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Modified PolyMAX for parametric estimation of Structures with modal interference †

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Abstract: The topic of this paper is an improved PolyMAX for a system with close modes or heavy damping. Due to the phenomenon of modal interference induced by close modes or heavy damping, the effectiveness of system identification may be therefore degraded. According to the theory of mechanical vibration, the response data function can be expressed in rational fraction form through the curve fitting technique, and the modal identification can be implemented from parametric estimation from rational fractional coefficients. However, we cannot acquire the mode shape information because the conventional common denominator model only indicates the frequency response function of a single-degree-of-freedom system. In this paper, we propose the matrix-fractional coefficients model constructed by the frequency response functions of a multiple-degree-of-freedom system to perform modal estimation. In addition, to get rid of the phenomenon of omitted modes from the distortion from modal interference among the vibration modes of a system, we introduce a system model with higher-order matrix-fractional coefficients in the proposed method. The vibration modes of systems and fictitious modes caused by the numerical computation can be effectively separated through the different-order constructed stabilization diagram. Modal identification can be implemented by solving the eigenproblem of companion matrix yielded from least square estimation. Numerical simulation of a full model of sedan, confirms the validity and robustness of the proposed parametric-estimation method for a system with modal interference.

Keywords: PolyMAX, modal interference, stabilization diagram

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1. Introduction

Among many techniques of modal estimation in the past, the frequency-domain methods deal with the frequency response functions of a structure under consideration from which modal parameters are estimated. Due to the fact that the frequency response functions are readily available from input and output data, the frequency-domain methods have been used extensively [1]. In 2001, Auweraer et al. [2] proposed a fast-stabilizing parametric estimation method in frequency domain (LSCF) using the frequency response function matrix to employ the curve-fitting technique. In 2003, Guillanume et al. [3] introduced the concept of matrix fraction description (MFD) to extend LSCF for a poly-reference case and proposed so-called poly-reference least squares complex frequency-domain method (PolyMAX). The main advantages of the PolyMAX are its computation speed and yields very clear stabilization diagrams [4] even with highly noise-contaminated measurement data. However, it may yield poor estimates in damping ratios especially for a system with heavy damping and insufficient modes to be completely excited under noisy conditions. In addition, by using the stabilization diagram in conjunction with PolyMAX method, the accuracy of the identification results of structural modes is relatively consistent due to the sufficient order of the model to be estimated, the system and fictitious modes can therefore be effectively separated [5]. In recently years, the application of PolyMAX method for modal estimation has been extensively considered [6] and investigated [7]. This method has been employed to experimental modal estimation in flight testing [8], and can be effectively used to identify the damping ratios of the offshore wind power system [9] as well as estimate the modal parameters of power transformer winding [10] in electric power system. Also, the PolyMAX method has been extended to the damage detection and assessment of large-scale structures [11].

In this paper, we propose an improvement of PolyMAX method to perform parametric estimation of structures with modal interference. The content of modal interference caused by the close modes and high damping ratios often affects the accuracy of modal estimation. The serious problems of modal interference may lead to difficulties in modal estimation, especially for identifying damping ratios. By introducing the matrix-fractional coefficients model consisting of the frequency response functions of a multiple-degree-of-freedom (MDOF) system, and using the different-order constructed stabilization diagram in the process of modal estimation, we can estimate the number of structural modes to be identified. The modal parameters of a system can be obtained by directly solving the eigenproblem of companion matrix yielded from least square estimation.

2. Poly-reference least squares complex frequency-domain method

PolyMAX algorithm is based on frequency response function matrix of symmetric form as primary data containing the FRFs between all inputs and outputs. Through the least-squares estimation between frequency response function matrix and matrix rational mathematical model, the coefficients of numerator and denominator matrix polynomials can be identified. The modal parameters of a system can then be estimated from the coefficients of denominator matrix polynomials. The higher the constructed order mathematical model is, the more complete the modal information that can be obtained. The sensitivity of polynomial coefficients is, however, affected by high-order polynomial curve fitting.

Based on the property of companion matrix, modal identification can be implemented by directly solving the eigenproblem of companion matrix instead of solving the coefficients of numerator and denominator matrix polynomials in rational function form of frequency response function, and therefore significantly reduce relatively much calculations required in conventional PolyMAX method. After deriving the denominator coefficient, the poles (indicate the information of natural frequencies and damping ratios) and mode shape vectors of a system are directly related to the eigenvalues and eigenvectors of their companion matrix.

