $\boldsymbol{w}$. In (4), $\boldsymbol{L} = \frac{\boldsymbol{w}}{\tan\boldsymbol{q}_L}$ and

$$F(\boldsymbol{x}) = \frac{T(\boldsymbol{x})}{T_0} \qquad (5),$$

is the nondimensional static tension function, being $\boldsymbol{z} = \tan\boldsymbol{q}(\boldsymbol{x})$ a convenient measure of the nondimensional arch-length $\boldsymbol{x} = s/L$. For an ideal catenary (no current loads), $F_c(\boldsymbol{z}) = \sqrt{1+\boldsymbol{z}^2}$, where *c* stands for an ideal catenary. This curve is almost linear, also suggesting a Bessel's function solution as a first approximation; see also [10] .

Equation (5) gives a general closed form solution for the inextensible tensioned-and-curved-heavy-string problem (not only for the riser-like problem but also for jumpers, for instance). Note that eigenmodes are sinusoidal functions, modulated in phase and amplitude and resembling Bessel's functions.

In the case of a *free-hanging catenary riser* with a touchdown point, $F_c(\boldsymbol{z}) = \sqrt{1+\boldsymbol{z}^2}$, and so

$$\boldsymbol{j}_n(\boldsymbol{q};\boldsymbol{q}_L) \cong A_n(\cos\boldsymbol{q})^{1/4}\sin\left\{\boldsymbol{L}_n\int_0^{q_L}\frac{d\boldsymbol{q}}{(\cos\boldsymbol{q})^{3/2}}\right\}$$

$$\boldsymbol{L}_n = \boldsymbol{L}_n(\boldsymbol{q}_L) \cong \frac{n\boldsymbol{p}}{\int_0^{q_L}\frac{d\boldsymbol{q}}{(\cos\boldsymbol{q})^{3/2}}} \qquad (6).$$

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[2] The tangential displacement $y(x)$ is given as a direct linear operation on $f(x)$;[10] .

For this case (consider a circular section) it can then be shown that

$$\Omega_n \cong \Lambda_n \sqrt{\frac{(1-\cos\theta_L)}{\cos\theta_L}} \sqrt{\frac{(1-a)}{(1+a)}} \sqrt{\frac{g}{H}} \qquad (7)$$

which gives a formula for the analytical evaluation of the natural frequencies of a catenary line, written solely in terms of water depth H and of the upper end angle with respect to horizontal, θ_L. In (7) $a = m_a/m$, with $m_a \cong \rho\pi D^2/4$, ρ is the water mass density and D the external diameter. For practical and immediate usage of formula (7), Fig. 8 gives $(\Lambda_n/n)\sqrt{(1-\cos\theta_L)/\cos\theta_L}$ as a function of θ_L.

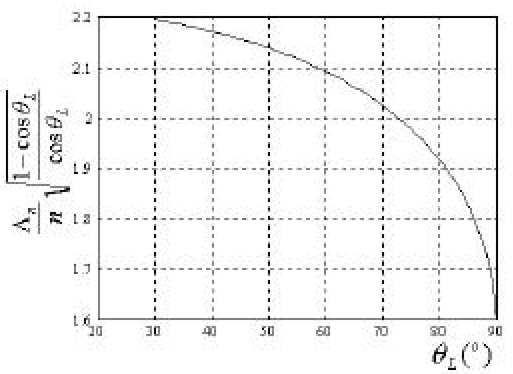

Fig. 8 Nondimensional eigenvalues for a catenary-riser, under no current, as a function of θ_L, the angle at the upper end, with respect to the horizontal; Pesce et al. (1999) [10] .

