Analytical assessment of the drilling risers stability in the Newfoundland deep offshore

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ABSTRACT
Marine drilling risers are amongst the key structural elements in deepwater oil and gas exploration projects. These risers are continuously subjected to dynamic environmental and operational loads. This has caused the drilling risers to be vulnerable against the fatigue loads. In this study, an analytical model was adopted by solving the governing differential equations to analyze the effect of Vortex-induced vibration (VIV) on the strength and stability of an arbitrary drilling riser operating in the Offshore Newfoundland. The case study showed that the amplitude of first-order dynamic oscillations is larger than that of higher-order responses. However, the higher-order responses can cause noticeable dynamic moment and shear force. The natural frequency of risers was found to decrease with increasing the water depth and increase for higher magnitudes of top tension. The analytical approach was found to be an appropriate solution for early assessment of the drilling riser fatigue life due to vortex induced vibration.

RÉSUMÉ
Les colonnes montantes de forage font partie des éléments structurels essentiels des projets d'exploration pétrolière et gazière en eau profonde. Ces colonnes montantes sont continuellement soumises à des charges environnementales et opérationnelles dynamiques. Cela a rendu les colonnes de forage vulnérables aux charges de fatigue. Dans cette étude, un modèle analytique a été adopté en résolvant les équations différentielles régissant l'analyse des effets des vibrations induites par vortex (VIV) sur la résistance et la stabilité d'une colonne montante de forage arbitraire en exploitation au large de Newfoundland. L'étude de cas a montré que l'amplitude d'oscillation dynamique du premier ordre est supérieure à celle des réponses en mode d'ordre supérieur. Il a également été observé que l'effet du moment dynamique et de la force de cisaillement des modes d'ordre supérieur pourrait être important. Il a été constaté que la fréquence naturelle des colonnes montantes diminuait avec l'augmentation de la profondeur de l'eau et augmentait lorsque les tensions maximales étaient plus élevées. L'approche analytique s'est avérée être une solution appropriée pour l'évaluation précoce de la durée de vie en fatigue du riser de forage due au vortex induit des vibrations.

1 INTRODUCTION

Marine drilling risers are the best solutions for oil and gas developments off the coast of Newfoundland and Labrador. For several decades and to the present day, new technologies have been adopted to reduce the fatigue of the drilling riser by carefully analyzing its strength and stability in deepwater drilling operations. The desire to drill in harsh, deepwater environments necessitates the need for riser strength and stability analysis. However, there is a requirement for more refined methods such as the use of finite element models to verify the strength and stability of the drilling riser system in extreme conditions.

The challenges of deepwater oilfields and inter-continental gas transportation present the biggest opportunities for advancements in pipeline technology. At some fields, strong waves, currents, high pressures and high temperatures (HP/HT), sour reservoirs and deepwater conditions are pushing the limits of the marine riser. The hydrodynamic forces have a strong influence on the riser stability vis-a-vis its strength. Vortex induced vibration (VIV) is a major consideration in determining the stability of riser and should not be allowed to occur at any time during the design life of the riser system. As the water depth increases, the influence of VIV becomes crucial and poses more challenges to the overall system response. Therefore it is of great importance to investigate the dynamic behavior of risers under combined wave-current interaction in deepwater environments.

In this study the analytical method proposed by Jin et al. (2007) was adopted to assess the stability of a drilling riser in the Newfoundland offshore. Additional details on the derivation of the equations presented herein can be found in the work of Jin et al. (2007).

2 NOMENCULTURE

V_c	current velocity
L_w	wavelength
ω_w	wave circular frequency
T_w	wave period
H	wave height
k	wave number
L	length of the riser

D	diameter of the riser
t_r	wall thickness
$\bar{m}$	mass per unit length
EI	flexural stiffness of the riser
T_0	top tension
ζ_s	damping ratio
ρ	density of the seawater
ω_s	vortex shedding frequency
C_L	lift coefficient
C_d	fluid damping coefficient
C_a	coefficient of additional mass
K_d	coefficient
m'	mass of adhered water per unit length

3 VIBRATION EQUATION OF MARINE RISERS

A simplified analytical model has been established by Jin et al. (2007) to analyze the influence of VIV on the strength and stability of marine risers. A method for estimating vortex-induced dynamic response was developed. The riser is considered as a vertical beam with current and wave acting perpendicular to its undeflected longitudinal axis, as shown in Figure 1.

