Journal of Petroleum Science and Engineering 163 (2018) 378–386

Contents lists available at ScienceDirect

Journal of Petroleum Science and Engineering

journal homepage: www.elsevier.com/locate/petrol

Analysis on multi-frequency vortex-induced vibration and mode competition of flexible deep-ocean riser in sheared fluid fields

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ARTICLE INFO

Keywords:
Vortex-induced vibration
Multi-frequency
Mode competition
Fluid-solid coupling
Long flexible riser

ABSTRACT

Multi-frequency vortex-induced vibration of flexible riser in lineally sheared fluid fields with different shearing parameters is explored by using the numerical approach. By combining the finite element method with a hydrodynamic model, the approach can consider mode competition based on modal energy and can carry out nonlinearly simultaneously dynamic response in time domain. Our analysis shows that multi-frequency VIV may occur both in non-uniform and uniform fluid fields. And, the behaviors of multi-frequency VIV are different from single-frequency VIV. Because several modes are involved and compete with each other, and consequently the determination of modal excitation region become more complicated. As the towing speed (or the shearing parameter) increases in sheared flow, the average RMS displacement does not regularly rise (or drop), but slightly fluctuates owing to changes of the participating mode and its excitation region. On the other hand, the average RMS stress gradually rises owing to higher-order modes being included. Moreover, it is found that the dominant frequency distributing along structural span significantly changes with the towing speed, and the length of the first dominant frequency gets smaller due to larger shearing parameter along with more intense competition between the participating modes.

1. Introduction

As the offshore industry moves towards deeper ocean, some floating platforms such as spar, tension leg and semi-submersible platforms have been put into services. Marine risers of these deep-ocean platforms are employed to transport gas and oil or optical and electrical information. The fundamental issues, such as vortex-induced vibration (VIV) and fatigue problems, of these long flexible risers experiencing ocean current (or wave) become more complicated as water depth increases. On the structural side, the dynamic characteristics of slender riser usually presents low-frequency and high-density natural modes due to its large structural flexibility. On the fluid side, the distribution of fluid field, in terms of velocity value and direction, is no longer uniform. Moreover, the shedding mode or frequency of wake vortex may vary, or even be grouped in cells, along structural span due to span-wise coupling of vortices (Mukhopadhyay et al., 1999; Lucor et al., 2001; Williamson and Govardhan, 2008). Therefore, VIV of a long flexible riser experiencing non-uniform flow often presents new phenomena such as multi-frequency (or called as multi-mode) VIV, travelling wave and wide–band random vibrations (Chen et al., 2016; Violette et al., 2010; Sarpkaya, 2004; Facchinetti, 2004; Huera-Huarte and Bearman, 2009). Since natural frequencies of slender riser are dense, i.e. the frequencies of adjacent modes are very close to each other, several modes with different frequencies are excited. Further understanding of multi-frequency VIV in non-uniform flow is significantly challenging.

In recent years, increasing researches about multi-frequency VIV have been reported (Vandiver et al., 1996; Huse et al., 1998; Lie and Kaasen, 2006; Dong and Karniadakis, 2005; Tang et al., 2007; Srinil, 2010; Huera-Huarte et al., 2006; Huang et al., 2011; Jaiswal and Vandiver, 2007; Tognarelli et al., 2004; Marcollo and Hinwood, 2002; Zhang et al., 2013), which were mostly based on experiments. Lie et al. implemented a large-scale tests (Huse et al., 1998; Lie and Kaasen, 2006), he studied if and under which circumstances the riser motions would be single-frequency or multi-frequency. He found that in general the

