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Modal damping ratio analysis of dynamical system with non-stationary responses

Da Tang[a], Ran Ju[a,*], Qianjin Yue[b], Shisheng Wang[c]

[a] Computer Science & Technology College, Dalian University of Technology, Dalian 116024, China
[b] School of Ocean Science and Technology, Dalian University of Technology, Dalian 116024,China
[c] China National Offshore Oil Corporation (CNOOC) Research Institute, Beijing 100027, China

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ABSTRACT

During the long term monitoring of the structure, the damping ratio reflects the characteristics of the structure from perspective of energy loss, and its changes can reflect the structural damages to some extent. But in the structural modal analysis based on the prototype measurement, the damping ratio identification results are difficult to identify, especially for the non-stationary structural responses, besides, ICA loses accuracy in the presence of higher-level damping. In allusion to these problems, a modal identification method based on the stationary filter-time frequency independent component analysis (filter-TFICA) is proposed. First a stationary filter using moving average is used to eliminate the non-stationary components. Then the modal identification of mooring system is incorporated into the blind source separation formulation where TFICA is introduced. The validity of the proposed method is confirmed through the identification of a multi degree of freedom numerical simulation system under non-stationary random excitations. Further, the prototype measurement data of the floating production, storage and offloading (FPSO) single point mooring system are analyzed. Compared with the traditional method, the differences of identified damping ratios at the same frequency range by the proposed method are smaller. The distribution of identified modal frequencies is more scattered and the mutually coupled modes are decoupled well. Each of the calculated motion tracks corresponding to the mode shapes presents single form of motion more prominently.

1. Introduction

The objective of health monitoring of the large-scale structures is to identify the modal parameters, diagnose the structural damages and ensure the safety in production. The key problem of structural analysis from perspective of energy loss is to identify the modal damping ratio accurately.

During the prototype measurement of the offshore platform FPSO single point mooring system, the mooring motions have obviously non-stationary characteristics and mooring response data show a significant change in both mean and variance. Therefore, the identification of the modal damping ratio is more difficult.

The problem of modal damping ratio identification has drawn much attention. Kim identified the nonlinear single degree of freedom damping ratio of the rolling motion of a FPSO using Hilbert transform [1]. There are also researches on modal damping ratio identification under non-stationary response. In Lin's research, the random decrement method based on difference was proposed to deal with the non-stationary response under the assumption that the non-stationary ambient excitation is zero-mean [2]. Avendaño-Valencia analyzed modal parameters of a multi degree of freedom structure through non-stationary time varying auto regressive (AR) model, moving average (MA) model and auto regressive moving average (ARMA) model [3]. Andrzei identified modal parameters of a non-stationary system with the wavelet based adaptive filtering method [4]. In the researches mentioned above, stationary filter based on difference, time sequence model and wavelet were used, the emphasis was placed on the modal parameter identification of non-stationary system with multi degree of freedom, but effective methods to ensure the accuracy of damping ratio identification were not mentioned.

The modal identification problems of offshore platform have also drawn much attention [5]. There are researches on instantaneous frequency identification of FPSO single point mooring system response and complex modal parameter identification of mooring system [6,7]. Tang and Wang identified modal parameters of FPSO

* Corresponding author.
E-mail address: juran@mail.dlut.edu.cn (R. Ju).

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single point mooring system, but the difference among damping ratios identified at the same frequency range is large and modal coupling exits [8].

The structural modal identification can be seen as a blind source separation (BSS) problem. In BSS problem, the modal responses are statistically independent signals and they are viewed as source signals. On the other hand, the system responses are viewed as mixed signals. The target is to estimate the modal responses using the output responses only [9]. Independent component analysis (ICA), which uses higher order statistics as independence criterion and separates the independent source signals from the mixed signals, is often used in the field of structural modal identification [10]. The impulse-like signal caused by structural damage was extracted from the wavelet transform of system response using ICA [11]. But it has been shown that ICA loses accuracy in the presence of higher-level damping [12] and improving the modal parameters identification effect is an important issue in this field.

As the motion of offshore platform FPSO single point mooring is slow, if the mooring can be seen as a time invariant structure in a short time [8] and the modal parameters can be identified with the method based on ICA. As a result, a modal identification method based on stationary filter-time frequency independent component analysis (filter-TFICA) is proposed. The non-stationary components caused by ocean ambient loads in the response are filtered out by a stationary filter using moving average, which makes it more beneficial for ICA to estimate the modal response from the stationary response. In the simulation experiment the effectiveness of the proposed method is verified. Through analysis of the prototype measurement data, the modal parameters of FPSO single point mooring system are identified. Compared with the method used in research [8], the identified results of damping ratio at the similar frequency are more close, which presents a better identification accuracy on damping ratio. The proposed method is also compared from perspective of the modal distribution and motion tracks of the mooring system, which also has a better identification results.