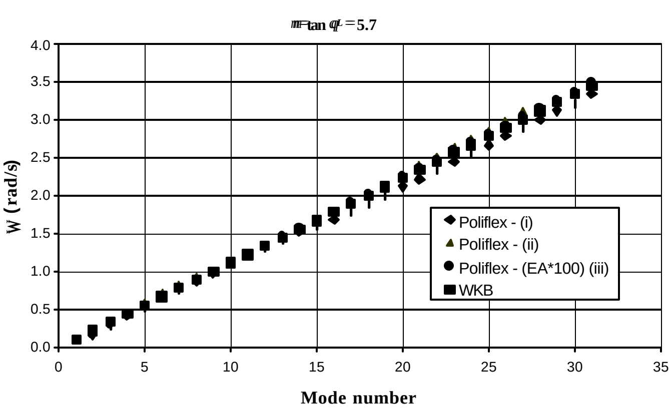

Fig. 9 Eigenvalues for a free-hanging flexible pipe catenary-riser. $\mu = \tan\theta_L = 5.7$; Pesce et al. (1999) [10] .

Figure 9 shows natural frequencies, comparing *PoliFlex* [17] numerical results and JWKB approximations. The third *PoliFlex* condition[3] corresponds to an almost inextensible pipe, comparing quite well with JWKB approximation. In this sense JWKB solution might be viewed as a benchmark for numerical codes, as *PoliFlex*, and a way to assess the effect of axial rigidity.

DYNAMIC COMPRESSION UNDER CURVATURE AND TWIST

Since the dynamic problem turned up and catenary riser configuration started to be used throughout the offshore oil industry, one of the major concerns was to avoid dynamic compression along the catenary. Typical flexible pipe structures used to have their engineering properties related to tension and bending loads rather than to compression. In addition, their response to pure compression could trigger the most catastrophic flexible pipe failure mode: *bird-caging*. Therefore, most of the riser design premises included a new constraint: to establish a catenary geometry avoiding compression. This practice pushed the riser design to increase the TDP tension to accommodate the compression wave induced by the motion imposed at the top end. Nevertheless, with the introduction of the FPSO concept and, consequently, more severe motions, the dynamic compression become inevitable to a certain amount. From the designer point of view, an analytical approach is worthwhile, as a critical value for compression is needed, at the very beginning, as a design criterion.

A research program was then started to investigate the stability of risers. Aranha and Pinto (2000) [8] proposed a stability model, deriving an analytic expression to determine the critical load in risers, subjected to dynamic compression and including the effect of the initial (static) curvature. Their treatment is restricted to the planar problem though. The authors point out the importance of an analytic estimate for the critical load based on the difficulty to interpret numerical results around the "saturation region" if a reference value in not known. Regarding the effect of the initial curvature on the critical load, they showed that the derived analytic expression recovers Euler's critical load in the limit when $\chi \to 0$ (straight beam). On the other hand, for moderately large curvatures, the expression gives a critical load about 9 times higher than Euler's critical load. Several comparisons with numerically obtained results (from full nonlinear simulation, without twisting) were made, indicating a fairly good agreement, in the sense that the numerically determined tensions tend, in fact, to "saturate compression" around the estimated critical load.

[3] Conditions in Fig. 11 are: (i) Riser hinged at TDP; hinge mounted on a linearly elastic horizontal spring, whose rigidity is the same as for the effective length of cable laid on the sea floor, assuming a Coulomb friction law with coefficient 0.4. Actual value for the axial rigidity is taken. (ii) Riser hinged at TDP, but with the actual value for the axial rigidity. (iii) Riser hinged at TDP, taking the axial rigidity 100 times larger than the actual value

By introducing a physical argument, according to which, a 'local buckling length-scale' can be estimated from the classical beam dispersion relation (Whitham, 1974) [18] as,

$$l = \frac{p}{k} = p.\sqrt[4]{\frac{EI}{(m+m_a).w^2}} \tag{8}$$

where k is the flexural wave number, the critical load is shown to satisfy an extended Euler's relation

$$P = EJ\left(\frac{pp}{l}\right)^2 \tag{9}$$

where p is the root of a simple characteristic equation

$$\tan p = p + \frac{p^3}{3} - \frac{p^5}{h^2} \tag{10}$$

The critical load is dependent upon the frequency w of the cyclically imposed tension. It turns out that for small curvature, such that $cl \le 10^{-2}$, $p \cong 1$, what recovers Euler's critical value. For moderately large curvatures or upper, such that $cl \le 3\times10^{-2}$, $p \cong 3$, thus increasing Euler's value by a factor of 9. Note that such a result was restricted to a planar analysis.

