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Nonlinearity measurement for low-pressure encapsulated MEMS gyroscopes by transient response

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A B S T R A C T

To measure the nonlinear dynamic features of micromechanical gyroscopes, a non-parametric method based on Hilbert transform is proposed. Using a sequence of frequency stepping sinusoidal pulses as the excitation signal, a set of transient responses in the vicinity of the resonant frequency are obtained. The envelopes of the time-domain response signals are calculated by Hilbert transform. The location of the resonant frequency, as well as whether the gyroscope is working in linear or nonlinear region, can be approximately assessed from the waveform of the envelopes. In order to obtain the dynamic parameters of the gyroscope, a modified FREEVIB algorithm is designed for analyzing the free damped oscillation signals. The instantaneous amplitudes and instantaneous frequencies that extracted by Hilbert transform are further processed by singular spectrum analysis (SSA). Numerical simulation results indicate that the algorithm behaves better anti-noise performance and can be practically used for processing the experimentally sampled transient signals. Vibrating ring microgyroscopes are experimentally tested under different air pressure (10–100 Pa). From the largest response segment of the response sequences, qualification of the operation state, i.e. whether the gyroscope is working in the nonlinear region, is obtained from the envelope of the forced transient signal. Other parameters, including the Backbone, frequency response function (FRF) and Q-value curves, are calculated from the free damped oscillation signals. The results are in good agreement with those obtained by traditional frequency sweeping method.

1. Introduction

With the development of MEMS technique and technology, micromechanical gyroscopes that fabricated by bulk silicon process have been extensively used in industrial automation, medical devices, consumer electronics, e.g. navigation systems of automobiles, 3D input devices for computers, and camera stabilization. Their wide use can be attributed to the major advantages of compact size, light weight, low cost, and feasibility of integration with IC technologies etc [1–3]. To obtain an acceptable sensitivity and signal-to-noise ratio (SNR), MEMS gyroscopes are usually driven close to or even into nonlinear regimes. In other words, nonlinear effects are readily manifested, and some unanticipated and undesirable behaviors are presented in the dynamics response. Particularly, instability caused by nonlinear effects will decrease the overall performances of the gyroscopes [2–6]. Experimentally measuring the nonlinear dynamics of MEMS gyroscopes will not only give useful

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guidances to the operating circuit configuration for manufactures, but also make a reference for future design optimization. Furthermore, the obtained nonlinear dynamic parameters provide a valued support for parameter resonance and other cases that employing the nonlinear characteristics [3,7–9].

For a typical second order system, approaches of nonlinear dynamics measurement can be classified into two types, i.e. sinusoidal wave frequency sweeping method and free damped oscillation method. The former is most commonly used, but requirements of multiple frequency sweeping operations make the method practically tedious and time-consuming [10]. Moreover, the corresponding analysis results strongly rely on the physical model. On the other hand, the free damped oscillation method is non-parametric operated and independent of the physical model [11,12]. Within the relevant studies on free damped oscillation method, FREEVIB, firstly introduced by Feldman M, is the most well-known algorithm [12–15]. Instantaneous frequency (IF) and instantaneous amplitude (IA), are extracted from the free damped oscillation signal by Hilbert transform. Then Backbone curve, damping curve, and other nonlinear characteristics of the system are obtained from the calculated IF and IA. The main advantages of FREEVIB lay in extensive applicability and no need for prior knowledge. However, since Hilbert transform is very sensitive to noise, most reported literature are mainly focused on theoretical analyses and numerical simulations [12,13]. Peng et al. extracted the nonlinear characteristics from free damped oscillation signals with Polynomial Chirplet Transform instead of Hilbert transform. The simulation results show a preferable anti-noise performance [16,17]. Together with the SSA (singular spectrum analysis) algorithm [18,19], the noisy fluctuations in the IFs and IAs that directly calculated by Hilbert transform are suppressed in our previous work [20–22]. The numerical simulation and experiment results reveal that the modified algorithm behaves better anti-noise performance than FREEVIB. However, only the free damped oscillation signals are utilized for calculating the Backbone and frequency response function (FRF) curves. The response signals of the forced vibration are not considered. Additionally, only a high-Q gyroscope is experimentally tested and the estimation of the damping curves for relatively low-Q gyroscopes is not given.