2. Modal analysis of dynamical system under non-stationary response

2.1. Non-stationary response analysis using the stationary filter

The dynamic equation of a linear time invariant system is

$$\mathbf{M}\ddot{x}(t) + \mathbf{C}\dot{x}(t) + \mathbf{K}x(t) = \mathbf{f}(t) \quad (1)$$

In Eq. (1), **M**, **C** and **K** are mass matrix, damping matrix and stiffness matrix respectively, $\mathbf{f}(t)$ is external force vector added to the system, $\ddot{\mathbf{x}}(t)$, $\dot{\mathbf{x}}(t)$ and $\mathbf{x}(t)$ are acceleration, velocity and displacement responses of the system respectively. Assuming that the dynamic system is a proportional damping system with n degree of freedom, the displacement response can be expressed as:

$$\mathbf{x}(t) = \Phi\mathbf{q}(t) = \sum_{i=1}^{n} \boldsymbol{\phi}_{\mathbf{i}} q_i(t) \quad (2)$$

where $\mathbf{q}(t) = [q_1(t), q_2(t), \ldots q_n(t)]^T$ and $\mathbf{x}(t) = [x_1(t), x_2(t), \ldots x_n(t)]^T$ are displacement response and modal response respectively, $\boldsymbol{\Phi}$ is the mode matrix, $\boldsymbol{\phi}_{\mathbf{i}}$ is the column i of $\boldsymbol{\Phi}$, which is known as mode shape and associated with modal response $q_i(t)$.

As the offshore platform is under complex ambient excitation, mooring response usually has obvious non-stationary characteristics, which makes the random decrement of the response deviate from the exponential decay signal and has a bad effect on modal identification. The non-stationary components in the mooring response can be filter out by the stationary filter using moving average.Let

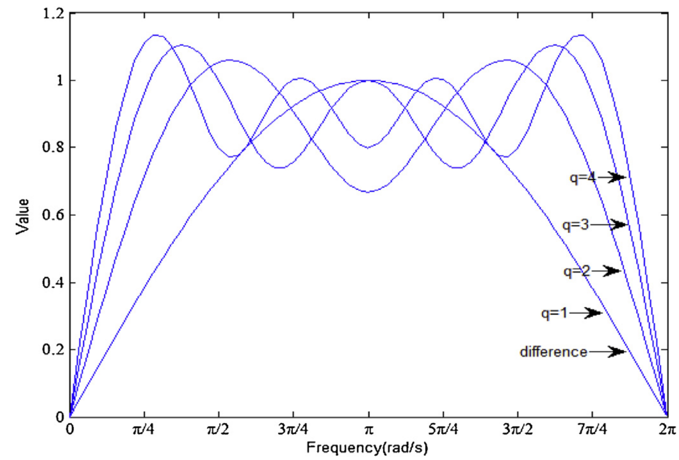

Fig. 1. The frequency domain of spectrum of the filter $h_Z(j\omega, q)$.

$$p(t) = f(t) - (q+1)^{-1}\sum_{i=0}^{q} f(t - i \times \Delta t) \quad (3)$$

In Eq. (3), i represents an integer (0, 1 ...), Δt represents an arbitrary time interval, $(q+1)^{-1}\sum_{i=0}^{q} f(t - i \times \Delta t)$ represents the unilateral moving average of $f(t)$, q is the order of moving average model and is called filter order in the left part of this paper. If q is chosen appropriately, $p(t)$ will become a stationary random process. On the basis of Eq. (1), for any time interval Δt and any integer i, Eq. (1) becomes

$$\mathbf{M}\ddot{\mathbf{x}}(t + i \times \Delta t) + \mathbf{C}\dot{\mathbf{x}}(t + i \times \Delta t) + \mathbf{K}\mathbf{x}(t + i \times \Delta t) = \boldsymbol{f}(t + i \times \Delta t) \quad (4)$$

Let

$$z(t) = x(t) - (q+1)^{-1}\sum_{i=0}^{q} x(t - i \times \Delta t) \quad (5)$$

The following equation can be obtained from Eq. (3), Eq. (4) and Eq. (5):

$$\mathbf{M}\ddot{z}(t) + \mathbf{C}\dot{z}(t) + \mathbf{K}z(t) = \mathbf{p}(t) \quad (6)$$

In Eq. (6), $z(t)$ is the stationary system response under the stationary external excitation $p(t)$, because through Eq. (5), the non-stationary external excitation is transformed into stationary excitation and the system response becomes stationary signal.

Let $\Delta t = 1$, then the Fourier transform of Eq. (6) can be express as

$$Z(j\omega) = [1 - (q+1)^{-1}\sum_{k=0}^{q} e^{-j\omega k}]X(j\omega) = h_Z(j\omega, q)X(j\omega) \quad (7)$$

$h_Z(j\omega, q)$ is viewed as a stationary filter and its frequency domain of spectrum is shown in Fig. 1. The non-stationary components with the frequency close to 0 and 2 π will be filtered out by $h_Z(j\omega, q)$. But the upper graphic of the spectrum shown in Fig. 1 are not stable and the power of the filtered signal will be distorted after being filtered by $h_Z(j\omega, q)$, which affects the accuracy of the modal parameters identification. In order to solve this problem, the stationary filter was improved as

$$\tilde{h}_Z(j\omega, q) = \frac{1}{n}\sum_{q=1}^{n} h_Z(j\omega, q)(n \geq 2) \quad (8)$$