Extending Aranha & Pinto's reasoning a little further, Ramos (2001) [12] , Ramos & Pesce (2001) [13] , considered the complete three-dimensional problem, in the Kirchoff sense; see Antman, 1974 [19] . It has been shown that the well-known Greenhill formula can be used to predict stability conditions in cables/risers having an initial curvature and subject to dynamic compression under twisting. The point: 'how to interpret the value of n that appears in Greenhill's expression' has been answered. A *general* procedure was developed, and an analytical tool oriented to the design of risers, specific to stability analysis, was presented.

There are two nondimensional numbers that regulate the instability phenomenon. The first one is the nondimensional curvature $c.l$, being l the 'local buckling length scale', determined from the classical beam dispersion relation. The second one is the nondimensional twist, $\tilde{k}_t l$, that gauges the twist under the same length scale. In fact, the nondimensional version of Greenhill relation, can be written as

$$Pl^2/EI + ((GJ/2EI)\,\tilde{k}_t l)^2 = (pp)^2 \tag{11}$$

where p, being a function of $c.l$ and $\tilde{k}_t l$, is the solution of a three-dimensional linear characteristic equation (Ramos and Pesce, 2001) [13] asymptotic to (10) when twisting is not considered.

From this generally valid theoretical analysis it was inferred that: a) when no twisting is applied to the riser, irrespective the frequency of oscillation on tension, we have $p @ n = 1$ for small curvatures ($c.l$~0.01), whereas $p @ n = 3$ for large curvatures ($c.l$>0.1); b) however, if curvature is large enough ($c.l$>0.1) and a very small twist is applied, the value of n jumps from 3 to 2, and buckling occurs three-dimensionally, as could be expected. In regions of moderate curvature, critical load can therefore be overestimated by a factor of 9/4, if twisting is not considered.

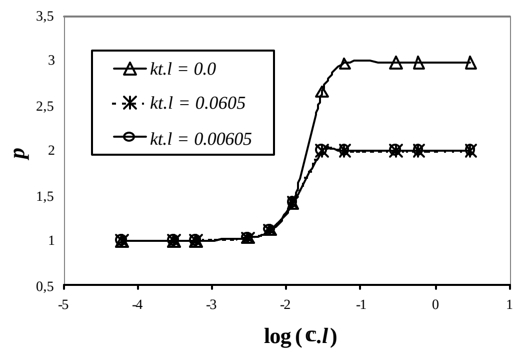

Figure 10 Curves p x log(χ.l) for different values of nondimensional twist $\tilde{k}_t$ l. (Ramos and Pesce, 2001 [13])

DYNAMIC TENSION AND RANDOM ENVIROMENT

The non-linear dynamic tension problem under random environmental conditions has also been addressed, (Aranha and Pinto, 2000 [8] ; Aranha, Pinto and Silva, 2000) [20] , and an asymptotic solution has been obtained. The analytical approach also gives the probability density function (pdf) of dynamic tension directly, as a nonlinear transformation on the surface waves pdf.

