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Nonlinear dynamics and impact load in float-over installation

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ABSTRACT

A time-domain 3 Degrees of Freedom model is developed to investigate nonlinear dynamics and impact loads during float-over installations, which generally involve multi-body interactions between wave-induced vessel motions and nonlinear constraint components. By replacing the time-consuming convolution in calculating the retardation function, a more efficient method, i.e. state-space model, is applied to evaluate part of the radiation force. The established model, incorporating the multi-body interactions, is applied to study the nonlinear impact on Leg Mating Unit (LMU) by considering the sway, heave and roll motions of the float-over system. The structural characteristics are considered when modelling the characteristics of LMU. The dynamic behaviors of a given system is investigated in the form of bifurcation diagrams, along with impact map, amplitude spectrum and power spectral density (PSD). It is found that bifurcation phenomena, or a large angle of docking cone could dominate the installation due to the increased impact loads.

1. Introduction

Float-over installation has been successfully applied in offshore operations due to relatively low costs and higher capacity. In this configuration, the topside modules will be offloaded from a transportation barge onto a fixed sub-structure (jacket) [1]. Guiding systems such as Leg Mating Unit (LMU), Deck Supporting Unit (DSU) and fenders are designed to provide the positive alignment and adequate control of the system motions. During the installation, the guiding pins on the topside module will induce great impact on the mating cones. In order to ensure efficiency and safety, it is vital to consider the nonlinear dynamics.

Significant efforts have been taken on the theoretical and numerical analysis for nonlinear dynamics of simple mechanical systems. For a single degree of freedom (DOF) system, the mass-spring-damper linear oscillator [2] and impact bilinear oscillator [3] are well developed. In their studies, the periodic and chaotic impacting motions were presented in terms of phase portraits, return maps and bifurcation diagrams. For the system which can be simplified as a 3 DOF model, the bifurcation phenomena of a given system were studied [4]. They analyzed the phenomena of periodic motions leading to chaos, where qualitative change occurs in the system. When applying the nonlinear analysis methods to the marine operations, Thompson et al. [5] introduced a bilinear spring to simulate the slack phenomena of mooring cables. Kim et al. [6] introduced a 3DOF maneuvering equation in the horizontal plane to analyze the coupled spreading mooring and riser dynamics. Nonlinear stability and bifurcation analysis were performed to obtain qualitative understanding of the dynamic behaviors of such a coupled system.

Generally, the dynamical behavior of float-over system considering the wave-induced LMU impacts, is extensively studied in time domain based on a linear potential flow approach. The numerical modelling of LMU, based on the characteristics of its geometry, were discussed in [7,8]. However, the motion response and hydrodynamic analysis were not fully investigated in their numerical models. Chen et al. [9] considered the heave motion of the float-over system and introduced a simple wave-induced impact model, where LMU was modelled as a bilinear impact oscillator. In their studies, single DOF model assumption for a realistic float-over system was well justified. Meanwhile, it is worried that the simple single DOF impact model might not demonstrate all the complex behaviors of the float-over installation.

In the present study, the nonlinear dynamics of a simplified 3DOF wave-induced impact model are considered. Assuming

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Nomenclature

M	Rigid body mass matrix
G	Restoring matrix
$\boldsymbol{A}(\omega)$	Frequency-dependent added mass
$\boldsymbol{B}(\omega)$	Frequency-dependent damping
$\boldsymbol{A}(\infty)$	Infinite frequency added mass
t	Time variable
$\boldsymbol{K}(t), \boldsymbol{K}(s)$	Retardation function in time and complex domain
$\tilde{\boldsymbol{K}}(s)$	Fitted retardation function
$\boldsymbol{f}^{exc}(t), \boldsymbol{f}^{exc}(\omega)$	1st excitation force in time and frequency domain
$\hat{A},\hat{B},\hat{C}$	State-space model matrices
μ	Vector of memory effect force
z	State variable
j	Imaginary unit
ω	Angular frequency
C_V	Linear viscous damping matrix
K_H^{DSU}, K_V^{DSU}	Horizontal stiffness and vertical stiffness of LMU
K^{LMU}, C^{LMU}	Stiffness and damping coefficient of LMU
K^{fender}	Stiffness of fender
x	State space of system, $\boldsymbol{x} = \left[\boldsymbol{\eta}^b, \boldsymbol{\nu}^b, \boldsymbol{\eta}^t, \boldsymbol{\nu}^t\right]^T$
$\{n\}, \{b\}$	Global coordinate system and body-fixed coordinate system
$\boldsymbol{f}^{\boldsymbol{DSU}}(\boldsymbol{x})$	Vector of DSU force
$\boldsymbol{f}^{\boldsymbol{LMU}}(\boldsymbol{x})$	Vector of LMU force
η^b, η^t	Displacement vector of barge and topside with respect to $\{n\}$
ν^b, ν^t	Velocity vector of barge and topside with respect to $\{b\}$
W^t	Gravity vector of topside
$\boldsymbol{J}(\boldsymbol{\eta})$	Motion transformation matrix
ρ_{XY}	Cross correlation coefficient
φ	Phase, $\varphi = mod(t, T)/T$
α	Angle of docking cone on LMU
T	The transform matrix between the oblique coordinate system and rectangular coordinate system
H_s	Significant wave height
T_P, ω_P	Spectral peak period and angular frequency
γ	Peak shape parameter for spectra
$A1, A2$	The windward and leeward LMUs