3. Estimation of identified Modes from the Phase of Frequency Response Function

Based on the theory of structural dynamics, the phase of the frequency of response function associated with a single-degree-of-freedom (SDOF) system will vary instantaneously from to when natural frequencies of a structure are equal to the applied loading frequency. We can therefore estimate the quantity of structural modes to be identified by roughly examining the phase of the frequency of response function [12]. However, for the most MDOF systems in practice, the number of structural modes to be identified may be erroneously determine due to the distortion the modal-identification information among the modes with relatively heavy damping and closely spaced modes. We cannot therefore relatively accurate determine the natural frequencies and quantity of structural modes by examining the phase of the frequency of response function due to the distortion the modal-identification information among the modes with relatively heavy damping and closely spaced modes. The more serious the problem of modal interference is, the more distortion the information of modal estimation has.

4. Numerical Simulations

The modal identification can be performed from both the excitation and response data of a structural system under external force excitation in dynamic tests. However, it is usually difficult to obtain the exact modal information in practical dynamic testing of large-scale structure. Consequently, it is necessary to verify in advance the effectiveness of the present method through the numerical simulations. To demonstrate the effectiveness of the present method for more complex structural systems, we consider a full vehicle model with two pairs of closely spaced modes (frequency separation smaller than 0.1 rad/sec) as shown in Fig. 1. A full model of motor vehicle can be generally viewed as a 7-dof system, which includes the bounce, pitch and roll motions for the body of motor vehicle, and 4 bounce motions for the wheels [13]. The vehicle model under consideration in this case is a 7-dof system with $\mathbf{u}=\left[\mathrm{u}_1,\mathrm{u}_2,\mathrm{u}_3,\mathrm{u}_4,\mathrm{u}_5,\mathrm{u}_6,\mathrm{u}_7\right]$, where $u_2=\varphi$ and $u_3=\theta$ are the rotational displacement of pitch and roll behavior of the motor vehicle, respectively, and others are the vertical displacement of bounce behavior of the motor vehicle and four wheels as shown in Fig. 1. The mass matrix is a diagonal matrix, diag $\mathbf{M}=\left[m_1,\mathrm{m}_2,\mathrm{m}_3,\mathrm{m}_4,\mathrm{m}_5,\mathrm{m}_6,\mathrm{m}_7\right]$, where the sprung mass m_1 represents the corresponding body mass of motor vehicle to the wheels as well as the unsprung mass, m_4, m_5, m_6, and m_7, represents the wheel and its associated components. $\mathrm{m}_2=\mathrm{I}_\mathrm{y}$ and $\mathrm{m}_3=\mathrm{I}_\mathrm{x}$ are, respectively, the pitch and roll moment of inertia of the motor vehicle. The stiffness matrix can be obtained as

$$\mathbf{K}=\begin{bmatrix} k_1+k_2+k_3+k_4 & -L_1k_1+L_2k_2-L_1k_3+L_2k_4 & -L_3k_1-L_3k_2+L_4k_3+L_4k_4 & -k_1 & -k_2 & -k_3 & -k_4 \\ -L_1k_1+L_2k_2-L_1k_3+L_2k_4 & L_1^2k_1+L_2^2k_2+L_1^2k_3+L_2^2k_4 & L_1L_3k_1-L_2L_3k_2-L_1L_4k_3+L_2L_4k_4 & L_1k_1 & -L_2k_2 & L_1k_3 & -L_2k_4 \\ -L_3k_1-L_3k_2+L_4k_3+L_4k_4 & L_1L_3k_1-L_2L_3k_2-L_1L_4k_3+L_2L_4k_4 & L_3^2k_1+L_3^2k_2+L_4^2k_3+L_4^2k_4 & L_3k_1 & L_3k_2 & -L_4k_3 & -L_4k_4 \\ -k_1 & L_1k_1 & L_3k_1 & k_1+k_{11} & 0 & 0 & 0 \\ -k_2 & -L_2k_2 & L_3k_2 & 0 & k_2+k_{12} & 0 & 0 \\ -k_3 & L_1k_3 & -L_4k_3 & 0 & 0 & k_3+k_{13} & 0 \\ -k_4 & -L_2k_4 & -L_4k_4 & 0 & 0 & 0 & k_4+k_{14} \end{bmatrix}$$