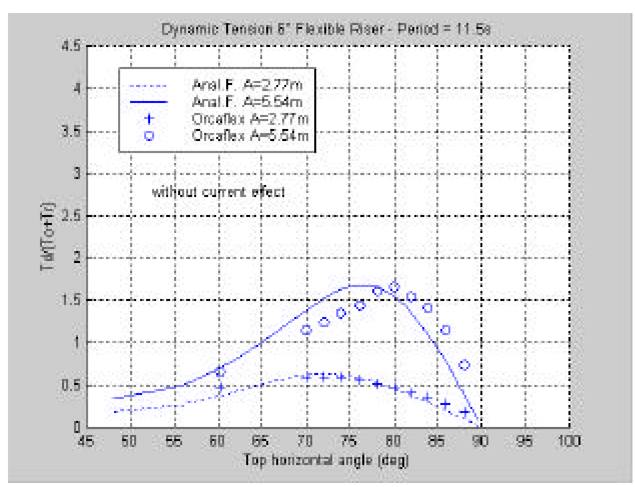

Figure 11 Dynamic tension. Analytical Solution compared to fully non-linear model results, for a 6 inches internal diameter flexible riser

(Pinto et al., 1999 [21]). T_0 is the tension at TDP. $T_r = 98\text{kN}$ is an experimentally determined compression limit value.

STRUCTURAL MECHANICS OF FLEXIBLE PIPES AND UMBILICAL CABLES

Another research branch re-addressed the already classical structural mechanics problem of flexible pipes and umbilical cables. A consistent and comprehensive formulation was presented, (Ramos 2001 [12] ; Ramos et al, 2000 [22] ; Ramos and Pesce (2001) [23] ; Ramos and Pesce, 2002 [24]), to estimate the structural behaviour of flexible risers subjected to any combination of tension, torsion and bending, as well as internal and external pressures applied to it. All the assumed hypotheses were highlighted, giving the readers more confidence on applying the proposed equations in specific cases. Comparisons between analytical results using the proposed model and experimental results obtained in literature showed a reasonable agreement. An alternative equation for bending stiffness calculation for flexible pipes was presented as

$$(EI)_{eq} = \sum_{i=1}^{m} E_i I_i + \sum_{i=1}^{n} n_i \cos\alpha_i \left[G_i I_{t,i} + \frac{3}{2}\left(E_i I_{y,i} - G_i I_{t,i}\right)\cos^2\alpha_i \right] \quad (12)$$

In Ramos and Pesce (2002) [25] an overview of some analytical equations proposed in the technical literature to estimate the flexural structural behaviour of flexible risers was presented, comparing some of these equations and showing the similarities and differences between them.

Further improvements should consider: modeling of friction interaction between layers, enabling to take into account coupling between bending and axi-symmetric loads, (consequently the effect of internal pressure on bending stiffness); pipe ovalization due to bending; a more detailed study on the contribution of carcass and zeta layers to bending strength.

SCR DESIGN ISSUES

The analytical approach, being simple in form, enables one to treat catenary riser problems not only from an analysis point of view, but also from a design oriented one. Note that design oriented approaches, based on classical methods of applied mathematics, have been rare in the riser subject (see, e.g., Bernitsas et al., 1985 [26]). Under the analysis approach, the code *PoliFlex*, Martins (1998) [17] , had been developed. Such a code solves the global dynamic problem in frequency-domain, correcting local flexural effects by means of the analytical solutions, summarized above. This code had been included earlier in a design methodology; Pesce et al. (1995) [27] .

Nevertheless, such a code must be run, parametrically, many times, during an optimization design procedure. Therefore, in order to improve the efficiency of the design methodology for the SCR concept, a specific computer code, *SeelCat*, has been developed for PETROBRAS, Martins (1998) [28] ; Martins (2000) [15] .

This code handles thousands of cases, having the static tension at top, in *'neutral condition'*, as a control parameter. Such cases correspond to several combinations of current and waves, concerning design environmental conditions. This procedure allows feasibility analysis to be carried out in a very efficient way. Not only extreme conditions can be accounted for but also fatigue analysis can be incorporated in the earlier design phases. First-order wave motions, as well as slow-drift motions of the vessel, are considered in a random environment. Vortex-induced vibrations effects are also taken into account, using a standard multiple-scale, Van der Pol like model approach.

