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Nonlinear Approaches in Engineering Application

Automotive Engineering Problems

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The beauty of nonlinear approaches to engineering problems is that you must be talented to get even a wrong result.

Dedicated to Xinming *and* Mojgan

Preface

This book is the eighth volume in the series of Nonlinear Approaches in Engineering Applications, organized by the editors. This series is collecting individual application on engineering problems in which the nonlinearity is quite important. Those system have been introduced, and modeled mathematically, in such a way that their nonlinearities have been used to make the system to work better regarding the objectives of the problems.

The nonlinear analysis, techniques, and applications have been developed in the past two to three centuries when the linear mathematical modeling of natural dynamical phenomena appeared not to be exact enough for some practical applications. The positive aspects of linear approximation of dynamic phenomena are simplicity and solvability. Linear approximation of a system provides us with the simplest model acting as the base and standard for which other nonlinear models should approach when the nonlinearities become negligible. Solvability is another positive characteristic of all linear systems. These two characteristics provide us great ability and desire to model dynamic systems linearly. However, the linear model of many systems cannot provide solutions to be good enough approximation of the real system behavior. For such systems, considering nonlinearities of the phenomena is unavoidable. Although the nonlinear approximation of a system provides us with a better and more accurate model, it also provides us with several complications. One of them is that it makes us to search for indirect methods to gain some information of the possible solutions. Due to the nonlinearity and complexity of nonlinear systems, usually, it is very difficult or impossible to derive any analytical and closed-loop solutions for the systems. In solving or simulating the nonlinear systems, we have to rely on approximate or numerical methods, which may only provide approximate results while errors are unavoidable during the processes of generating the approximate results.

Level of the Book

This book aims at engineers, scientists, researchers, engineering, and physics students of graduate levels, together with the interested individuals in engineering, physics, and mathematics. This book focuses on application of the nonlinear approaches representing a wide spectrum of disciplines of engineering and science. Throughout the book, great emphases are placed on engineering applications, physical meaning of nonlinear systems, and methodologies of the approaches in analyzing and solving for the systems. Topics that have been selected are of high interest in engineering and physics. An attempt has been made to expose the engineers and researchers to a broad range of practical topics and approaches. The topics contained in the present book are of specific interest to engineers who are seeking expertise in modern applications of nonlinearities.

The primary audience of this book are researchers, graduate students, and engineers in mechanical engineering, engineering mechanics, electrical engineering, civil engineering, aerospace engineering, mathematics, and science disciplines. In particular, the book can be used for training graduate students as well as senior undergraduate students to enhance their knowledge by taking a graduate or advanced undergraduate course in the areas of nonlinear science, dynamics and vibration of discrete and continuous systems, structure dynamics, and engineering applications of nonlinear science. It can also be utilized as a guide to the readers' fulfillment in practices. The covered topics are also of interest to engineers who are seeking to expand their expertise in these areas.

Organization of the Book

This book is a collection of 10 important problems set in 2 parts: Modeling Dynamic Systems and Applied Dynamic Systems. Both parts are focused on applications of practical engineering problems. There are five chapters in Part 1. Chapter 1 is on three-dimensional nonlinear vibration model and response characteristics of deep-water riser-test pipe system, showing how mechanical vibrations can be used in extracting underground liquids such as water and oil. Chapter 2 deals with model-prototype experiment in vehicle dynamics. Chapter 3 is on flexible mechanisms. Chapter 4 is on modeling and analysis of tire-road separation problem in vehicle vibrations. Chapter 5 is on employing neural network to solve nonlinear differential equations. There are five chapters in Part 2. Chapter 6 is on interesting engineering question from energy point of view, if writing from left to right is better or writing from left to right. This question has been answered by a robotic simulation. Chapter 7 discusses the problems in measurement of the sea level rise. Chapter 8 is on further discussion on problems of sea level rise. Chapter 9 illustrates how game theory can be used in solving engineering problems. Chapter 10 studies the nonlinear pull-in instability problem in micro- and nano-mechanisms.

Each of the chapters covers an independent topic along the line of nonlinear approach and engineering applications of nonlinear science. The main concepts in nonlinear science and engineering applications are explained fully with necessary derivatives in details. The book and each of the chapters are intended to be organized as essentially self-contained. All necessary concepts, proofs, mathematical background, solutions, methodologies, and references are supplied except for some fundamental knowledge well known in the general fields of engineering and physics. The readers may therefore gain the main concepts of each chapter with as less as possible the need to refer to the concepts of the other chapters and references. Readers may hence start to read one or more chapters of the book for their own interests.