In practical dynamic measurement of a specific gyroscope, because of the non-idealities in micro-machining process and the amplitude-dependent effect resulting from nonlinear stiffness [4], the actual resonant frequency cannot be accurately estimated in advance. Therefore, the frequency span of the stepped frequency sweeping excitation method reported in our previous work [20–22] usually needs to be experimentally adjusted several times before satisfied data can be obtained. Meanwhile, the amplitude of the excitation signal also needs to be carefully adjusted to avoid unnecessary large vibrations. Since the sampled response signals are processed offline in computers, it generally takes a lot of time to complete one measurement process. Therefore, in order to improve the working efficiency, a simple criterion for determining whether the response signal meets the test requirements is needed. Another problem appears in gyroscope chip selection stage. Before vacuum encapsulation, gyroscope chips with good quality are picked out according to the measured Q values under low air pressure conditions. Therefore, the Q values are also needed to be calculated from the response signals. Furthermore, in order to comprehensively analyze dynamic responses of a specific gyroscope, the nonlinearity of the damping ratio should also be taken into consideration. For the purpose of this paper, the algorithms of our previous work [20–22] is further analyzed and improved to fulfill such experimental requirements. A simple criterion for quickly determining whether the excitation amplitude is high enough for nonlinearity measurement is obtained. With the aid of such a criterion, the nonlinear performance can be visually observed without complex offline computation. Algorithms for calculating the Backbone, FRF, damping coefficient and Q-value curves from the damped oscillation signals are given in this paper. As the response signal may be corrupted by various kinds of noises in practice, the anti-noise performance of the proposed algorithm is further evaluated by various types of noises. Nonlinearities of the vibrating ring microgyroscopes are experimentally tested under different working pressures (10–100 Pa). Satisfied results indicate that the proposed method is practically available for evaluating dynamic characteristics of the encapsulated MEMS gyroscopes.

2. Working principles and signal processing algorithms

For a single degree-of-freedom (SDOF) second-order system, the dynamic equation under the sinusoidal excitation is given by,

$$m\ddot{x} + c\dot{x} + k(x)x = F(t), F(t) = \begin{cases} B\sin(2\pi f_d t), 0 \leqslant t \leqslant t_1; \\ 0, t > t_1; \end{cases} \tag{1}$$

where m, c, k(x) are mass, damping coefficient and stiffness coefficient respectively. B, f_d denote the amplitude and frequency of the sinusoidal excitation signal F(t) respectively. t_1 is the duration time of excitation. For $t \geq 0$, the sinusoidal pulses drive the system into the forced vibration state. When $t > t_1$, the excitation is stopped and the system comes into the free damped oscillation state.

A Duffing system with high frequency (25 kHz) and low damping (Q ≈ 10,000) is taken as an example. Where m is 10^{-7} kg, and c is 1.571×10^{-6} N/(m/s) that independent of the vibration amplitude, k(x) is vibration amplitude-dependent and equals to $2.467 \times 10^3 \times (1 - 0.002x^2)$ N/m to gain a "softened-spring" nonlinearity, t_1 is 1.5 s. The simulation time and the sampling frequency *fs* are set to be 2.5 s and 1 MHz respectively.

When the amplitude B is relatively small, the Duffing system is forced to vibrate in the approximately linear region and the response amplitude is therefore relatively small. With the aid of Matlab solver, the system responses are numerically calculated as Fig. 1, where the excitation amplitude B = 0.01 N, the excitation frequency f_d is 25 kHz (resonant frequency)