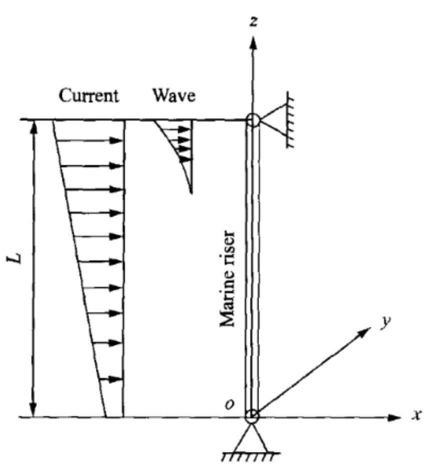

Figure 1. Marine riser model (Jin et al, 2007)

Dong (1994) proposed an equation for the transverse motion of a marine riser under small-amplitude linear wave with the assumption of uniform cross section and mass distribution for the riser along z-direction. This equation is presented below:

$$EI.\frac{\partial^4 y}{\partial z^4} - T_0.\frac{\partial^2 y}{\partial z^2} + c.\frac{\partial y}{dt} + m.\frac{\partial^2 y}{\partial t^2} = F_y(z,t) \quad [1]$$

where, EI is the flexural stiffness of the riser [N.m^2];
T_o is the riser top tension [N];
c is the coefficient of viscous damping;
m is the mass per unit length of the beam [kg/m]; and,
F_y (z, t) is the total external fluid force per unit length in the y-direction [N/m]:

$$F_y(z,t) = F_L(z,t) - F_r(z,t) \quad [2]$$

where, F_L *(z, t)* is the vortex lift force per unit length [N/m]; and,
F_r *(z, t)* is the nonlinear fluid damping force.
Both F_L *(z, t)* and F_r *(z, t)* are caused by the motions of riser in y-direction.
F_L *(z, t)* can be expressed as follows:

$$F_L(z,t) = \frac{1}{2}\rho D(V_c+u)^2 C_L \cos(w_s t) = K_L(z).C_L\cos(w_s t) \quad [3]$$

where, ρ is the density of seawater [kg/m^3]; D is the external diameter of the riser [m]; C_L is the lift coefficient; ω_s is the frequency of vortex shedding [rad/s]; and, K_L is the lift force distribution coefficient [kg/s^2] which can be obtained as:

$$K_L(z) = \frac{1}{2}\rho D(V_c(z)+u)^2 \quad [4]$$

where, V_c is the current flow velocity and can be expressed as a linear function of water depth:

$$V_c(z) = a + bz \quad [5]$$

The horizontal propagation velocity of the linear wave, u, can be expressed as:

$$u = u(z,t) = \frac{\pi.H}{T_w} e^{k(z-l)}\cos(\omega_w t) \quad [6]$$

where, H is the wave height [m]; T_w is the wave period [sec]; ω_w is the wave circular frequency [rad/sec]; l is the length of the riser [m]; k is the wave number, $k = 2\pi/L_w$; and, L_w is the wavelength [m].

Substituting Eq. [6] into Eq. [4], the lift force distribution coefficient is obtained as:

$$K_L(z) = \frac{1}{2}\rho D\left(V_c(z) + \frac{\pi.H}{T_w}e^{k(z-l)}\cos(\omega_w t)\right)^2 \quad [7]$$

The nonlinear fluid damping force, F_r (*z*, *t*), can be defined using Morison Equation as follows:

$$F_r(z,t) = \frac{1}{2}\rho D C_d y'|y'| + C_a\rho\frac{\pi D^2}{4} = K_d C_d y'^2 sgn(y') + m'y''$$

$$K_d = \frac{\rho D}{2} \qquad [8]$$

$$m' = \frac{\rho C_a \pi D^2}{4}$$

where, C_d is the damping coefficient of water; m' is the added mass per unit length [kg/m]; and, C_a is the added mass coefficient.