incident waves are beam-on, the numerical model accounts for motion responses in heave, sway and roll. A combination of the wave-induced dynamics, and the piecewise linear impacts, is applied to investigate the characteristics of a float-over system. In addition to the harmonic response, one can expect to find the conditions for which sub-harmonic, quasi-periodic and chaotic responses do occur. We numerically studied the motion characteristics by using the nonlinear analysis tools such as bifurcation diagram, impact map and Fast Fourier Transform (FFT). Parametric analysis are also performed, where only angle of docking cone is varied, to reveal the effect of such a structural parameter of LMU. Conclusions are drawn at the end of this study.

2. Mathematical model

A typical float-over system consists of a transportation barge, topside module and a simplified guiding systems including LMU, DSU and fenders (see Fig. 1). Fenders are used to reduce stresses in the jacket and barge, and limit the relative motions. DSU, installed on the barge, will support the topside module. And LMU, installed on the jacket, will receive the guiding pin on the topside module. Both of them are designed as shock absorbers to provide vertical and horizontal motion and forces during operation. To facilitate the numerical simulations, three coordinate systems are introduced in this system: (i) topside-fixed coordinate system $o_t - y_t z_t$ with its origin located at the center of gravity (COG) of the topside (ii) barge-fixed coordinate system $o_b - y_b z_b$ with its origin located at still waterline. (iii) global coordinate system. Lowering velocity is relatively small and not considered in the model.

2.1. Calculation of hydrodynamic forces—state space model

The steady-state response of float-over operation under wave excitation is generally analyzed in frequency domain using the well-known hydrodynamics program WAMIT [10]. Through the frequency-domain analysis, we obtain the frequency-dependent hydrodynamics coefficients, such as added mass and damping coefficients. Based on these frequency-dependent coefficients, Cummins [11] introduced a seakeeping equations in time domain:

$$[\boldsymbol{M} + \boldsymbol{A}(\infty)]\ddot{\boldsymbol{x}}(t) + \int_0^t \boldsymbol{K}(t-\tau)\dot{\boldsymbol{x}}(\tau)d\tau + \boldsymbol{G}\boldsymbol{x}(t) = \boldsymbol{f}^{exc}(t) \quad (1)$$

where, $\boldsymbol{x}(t)$ is the perturbation with respect to global coordinate system $\{n\}$. The retardation function $\boldsymbol{K}(t)$ is given by

$$\boldsymbol{K}(t) = \frac{2}{\pi}\int_0^\infty \boldsymbol{B}(\omega)\cos(\omega t)d\omega \quad (2)$$

where, $\boldsymbol{B}(\omega)$ is the frequency-dependent damping matrix.

Ogilvie [12] established the relationship between $\boldsymbol{A}(\omega), \boldsymbol{B}(\omega)$ and $\boldsymbol{K}(\boldsymbol{t})$

$$\boldsymbol{A}(\omega) = \boldsymbol{A}(\infty) - \frac{1}{\omega}\int_0^\infty \boldsymbol{K}(t)sin(\omega\tau)d\tau \quad (3)$$

$$\boldsymbol{B}(\omega) = \int_0^\infty \boldsymbol{K}(t)cos(\omega\tau)d\tau \quad (4)$$

Then

$$\boldsymbol{K}(j\omega) = \int_0^\infty \boldsymbol{K}(\tau)e^{-j\omega\tau}d\tau = \boldsymbol{B}(\omega) + j\omega[\boldsymbol{A}(\omega) - \boldsymbol{A}(\infty)] \quad (5)$$

This type of model in Eq. (1) involves the convolution terms

$$\boldsymbol{\mu} = \int_0^t \boldsymbol{K}(t-\tau)\dot{\boldsymbol{x}}(\tau)d\tau \quad (6)$$

However, Eq. (6) is time-consuming for numerical simulation [13]. To overcome this problem, the state-space model [14] was used to replace the convolution integral. We take the Laplace transform of the state-space model in Eq. (6), obtaining the corresponding transfer matrix:

$$\boldsymbol{\mu}(s) = \tilde{\boldsymbol{K}}(s)\dot{\boldsymbol{x}}(s) \quad (7)$$

$$\tilde{\boldsymbol{K}}(s) = \frac{P(s)}{Q(s)} = \frac{p_{n-1}s^{n-1} + p_{n-2}s^{n-2} + \ldots + p_1 s + p_0}{s^n + q_{n-1}s^{n-1} + \ldots + qs + q_0} \quad (8)$$