where L_1, L_2, L_3, and L_4 are, respectively, the half of axle track of front and rear wheel as well as the distances to the front and rear axle from the center of gravity of a motor. The summation of L_3 and L_4 is the wheelbase of a motor. $k_1\left(=k_2\right)$, $k_3\left(=k_4\right)$, $k_{11}\left(=k_{12}\right)$, and $k_{13}\left(=k_{14}\right)$ are the front and rear suspension spring stiffness as well as the front and rear tire stiffness, respectively. Throughout this numerical study, $\left[m_1,\mathrm{m}_4,\mathrm{m}_5,\mathrm{m}_6,\mathrm{m}_7\right]=\left[1365,46.8,46.8,41.4,41.4\right]kg$, $\mathrm{m}_2=\mathrm{I}_\mathrm{y}=1.831\times10^3\,kg\cdot m^2$ and $\mathrm{m}_3=\mathrm{I}_\mathrm{x}=4.98\times10^2\,kg\cdot m^2$; $k_1=k_2=2.2428\times10^4\,N/m$, $k_3=k_4=2.7022\times10^4\,N/m$, $k_{11}=k_{12}=2.32342\times10^5\,N/m$, and $k_{13}=k_{14}=2.92982\times10^5\,N/m$; $L_1=L_2=0.7165m$, $L_3=1.1135m$, and $L_4=1.5415m$; $\boldsymbol{C}=0.1\boldsymbol{M}+0.001\boldsymbol{K}\,N\cdot\sec/m$. Note that the system has proportional damping, because the damping matrix C can be expressed as a linear combination of M and K. The results of modal estimation are summarized in Table 1, which shows that the average errors in natural frequencies are less than 5 % and those in damping ratios are less than 10%. The identified mode shapes are also compared with the exact values in Fig.2, in which we observe good agreement with the minimum value of MAC [14] of 0.93. The first three mode shapes are modal behavior with bounce, pitch, and roll modes, respectively, of the global motor vehicle, while the last four mode shapes are modal behavior with bounce modes of the local left front, right front, left rear, and right rear wheels, respectively. It is relatively complicated for modal analysis of this model of motor vehicle, but it is applicable for accurate numerical simulation to confirm the validity of the proposed modal-estimation method.

Table 1 Results of the modal estimation of a 7-DOF system of a motor vehicle through the PolyMAX method choosing polynomials order $m = 2$

Mode	Natural Frequency (rad/sec)			Damping Ratio (%)			MAC
	Exact	PolyMAX	Error (%)	Exact	PolyMAX	Error (%)	
1	5.03	5.11	1.49	1.25	1.21	2.56	0.99
2	7.82	7.89	0.92	1.03	1.01	1.64	0.95
3	18.48	18.50	0.13	1.19	1.18	1.11	0.93
4	73.79	70.77	4.09	3.76	3.45	8.20	0.99
5	73.87	70.85	4.10	3.76	3.45	8.22	0.97
6	87.93	82.93	5.68	4.45	3.96	11.10	0.99
7	88.07	83.05	5.70	4.46	3.96	11.14	0.96

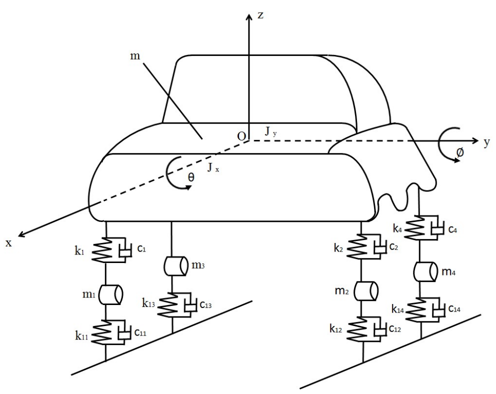
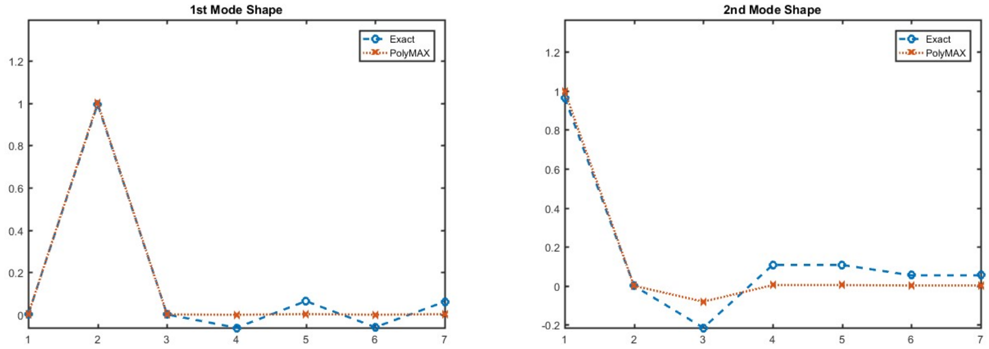