Method of Presentation

The scope of each chapter is clearly outlined and the governing equations are derived with an adequate explanation of the procedures. The covered topics are logically and completely presented without unnecessary overemphasis. The topics are presented in a book form rather than in the style of a handbook. Tables, charts, equations, and references are used in abundance. Proofs and derivations are emphasized in such a way that they can be straightforwardly followed by the readers with fundamental knowledge of engineering science and college physics. The physical model and final results provided in the chapters are accompanied with necessary illustrations and interpretations. Specific information that is required in carrying out the detailed theoretical concepts and modelling processes has been stressed.

Prerequisites

The present book is primarily intended for researchers, engineers, and graduate students, so the assumption is that the readers are familiar with the fundamentals of dynamics, calculus, and differential equations associated with dynamics in engineering and physics, as well as a basic knowledge of linear algebra and numerical methods. The presented topics are given in a way to establish as conceptual framework that enables the readers to pursue further advances in the field. Although the governing equations and modelling methodologies will be derived with adequate explanations of the procedures, it is assumed that the readers have a working knowledge of dynamics, university mathematics, and physics together with theory of linear elasticity.

Acknowledgments

This book is made available under the close and effective collaborations of all the enthusiastic chapter contributors who have the expertise and experience in various disciplines of nonlinear science and engineering applications. They deserve sincere gratitude for the motivation of creating such book, encouragement in completing the book, scientific and professional attitude in constructing each of the chapters of the book, and the continuous efforts toward improving the quality of the book. Without the collaboration and consistent efforts of the chapter contributors, the completion of this book would have been impossible. What we have at the end is a book that we have every reason to be proud of.

It has been gratifying to work with the staff of Spinger through the development of this book. The assistance provided by the staff members have been valuable and efficient. We thank Spinger for their production of an elegant book.

Regina, SK, Canada — Liming Dai
Melbourne, VIC, Australia — Reza N. Jazar

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Part I
Modeling Dynamic Systems

Three-Dimensional Nonlinear Vibration Model and Response Characteristics of Deep-Water Riser-Test Pipe System

Xiaoqiang Guo, Liming Dai, Jun Liu, Qingyou Liu, and Yufa He

Abbreviations

3D	three-dimensional
BOP	blowout preventer
CF	crossflow
CFD	computational fluid dynamics
IL	inline
RMS	root mean square
RTS	riser-test pipe system
VIV	vortex-induced vibration

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Nomenclature

m^*	mass ratio of riser
E	elastic modulus of the RTS, Pa
$\upsilon_i', i = x, y, z$	first-order derivative of riser displacements versus z
$\dot{\upsilon}_i, i = x, y, z$	first-order derivative of riser displacements versus time
ρ_υ	density of the riser, kg/m^3
$F_x(z, t)$	contact/impact force of riser-test pipe in x-directions, N
$F_z(z, t)$	friction force of riser-test pipe in z-directions, N
$F_L(z, t)$	lateral lift in the CF direction, N
ω_υ	natural angular frequency of riser
L_υ	length of riser, m
D_o	riser outer diameter, m
m_i	the mass of the gas per unit length (kg)
$S_i, i = x, y, z$	displacement components of the test pipe, m
$S_i'', i = x, y, z$	second derivative of test pipe displacements versus z
$f_x(z, t)$	high-speed fluid impact load in test pipe in x-direction, N
$f_z(z, t)$	high-speed fluid impact load in test pipe in z-direction, N
ω_s	natural angular frequency of test pipe
$w_s(=m_s g)$	weight of test pipe per unit length, N
V_r	relative velocity between the fluid and the riser, m/s
U_c	outflow velocity of the riser, m/s
$F_D{}'$, C_D	component forces of the fluctuating drag force and corresponding coefficient
$q_i, i = x, y$	dimensionless wake oscillator variables in IL and CF directions
S_t	Strouhal number
$R_i, i = 1, 2$	radius of riser and test pipe, m
$\omega_i, i = 1, 2$	axial displacements of the riser and test pipe, m
$E_i, i = 1, 2$	elastic modulus of the riser and test pipe material, Pa
$q(x)$	uniform load distribution for riser and test pipe, N
ρ_i	density of gas in the test pipe, kg/m^3
$\alpha(t)$	deflection angles of test pipe in x- direction, rad
$\alpha(s)$	inclination angle, rad
K_{U}	rotational stiffness of the upper flexible joint
$u_{\mathrm{boat}}(t)$	heave displacement of the platform, m
m_{p}	mass of platform, kg
$\eta(t)$	surface displacement of random wave, m
$\hat{\omega}_i$	circular frequency of the ith harmonic, Hz
a_i	amplitude of the ith harmonic component, m
$S(\omega)$	random wave spectrum
ω	circular frequency, Hz
$T_{1/3}$	significant period of the wave, s
T_p	peak period of the wave, s
σ	peak shape coefficient