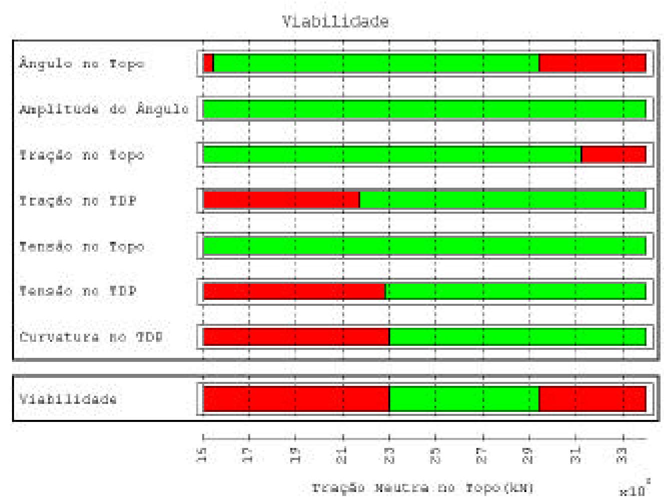

Fig. 12. Example of feasibility analysis for extreme (centenary) conditions, for a 12¾inch external diameter SCR in 855 meters deep water in Campos Basin. Current South; Wave SE - FPSO 100% loaded. Martins (2000) [15] .

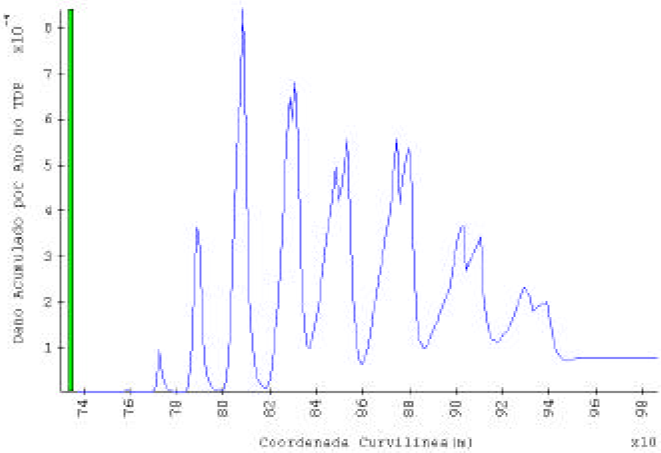

Fig. 13 Yearly cumulative damage along the arch-length of a SCR in the TDP region, for a ('neutral') static tension imposed on top of 3000 KN. Damage dominated by "first-order" motions imposed by the floating unit; Martins (2000) [15] .

An interesting result from *SeelCat* is given as a chart, where, according to the various design limits, e.g., stresses at TDP and top end, tension at top, angle at top, curvature, the structural feasibility is addressed. Figure 12 and 13 show results of an example for a SCR at 855m water depth. The FPSO is the converted Petrobras' tanker Henrique Dias; see [15] for general data. Turret positioning and FPSO heading are determined according to Pinto et al, 1999 [21] and Leite et al, 1999 [29] . The red color indicates unfeasibility. As can be seen, a static top tension of about 2700kN should be adopted in this case.

It should be noted that, although initially addressed under the SCR concept, this methodology is rather general and can be applied to umbilicals and flexible risers, as well. In this case, the internal distributions of stresses on the structural components, as well as wear in armours, must be properly considered; see, e.g., Ramos Jr. et al (2000) [12] .

ASSESSING THE FATIGUE-LIFE OF FLEXIBLE RISERS

Evaluating the fatigue life of a flexible riser requires several dynamic problems to be solved, considering a consistent representation of all environmental loads imposed to the structure. For each load case the stress field over the distinct layers has to be determined. Accordingly, slip between adjacent layers must be considered as the resulting wear may reduce the section of the armour wires.

A design-oriented simplified methodology to assess the fatigue life of flexible risers, directly applicable in preliminary and early design stages has then been proposed (Martins and Pesce, 2002 [30]).