4 VIV ASSESSEMENT APPROACHES OF MARINE RISERS

VIV has been a major concern for marine risers due to the potential to cause severe fatigue damage. In order to design marine risers, it is necessary to have good theoretical and analytical models for the prediction of VIV at the initial stage of design and prior to the development of a comprehensive numerical model, which is required in order to evaluate the problem and envisage solutions.

A simplified assessment of VIV is proposed in DNV-OS-F201. This simplified estimate of the induced fatigue damage is computed by conservatively assuming that 2D sheared current profiles are applied on the riser (e.g., unidirectional with a magnitude that varies with distance below the sea level). A lot of developments have been identified in this particular area. Software, such as SHEAR7 and VIVA by MIT and OrcaVIV by Orcina are now potentially suitable for handling VIV issues.

Jin et al. (2007), carried out some studies on the strength and stability analysis of deepwater marine drilling risers by analyzing the influence of VIV on the strength and stability of the risers. They established a simplified analytical model and developed a method for calculating vortex induced dynamic response.

The VIV response of the riser is obtained by transforming the partial differential equation (PDE) of the transverse motion, Eq. [1], into a set of nonlinear ordinary differential equations (ODEs) using Galerkin method (Jin et al. 2007). The riser is assumed as a simply supported beam at both ends with the following boundary conditions:

$$y(0,t) = 0 \quad \frac{\partial^2 y(0,t)}{\partial z^2} = 0 \qquad [~~8~~9]$$

$$y(l,t) = 0 \quad \frac{\partial^2 y(l,t)}{\partial z^2} = 0$$

The lateral displacement $y(z, t)$ is given as a series of vibration mode shapes as follows:

$$y(z,t) = \sum_{n=1}^{\infty} y_n(t)\sin(\lambda_n z) \qquad [10]$$

$$\lambda_n = \frac{n\pi}{l}$$

The total external fluid force per unit length in the y-direction is then obtained by substituting Eq. [3], Eq. [8], and Eq. [10] into Eq. [2]. Also, by rearranging Eq. [1] via substituting Eq. [10] and [2] and applying the Galerkin method, we obtain:

$$y'' + \left[\lambda_{B_n}^2 + \lambda_{C_n}^2\right] + \frac{C_n}{\bar{m}} y_n' + \frac{2D_n}{l\bar{m}}$$

$$= \frac{2C_L}{\bar{m}l}\, cos(\omega_s t) \int_0^l K_L(z)\sin(\lambda_n z) dz$$

$$= \frac{2C_L}{\bar{m}l}\, cos(\omega_s t) \int_0^l \left[\frac{1}{2}\rho D \left(V_c(z) + \frac{\pi.H}{T_w} e^{k(z-l)} \,cos\,(\omega_w t)\right)^2\right] . sin(\lambda_n z)\, dz$$

$$= A_c \,.cos(\omega_s t) + \left[A_w \,.cos(2\omega_w t) + 2\,.\; A_{cw}\,.cos(\omega_w t)\right].cos(\omega_s t)$$

$$\bar{m} = m + m' \qquad [11]$$

$$\lambda_{B_n}^2 = \lambda_n^4 (EI/\bar{m})$$

$$\lambda_{C_n}^2 = \lambda_n (T_0/\bar{m})$$

$$C_n = 2\bar{m}(\lambda_{B_n}^2 + \lambda_{C_n}^2)^{1/2}\zeta_s$$

$$A_c = \begin{cases} \frac{\rho D C_L}{\bar{m}l} \cdot \left\{a^2 \cdot \left(\frac{2l}{\pi}\right) + 2ab\left(\frac{l^2}{\pi}\right) + b^2\left[\frac{l^2}{\pi} - 4\left(\frac{l}{\pi}\right)^8\right]\right\}, & n = 1{,}3 \\ -\frac{\rho D C_L}{\bar{m}l}\left[ab\left(\frac{l^2}{\pi}\right) + b^2\left(\frac{l^3}{2\pi}\right)\right] & , n = 2{,}4 \end{cases}$$