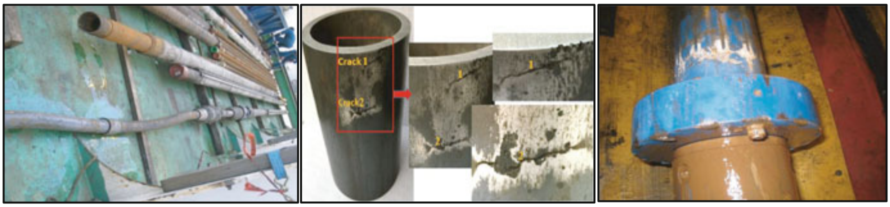
$F_p(t)$	exciting force of the random wave on the heave plate
$J_1(\cdot)$	first order Bessel function of first kind
z_{plate}	depth of heave plate, m
$\mathbf{d}$	displacement vector of riser unit
$\varphi_i, i = x, y, z$	vibration shape function of riser and test pipe unit
$\mathbf{F}(t)$	load column vector
$\mathbf{M}(t)$	matrices of the overall mass
ρ_{p}	density of the actual RTS
ρ_{m}	density of the RTS in the simulation experiment
λ	radial similarity ratio
$\upsilon_i, i = x, y, z$	displacement components of riser, m
A_υ	cross-sectional area of the riser, m^2
$\upsilon_i'', i = x, y, z$	second derivative of riser displacements versus z
I_υ	polar moment of inertia of the riser, m^4
m_υ	mass of the per unit length riser, kg
$F_y(z, t)$	contact/impact force of riser-test pipe in y-directions, N
$c_\upsilon(=2m_\upsilon\omega_\upsilon\zeta)$	structural damping coefficient of riser
$F_D(z, t)$	drag force in the IL direction, N
w_g	buoyant weight of riser per unit length, N
ρ_w	density of the sea-water, kg/m^3
A_s	cross-sectional area of the test pipe, m^2
m_s	mass of the per unit length test pipe, kg
$S_i', i = x, y, z$	first-order derivative of test pipe displacements versus z
$\dot{S}_i, i = x, y, z$	first-order derivative of test pipe displacements versus time
$f_y(z, t)$	high-speed fluid impact load in test pipe in y-direction, N
$c_s(=2m_s\omega_s\zeta)$	structural damping coefficient of test pipe
$v_i, i = x, y, z$	absolute velocities of the internal high-speed fluid (m/s)
V	fluid flow velocity in the test pipe, m/s
$\overline{C}_d$	coefficient of steady-state drag force
$\overline{C}_l$	coefficient of steady lift force
$F_L{}', C_L$	fluctuating lift force and corresponding coefficient
$z_i, i = 1, 2$	radial distance from the contact point of pipe to the inner wall of riser, m
ω_s'	vortex shedding frequency
r	horizontal distance from the contact point to the test pipe axis, m
F	contact load of riser-test pipe, N
$\mu_i, i = 1, 2$	Poisson's ratio of riser and test pipe material
ξ	friction coefficient between the riser and test pipe
A_i	cross-sectional area of the wellbore, m^2
$\varphi(t)$	deflection angles of test pipe in y- direction, rad
$\varphi(s)$	azimuth, rad
K_{L}	rotation stiffness of the BOP

$B_i, i = 1, 2$	heave radiation and heave viscous damping
A_w	area of the platform at sea level, m^2
$\overline{F}_z$	random heave wave exciting force on platform, N
ε_i	initial phase of the *i*th harmonic component, rad
$\Delta\omega$	frequency step
f	Frequency, Hz
$H_{1/3}$	significant wave height, m
f_p	peak frequency of the wave, Hz
γ	peak parameter
R	platform radius, m
$F_s(t)$	exciting force of the random wave on the platform body
d	draft of platform, m
B_{plate}	width of heave plate, m
$\overline{\mathbf{d}}$	displacement vector of test pipe unit
$\mathbf{D}$	matrix of overall displacement
$\mathbf{K}(t)$	matrices of the overall stiffness
$\mathbf{C}(t)$	matrices of the overall damping
E_{p}	elastic modulus of the actual RTS
E_{m}	elastic modulus of the RTS in the simulation experiment