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https://doi.org/10.1016/j.petrol.2018.01.008
Received 12 September 2017; Received in revised form 14 December 2017; Accepted 3 January 2018
Available online 6 January 2018
0920-4105/

response was irregular (i.e. broad-banded) and that the degree of irregularity increases with the flow speed. In some tests distinct spectral peaks could be seen, corresponding to a dominant mode. Huera-Huarte et al. (2006). gave the force distribution of a riser in a stepped current by using experimental data and the finite element method. The experiments showed a correlation between the mean drag and the transverse response along the cylinder. Huang et al. (2011). used the measured data in uniform current to study the drag amplification and considered the spatial variation of amplitude. By using the semi-empirical wake oscillator model, Srinil (2010) found that multiple modal responses overlap in specific velocity ranges and the lock-in band width is mode-dependent. And, Zhang et al. (2013). found that the unstable zone may grow and a small excitation can induce large dynamic response.

As for the mechanism of multi-frequency VIV and its mode competition, it is still somewhat unclear and has drawn some interesting discussions. Jaiswal et al (Jaiswal and Vandiver, 2007). used the concepts of "time sharing'' to describe the "mode switching'' along the time coordinate, while Tognarelli et al. (2004). called it "space sharing''. Violette et al. (2010). performed a linear stability approach to identify the mode switching of two excited modes in cable VIV. He theoretically explained this behavior based on the linear stability approach, regarding different modes could be excited at different and coincident time instants. Marcollo and Hinwood (2002) examined the area where a cylinder's VIV could vary from single-frequency to multimodal. He found an interesting and unexpected mechanism that the damping region of higher-frequency mode may provide power-in effect to support other modes.

Still, there are some problems remain interesting and, particularly, mode competition and its expression need further studies. In this study, multi-frequency VIV in sheared fluid profiles, i.e. sheared flows with different shearing intensities, is examined and compared with precious single-frequency VIV. A nonlinearly numerical approach, where FEM is combined with the hydrodynamic model and mode competition can be particularly considered based on modal energy, is developed. Then the effects of towing speed (or shearing parameter) on displacement, stress and dominant frequency (or participating mode) distribution are studied, and mode competition during VIV is discussed.

2. Numerical simulation based on finite element method and the hydrodynamic model

The governing equation of a slender riser (see Fig. 1), generally regarded as a tensioned Euler beam, undergoing VIV can be written as

$$m\frac{\partial^2 y(z,t)}{\partial t^2}+\gamma\frac{\partial y(z,t)}{\partial t}+EI\frac{\partial^4 y(z,t)}{\partial z^4}-T\frac{\partial^2 y(z,t)}{\partial z^2}=f(z,t) \tag{1}$$

where $y(z,t)$ is the displacement which is in cross-flow direction (here

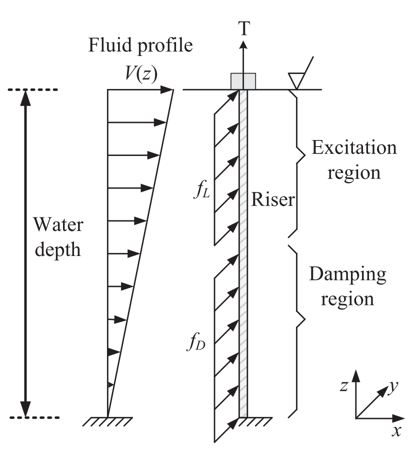

Fig. 1. Scheme of a riser in sheared fluid profile.

the coupling of cross-flow and in-line motion is not considered), m and γ are the structural mass and damping per unit length; EI and T are the bending stiffness and axial tension. $f(z,t)$ is the hydrodynamic force per unit length consisting of the vortex-induced lift force $f_L(z,t)$ and fluid force $f_D(z,t)$. Theoretically, to get the solution of Equation (1) is pretty difficult. Because the vortex-induced lift force $f_L(z,t)$ actually couples with body motion and can hardly have a precisely theoretical expression owing to complicated wake field during lock-in. And, by now, CFD of VIV still needs expensive cost of computation resource, especially for a deep water riser at high Reynolds number.