$$A_w = \begin{cases} \frac{\rho D C_L}{\bar{m}l} \cdot \left(\frac{\pi H}{T_w}\right)^2 \cdot \frac{n\pi/l}{4k^2 + \left(n\pi/l\right)^2} \cdot (e^{-2kl} + 1), & n = 1{,}3 \\ 0 & , n = 2{,}4 \end{cases}$$

$$A_{cw} = \begin{cases} \frac{\rho D C_L}{\bar{m}l} \cdot \left(\frac{\pi H}{T_w}\right) . \left[aB_n \,.\,(e^{-kl} + 1) + b.\left(lB_n - 2k\bar{B}_n \,.\,(e^{-kl}+1)\right)\right] & , n = 1{,}3 \\ -\frac{\rho D C_L}{\bar{m}l} \,. b . \left(\frac{\pi H}{T_w}\right) . \,l\,\bar{B}_n & , n = 2{,}4 \end{cases}$$

$$B_n = \frac{n\pi/l}{k^2 + \left(n\pi/l\right)^2}$$

$$\bar{B}_n = \frac{n\pi/l}{\left[k^2 + \left(n\pi/l\right)^2\right]^2}$$

where, $\bar{m}$ is the virtual mass of the riser per unit length, [kg/m]; $\lambda_{B_n}^2$ is the riser natural frequency for bending vibration; $\lambda_{C_n}^2$ is the riser natural frequency for for axial vibration; C_n is the viscous damping coefficient; ζ_s is the dimensionless damping ratio of the riser; A_c, A_w, A_{cw}, B_n, and $\bar{B}_n$ are the coefficients; and D_n is obtained through a numerical algorithm (Jin et al. 2007):

$$D_j = K_d C_d \int_0^l sgn(y')y'^2 \sin(\lambda_j z)dz, \qquad j = 1,2,3, \dots n \qquad [12]$$

$$y' = y'(z,t) = \sum_{i=1}^{n} y_i'(t)\sin(\lambda_n z) \qquad [13]$$

Solving this equation, the dynamic displacement of the riser, *y (z, t)*, can be obtained using Eq. [10].

According to Ma et al. (2000), the dynamic moment is then given as:

$$M(z,t) = EI\frac{\partial^2 y}{\partial z^2} = EI\sum_{n=1}^{\infty} y_n^2 y_n(t)\sin(\lambda_n z) \qquad [14]$$

The dynamic shearing force acting on the riser can be obtained from the following equation:

$$Q(z,t) = EI\frac{\partial^3 y}{\partial z^3} = EI\sum_{n=1}^{\infty} y_n^3 y_n(t)\cos(\lambda_n z) \qquad [15]$$

5 FATIGUE ANALYSIS

Fatigue life assessments of drilling risers is a challenging aspect of the design due to contribution of several complex and interactive loading mechanisms such as operational vibrations, wave-induced oscillations, and vortex-induced vibrations (VIV).

Analytical solutions are adopted for the fast assessment of drilling riser performance at the early stages of design, prior to comprehensive numerical simulations of the complex loading conditions.

The total fatigue damage is assumed to be generated by the combined action of the following contributions:

- Mean motions of the vessel caused by the sequence of storms foreseen in the long-term environmental conditions.
- Slow-drift motions of the vessel inside each storm event.
- Wave-frequency motions of the vessel and hydrodynamic loads applied directly to the riser for each of the above events.
- Vortex induced vibration (VIV) effects in the length of riser exposed to high current profiles (e.g., first few 100 meters below the sea surface).
- Installation operations.

The admissible fatigue life is assumed to be equal to 10 times the design life for the entire pipe length. Suitable criteria will be defined to couple a particular environmental condition (wave and current) with the corresponding vessel offset. A time-domain approach will be followed to describe the dynamic response of the riser generated by the representative sea states of the long-term distribution. The total fatigue damage is then evaluated by means of a suitable procedure that shall be aligned with the solution approach, considering a reference S-N (stress range – number of cycles to failure) curve and the Palmgren-Miner law for summing the partial contributions.