1 Introduction

With the increasing demand for oil and gas resources in the world, the exploitation trend of offshore oil and gas resources gradually develops from shallow water (water depth is less than 500 m) to deep water (water depth is between 500 m and 1500 m). The riser-test pipe system (RTS) is the core equipment for deep-water oil and gas exploitation but the weakest equipment. Compared with conventional water depth testing conditions, the RTS is subjected to greater risks in deep-water test conditions; these risks are mainly caused by severe non-periodic vibrations of the riser and test pipe (RTS) induced by the vortex induced effect on riser, flow-induced effect on test pipe, nonlinear contact/collision of the tube in tube, and longitudinal/transverse coupling effect, thereby making the RTS more susceptible to buckling deformation (Fig. 1a), fatigue fracture (Fig. 1b), and friction perforation (Fig. 1c) (Zhou et al. 2013). Once the system structure is damaged, it will lead to serious offshore oil and gas accidents, resulting in significant economic losses and environmental pollution. Therefore, the three-dimensional (3D) nonlinear vibration model and response characteristics for deep-water RTS should be investigated.

In early vortex-induced vibration (VIV) studies, most of the work focused on rigid cylinders (Sarpkaya 1979; Govardan and Williamson 2000; Bearman 2003), in which the general VIV mechanism and law, such as the frequency-locked phenomenon (Dahl et al. 2010) and lagging behavior (Facchinetti et al. 2004). In recent years, driven by offshore oil and gas exploitation, more and more attention



(a) (b) (c)

Fig. 1 Failure forms of the riser-test pipe system (RTS). (**a**) Buckling deformation (**b**) Fatigue fracture (**c**) Friction perforation

has been paid to the VIV problem of flexible cylinders in which the aspect ratio is a very important parameter. Physical experiments (Chaplin et al. 2005; Trim et al. 2005; Vandiver et al. 2009; Huera-Huarte et al. 2014; Gao et al. 2015) and computational fluid dynamics (CFD) numerical simulations (Newman and Karniadakis 1997; Bourguet et al. 2011, 2013; Mao et al. 2019, 2020) are the two most common methods in these studies, and remarkable progress was made. However, when the aspect ratio of a cylinder is large or a solid model is used, physical experiments usually become very expensive and impractical, and CFD numerical simulation is too time-consuming and difficult. Therefore, in the VIV study of risers, there are relatively few works that consider a large aspect ratio or the actual size. In addition to a large aspect ratio, the impact of the ocean environment load on the VIV behavior of a riser is significant. The VIV response mechanism of a flexible riser under shear flow was examined by Mathelin and Langre (2005) using a wake oscillator model presented by (Facchinetti et al. 2004). Since the VIV amplitude in the crossflow (CF) direction is larger than that in the inline (IL) direction, most work has focused on the VIV in the CF direction (Khalak and Williamson 1999). The effects of the flow velocity, top tension, and pipe diameter on VIV behavior in the crossflow direction of a riser were studied by Xu et al. (2017) and He et al. (2017) using a VIV model. In the above studies, the VIV behavior in the IL direction and its influence were not taken into account. However, it was found in the work of Jauvtis and Williamson (2003) that as the mass ratio $m^*(=4m_s/\rho\pi D^2)$, the ratio of the structural quality to the mass of discharged fluid was less than 6.0, and the IL vibration of a cylinder could not be neglected. The VIV characteristics study of a rigid cylinder presented by Gu et al. (2016), Martins and Avila (2019) and Gao et al. (2019) also showed that the effect of the IL vibration was significant. In our recent work (Liu et al. 2018a, b, 2019), the response characteristics of VIV of marine risers with consideration of the coupling effects of the CF and IL vibration were investigated. It was found that the frequency locking effect in the uniform flow and the multi frequency effect in the shear flow for the IL vibration.