Here, a numerical approach based on finite element method and the hydrodynamic model is introduced to analyze the nonlinear simultaneously dynamic response of a riser undergoing VIV. This approach can capture the main VIV's traits during lock-in under an acceptable computation cost by avoiding CFD. More specifically, the riser is divided into finite beam elements upon which the hydrodynamic models are loaded. Then direct numerical integration is used to solve the governing equation and get dynamic response in time domain.

2.1. Structural finite element model

The slender riser is assumed as an Euler beam and can be uniformly divided into finite elements. Here, two-node Euler beam element (Craig, 1981) is used and the number of beam elements is N. For representativeness and simplicity, only one translation displacement y_i, $i=1,2,\ldots,N+1$, ($N+1$ is the total number of nodes), and one rotation $\theta_i, i=1,2,\ldots,N+1$ per node, are considered. The displacement function of the beam element is written as

$$y(\xi)=\sum_{i=1}^{2}\varphi_i^0(\xi)y_i+\sum_{i=1}^{2}\varphi_i^1(\xi)\theta_i \tag{2}$$

where $\varphi_1^0(\xi)=1-3\xi^2+2\xi^3$, $\varphi_2^0(\xi)=3\xi^2-2\xi^3$ and $\varphi_1^1(\xi)=(\xi-2\xi^2+\xi^3)/l_e$, $\varphi_2^1(\xi)=(\xi^3-\xi^2)/l_e$. ξ is the internal coordinate of beam element, $\xi=(z-z_1)/L$, $0\le\xi\le1$. l_e and L are the element length and overall structural length respectively. The element matrices are respectively as mass matrix $\mathbf{M}^e=\int_{l_e}\rho\boldsymbol{\varphi}^T\boldsymbol{\varphi}d\xi$, stiffness matrix $\mathbf{K}^e=\int_{l_e}\mathbf{B}^TK'\mathbf{B}d\xi$ and geometry stiffness matrix $\mathbf{K}_t^e=\int_{l_e}\overline{\mathbf{B}}^T\overline{\mathbf{B}}d\xi$, where $\boldsymbol{\varphi}=\left[\varphi_1^0\quad\varphi_1^1\quad\varphi_2^0\quad\varphi_2^1\right]$ is the element deformation function. Coefficient matrices $\mathbf{B}=\frac{d^2\boldsymbol{\varphi}}{dx^2}$, $\overline{\mathbf{B}}=\frac{d\boldsymbol{\varphi}}{dx}$ and $K'=EI$.

Then the governing equation of the riser with many degrees of freedom can be written as:

$$\mathbf{M}\ddot{\mathbf{Y}}+\mathbf{C}\dot{\mathbf{Y}}+KY=\mathbf{F} \tag{3}$$

where **M**, **C** and **K** are the structural mass, damping and stiffness matrices respectively, which can be assembled by the corresponding element matrices. **Y** and **F** are the displacement and load vector of the nodes.

For case of small structural damping, the Rayleigh damping can be used and is written as

$$\mathbf{C}=a\mathbf{M}+b\mathbf{K} \tag{4}$$

where a and b are positive constants of which values can be determined by experiments, or approximately, the natural frequencies of structure as follows:

$$a=\frac{2\omega_1\omega_2(\zeta_1\omega_2-\zeta_2\omega_1)}{\omega_2^2-\omega_1^2},\quad b=\frac{2(\zeta_2\omega_2-\zeta_1\omega_1)}{\omega_2^2-\omega_1^2} \tag{5}$$

where ζ_j, and ω_j $(j=1,2)$ are respectively damping ratio and natural frequency of mode j. Generally, the value of structural modal damping ratio is 3 percent, or $\zeta_1=\zeta_2=0.03$ (Craig, 1981; Vandiver and Li, 1999).

The vortex-induced lift force $f_L(z,t)$ and the drag force $f_D(z,t)$ exerted by the ambient fluid are applied at the nodes respectively in the excita-