The fatigue life of the marine riser can be estimated using Palmgren-Miner theory. Hence, the damage criterion is given
by:

$$D_i = \sum_i \frac{n(\Delta\varepsilon_i)}{N(\Delta\varepsilon_i)} \qquad [16]$$

where, $n(\Delta\varepsilon_i)$ is the number of cycles of alternate strain that occurred in the range of $\Delta\varepsilon_i$ and can as well be expressed as:

$$n(\Delta\varepsilon_i) = f_i t_i \qquad [17]$$

f_i is the frequency corresponding to the *i*th amplitude [rad/ses]; and t_i is the vibration time [sec].

Similarly, the denominator in Eq. [16] can be obtained from the relevant S-N curve and it is related to the equation below:

$$N(\Delta\varepsilon_i) = c.(\Delta\varepsilon)^{-b} \qquad [18]$$

where *c* and *b* are constants.
The symbol $\Delta\varepsilon$ is the maximum difference of strain in one cycle and is taken from the middle point of the riser having two joint and is expressed as follows:

$$\Delta\varepsilon = \pi^2 A_o(\frac{D}{L})^2 \qquad [19]$$

where A_o is the amplitude of the middle point of the riser [m]. By substituting Eqs. [17], [18] and [19] into Eq. [16] we can obtain the value of D_i for *t* = 1 year as follows:

$$D_i = \sum_i \frac{f_i t_i A_{0,i}^4 (\frac{D}{L})^8}{6.745 \times 10^{-12}} \qquad [20]$$

Assuming, $T_i = \frac{t_i}{3600 \times 365}$ and substituting it into Eq. [20], we can obtain the fatigue lifetime of the riser in years as follows:

$$D_i = \frac{5.133 \times 10^{-18} (\frac{L}{D})^8}{f_n \sum_i (\frac{f_i}{f_n}) T_i A_{o,i}^4} \quad [21]$$

6 CASE STUDY

Table 1. Basic parameters

Parameter	Value	Unit
V_c	0.78	m/s
L_w	224	m/s
ω_w	0.523	Rad/s
T_w	12.014	S
H	10	m
k	0.0280	m^{-1}
c	1000	m
D	0.6049	m
t_r	0.0254	m
$\bar{m}$	974.2	Kg/m
EI	418	$MN.m^2$
T_0	150000	N
ζ_s	0.0018	
ρ	1025	Kg/m^3
ω_s	0.2917	rad/s
C_L	2.4	
C_d	0.6	
C_a	1	
K_d	310.0113	Kg/m^2
m'	294.5648	Kg/m

The calculated coefficients considering top tension are shown in Table 2.

Table 2. Natural frequencies of riser

Mode n	Natural frequencies of riser (λ_n)	Natural frequencies of bending vibration of the riser ($\lambda_{B_n}^2$)	Natural frequencies of axial vibration of the riser ($\lambda_{C_n}^2$)
1	0.003142	0.00004180	0.4837
2	0.006283	0.0006687	0.9674
3	0.009424	0.003385	1.4512
4	0.01257	0.01070	1.9349

6.1 Dynamic response analysis of marine risers

As the length of the marine risers increases, their natural frequencies tend to decrease. However, resonance is likely to occur when the natural frequency of the riser is close to the vortex shedding frequency. Applying the Runge-Kutta Method, the equations can be reduced as follows:

$$Y_1 = y_1 \quad [22]$$

$$Y_2 = \frac{dY_1}{dt} = \frac{dy_1}{dt} = y_1' \quad [23]$$

$$Y_3 = y_2 \quad [24]$$

$$Y_4 = \frac{dY_3}{dt} = \frac{dy_2}{dt} = y_2' \quad [25]$$

$$Y_5 = y_3 \quad [26]$$

$$Y_6 = \frac{dY_3}{dt} = \frac{dy_3}{dt} = y_3' \quad [27]$$

$$Y_7 = y_4 \quad [28]$$

$$Y_8 = \frac{dY_7}{dt} = \frac{dy_4}{dt} = y_4' \quad [29]$$

where, Y_1, Y_2, Y_3, Y_4, Y_5, Y_6, Y_7 and Y_8 are values of iterations in meters.

Fig. (2) shows the first four modal responses of the riser under the combined interaction of wave and current. Based on this figure, the dynamic amplitude of the first-order mode is greater than the amplitude of higher-order responses.

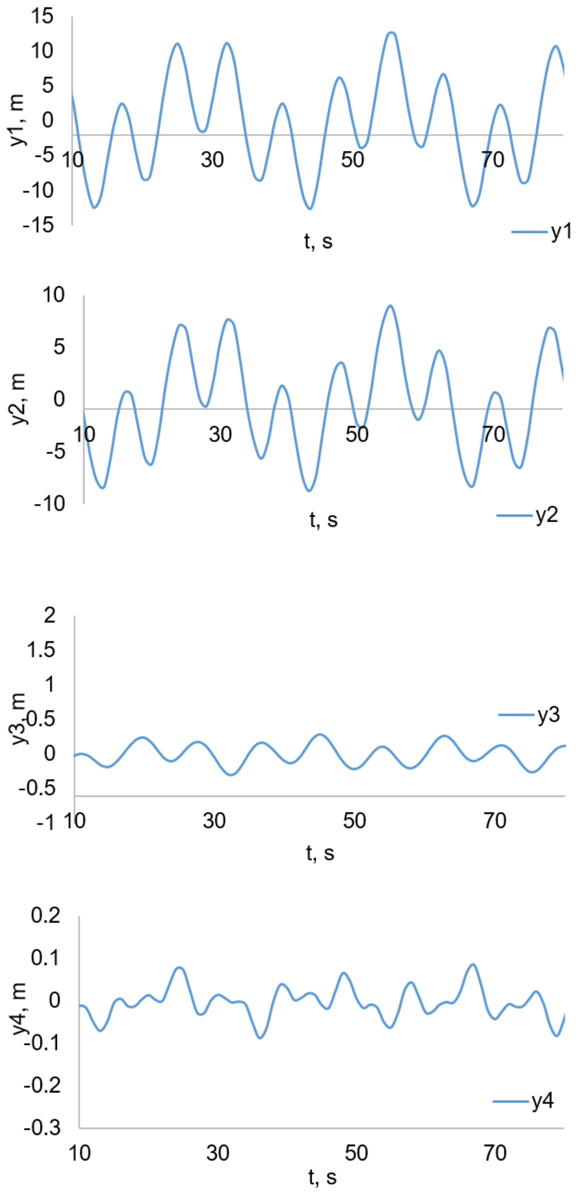

Figure 2. Modal response considering the combined wave-current interaction

Eq. [14] can be modified as follows:

$$M(z,t) = EI\frac{\partial^2 y}{\partial z^2} = EI\left\{y_1(t)\sin\left(\frac{\pi z}{l}\right) + 4y_2(t)\sin\left(\frac{2\pi}{l}\right) + 9y_3(t)\sin\left(\frac{3\pi z}{l}\right) + 16y_3(t)\sin\left(\frac{4\pi z}{l}\right)\right\} \quad [30]$$

Hence, $z = l/2$, for the middle point of the riser. Therefore, the dynamic displacement becomes:

$$M\left(\frac{l}{2},t\right) = EI\frac{\partial^2 y}{\partial z^2} = EI\left\{y_1(t)\sin\left(\frac{\pi z}{l}\right) + 9y_3(t)\sin\left(\frac{3\pi z}{l}\right)\right\} \quad [31]$$

Eq. [31] shows that $y_3(t)$ has a significant influence on the bending moment when, $z = l/2$.

Figures 3 and 4 show the responses of dynamic moment and dynamic shear force when the primary resonance is generated. Figure 5 shows the responses of dynamic moment at the middle point and the dynamic shear force at the bottom of the riser.

Since the natural frequency of marine risers increases with top tension, vortex-induced vibration can be avoided by increasing the top tension.