The procedure incorporates analytical models for the local stress-strain-slippage fields, to a standard numerical code that deal with the riser's global structural dynamics. Many of such analytical models can be found in the literature; e.g., Lanteigne (1985) [31] , Féret and Bournazel (1987) [32] , Féret et al. (1995) [33] , Witz and Tan (1992a,b) [34] [35]). Most of those have been recently improved by many authors. A recent work, following a deep consistency analysis, including Antman's analysis (1974) [19] , can be found in Ramos (2001) [12] . The detailed fatigue assesment procedure can be found in Pesce et al (1997) [36] .

Figure 14 shows a final result, presenting a relative safety factor, along the riser length, referenced to the TDP region, considering a fatigue limite criterion.

It worths mentioning that theoretical service-life prediction of flexible pipes has been an issue of many investigations as, e.g., Féret et al (1986) [38] ,Claydon et al. (1992) [39] , Fuku et al (1992) [40] . Nevertheless service-life assesment continues to be a rather difficult research subject, involving a vast source of combined modelling aspects, not yet comprehensively resolved as, for example, the simultaneous occurrence of wear, mechanical fatigue and material aging, what has serious implications on the definition of proper limit criteria.

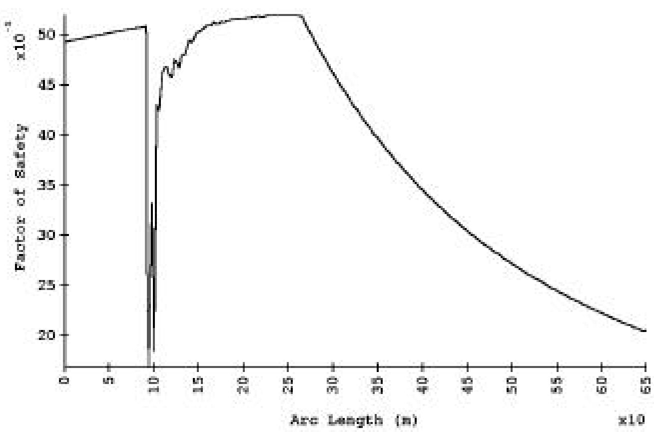

Figure 14 – Relative Safety Factor along the riser, with respect to the TDP region, considering a fatigue limit stress criterion; (Martins and Pesce, 2002 [30])

CONCLUSIONS

Industry-University cooperation is a key to consistent developments in any area and a must in offshore engineering. Particularly such an effort in riser developments has enabled design offshore engineers to face new challenges in deep-water frontier.

This first of two companion papers summarized some results concerning the "catenary" riser problem in a free-hanging configuration, obtained during last years, through a cooperative research program. The analytical approach, being simple in form, enables one to treat catenary riser problems not only from an analysis point of view, but also from a design oriented one. Merging frequency-domain methods with asymptotic techniques, the following points were consistently and successfully addressed:

- Dynamic curvature variation in the *touch down point (TDP)* and in the *upper endfit* regions;
- Catenary risers eingen-value problems;
- Dynamic compression under bending and twist;
- Soil elasticity effect at TDP;
- Riser design issues, by developing specific analysis and design oriented computer codes to deal with extreme loads and fatigue.

Numerical comparisons with fully non-linear time domain simulation codes showed outstanding agreement. Experimental results gave support for analytical solutions within a very good agreement as well. Such analytical - frequency domain and asymptotic - approaches turn easy to treat catenary riser problems from a design-oriented point of view. Not only extreme conditions can be accounted for but also, particularly, fatigue analysis can be incorporated in the early stages of design.

Much more research on risers[4] remains to be addressed , for example:

- Further development of CFD-VIV design oriented codes;
- Extensive experimental program on dynamic compression;
- Adequacy of fatigue models;
- Analysis of field data.

It should pointed out that there are many opportunities and several technological gaps that could bring down the barrel lift cost, offshore Brazil. It means that academy and industry can and should keep interacting, supporting the continuous growth of engineering capability and helping to consolidate a solid critical mass among professors, lecturers and research students involved in such agreements.

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