Fig.1 Schematic plot of the 7-dof system of a motor vehicle



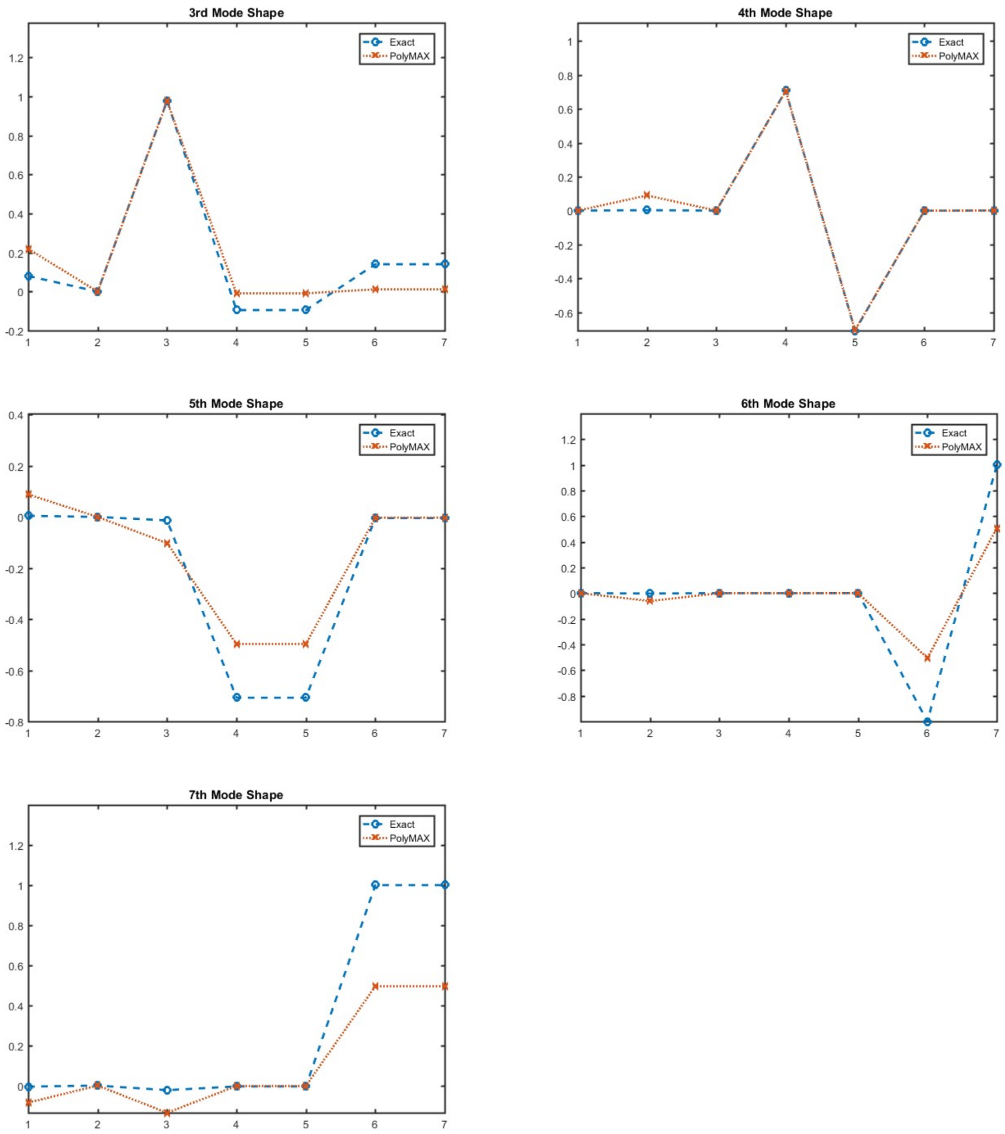


Fig. 2 Comparison between the identified mode shapes and the exact mode shapes of the 7-DOF system of a motor vehicle with two pairs of closely spaced modes

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