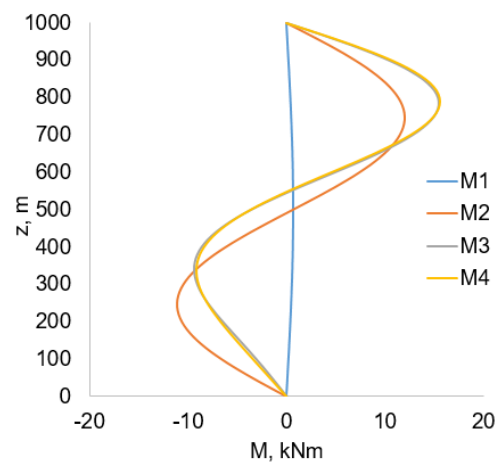

Figure 3. Dynamic moment responses considering combined wave-current loads

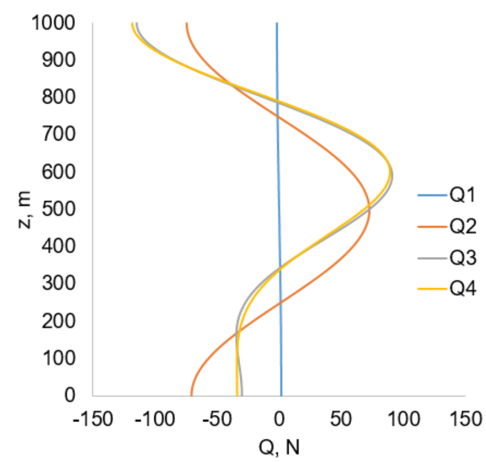

Figure 4. Shear force responses considering combined wave-current loads

6. CALCULATION OF FATIGUE LIFE

In Newfoundland and Labrador, the offshore fields are located on the Grand Banks in the Jeanne d'Arc Basin and the current is assumed to be constant all year round. In order to perform the fatigue analysis on a riser for a 20 year period Eq. [21] is used. One of the key technical challenges of deepwater drilling is riser fatigue due to VIV. Therefore, the effect of vortex induced vibration should be considered during the design of the marine riser. Figures 5a and 5b show the dynamic moment response and shear force at the middle point and bottom point of the riser respectively.

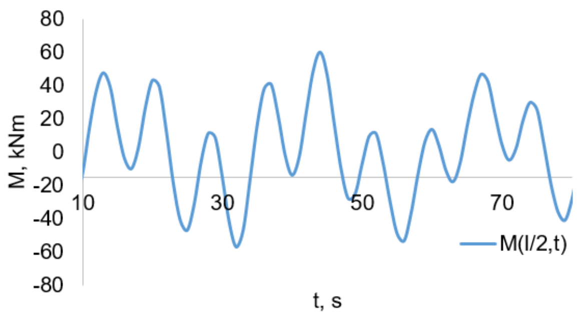

Figure 5a: Dynamic moment response at the middle point.

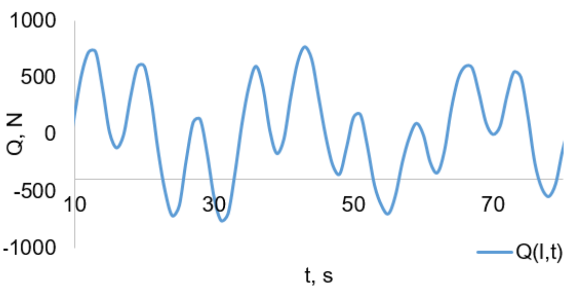

Figure 5b: Shear force at the bottom of the riser

7 CONCLUSIONS

Offshore drilling is very challenging and with the increase of operations in Newfoundland and Labrador's offshore oil and gas industry, the need for thorough assessment of drilling riser strength and stability is of paramount importance as it becomes more critical in harsh environments. From the analytical results, it can be concluded that the first-order mode dynamic response is greater than higher-order mode responses at primary resonance. In addition, the natural frequency of a riser decreases with increasing length but increases with increasing top tension. The need for VIV suppressors should be investigated with respect to VIV effects as part of a riser fatigue damage assessment. Should excessive motions be expected, effective suppressors can be selected and applied to the required riser length. Numerical modeling of the problem is required in order to evaluate the envisaged solutions.

8 AKNOWLEDGEMENTS

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