Vibration of tubular structure caused by inside flow has attracted some researchers' attentions. In the early research, preliminary research was conducted on string vibration under the action of internal flow (Aitken 1878), and initially confirmed the phenomenon of pipe vibration induced by fluid in pipe without elaborating the interaction mechanism between fluid and pipe (Shilling and Lou 1980). Subsequently, many scholars carried out detailed research on the string vibration model, and established the calculation method of fluid force (Deng 2006), the string vertical vibration (Paidoussis et al. 2008; Ju and Tong 2014), the lateral vibration (Zhang and Miska 2003; Bagdatli 2015), and the fluid-structure coupled vibration model (Dai et al. 2014; Yu 2017). In recent years, some scholars (Xing and Liang 2015; Li 2016; Liu et al. 2018a, b) have found that the longitudinal/lateral coupling effects of slender tubular columns cannot be ignored. Aimed at the static contact problems of slender structures, a few researchers (Hong et al. 1982; Ding et al. 1989; Shen and Ding 1990; Zhang and Song 2015) have tried to give the calculation methods of contact force between a beam and support structure, and the correctness of the methods was verified by experimental data. Moreover, the bracing effect of the outer pipe was taken into account by some researchers (Tan 2005; Wang et al. 2015; Li 2017) to analyze the static buckling deformation of a tubing string. About the dynamic contact/collision problem of slender structures, the commercial software such as ANSYS and ABAQUS were used by researchers (Zhu et al. 2007; Liu et al. 2016; Yang et al. 2016) to investigate the impact force and friction force in the flow-induced vibration of slender structures in vertical well. Also, in our recent work (Liu et al. 2020; Guo et al. 2021a, b), the flow-induced nonlinear vibration model of tubing string in conventional oil and gas wells was established using micro-finite element and energy methods along with the Hamilton variational principle, which considers the longitudinal/lateral coupling effect of tubing string and the nonlinear contact collision effect of tubing-casing. In summary, the interaction between riser and test pipe is ignored in the above studies, which make the calculation results by the single vibration model not in accordance with the actual. Especially in the deep water test condition, with the increase of length diameter ratio, the interaction between riser and test pipe cannot be ignored.

In this study, the 3D nonlinear vibration model of deep-water RTS is established. The Lagrange and cubic Hermite functions are used to discretize the governing equations. Then, the incremental form of Newmark-β and Newton-Raphson are used to solve the 3D nonlinear vibration model. Meanwhile, a vibration test bench for the RTS is designed using similarity principle, and the correctness and effectiveness of the proposed 3D nonlinear vibration model are verified by comparing with experimental data. Finally, the vibration characteristics of the RTS in the South China Sea are analyzed.

2 3D Nonlinear Vibration Model of the RTS

2.1 Nonlinear Vibration Control Equation of the RTS

In this section, the 3D vibration control equations of infinitesimal riser-test pipe (RTS) were established through the energy method and Hamilton variational

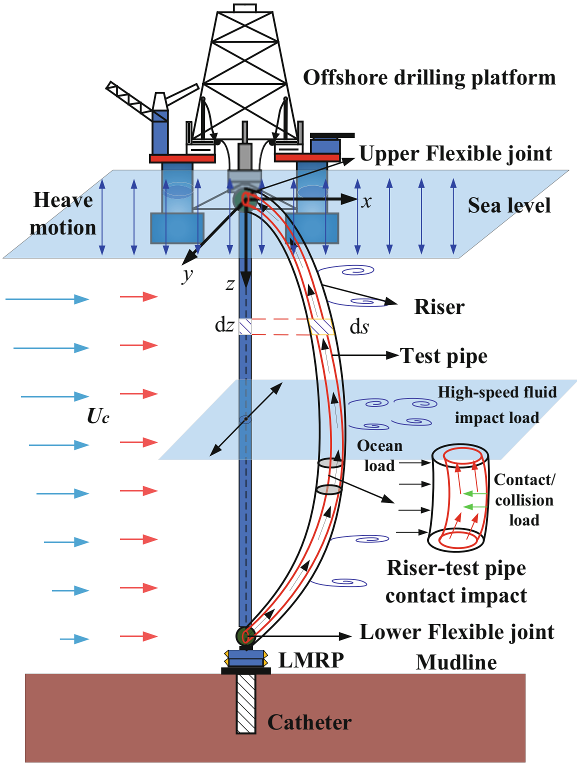


Fig. 2 Structure diagram of the RTS

principle. The infinitesimal segment of the RTS is too short, such that it can be regarded as a straight segment. Therefore, a coordinate system is established in which the depth direction is set as z-axis, the horizontal direction (the IL direction) is set as x-axis and the y-axis (the CF direction) satisfies the right-hand rule (Fig. 2). The following basic assumptions are made before modeling.

① The mechanical property of the material of riser and test pipe is ideal isotropic and elastic.
② The gravity and frictional resistance are evenly distributed on the tubing element.
③ The test pipe axis is coincided with the riser axis at initial moment, and the gravity of the RTS acts on itself at initial moment.
④ The friction coefficient at each location of the system is constant.

(1) Vibration control equation of riser

Based on the small deformation hypothesis and the Kirchhoff hypothesis (Liu et al. 2019), the three displacement field components u_1, u_2, and u_3 along the coordinates x, y, and z, respectively, can be written as: