



Sensitivity analysis of the Nonlinear Integral Positive Position Feedback and Integral Resonant controllers on vibration suppression of nonlinear oscillatory systems



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ABSTRACT

A new Nonlinear Integral Positive Position Feedback (NIPPF) approach and the Integral Resonant Control (IRC) method are implemented and analyzed for nonlinear vibration control. The performances of both controllers are compared to the Positive Position Feedback (PPF) method. A comprehensive form of the nonlinear feedback controller that includes all three approaches is considered for the closed-loop system; this form is then solved innovatively using the Method of Multiple Scales. The particular form of the modulation equations for each controller is obtained via the calculated comprehensive solution. Sensitivity analysis is performed on the controller parameters of each approach, and the influence of the parameters on the system response is studied. Results demonstrate that controller structure has the most salient role in the suppression performance. Even by selecting the optimal control parameters for each approach, improvements in the control performances are limited. The NIPPF controller delivers superior results compared to the other two approaches in the frequency and time domains. Using this approach, not only is the vibration amplitude at the exact resonant frequency sufficiently suppressed, but the subsequent peaks in the frequency domain are also reduced significantly compared to the PPF method.

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1. Introduction

High-amplitude resonant vibrations that occur in various structures and machines are always undesirable. Mechanical malfunctions or structural failures are unpleasant results of these vibrations. Resonant vibrations, however, do not always occur in a linear manner [1]. An example is nonlinear vibrations in the Atomic Force Microscopy and micro-cantilever beams [2]. In micro-scale motions, nonlinearity of systems requires more consideration as more accurate measurements [3] or robust controlled motions are needed [4].

Geometric nonlinearities (due to large displacements, strains, or rotations), nonlinear properties of materials, and nonlinear external forces are the main causes of nonlinear vibrations. These nonlinearities are modeled and represented by nonlinear terms in the general equation of systems. Hence, the nonlinear vibrating system cannot be solved and controlled by regular linear approaches. Two examples of applied methods are the open-loop and the adaptive nonlinear vibration control [5,6]. In addition, Nonlinear Saturation Control (NSC) [7,8] is studied for nonlinear suppression in cantilever beams and more complicated structures. Cubic velocity feedback [9] and nonlinear implementation of Positive Position Feedback (PPF)

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[7,10,11] are two other commonly used approaches. According to the results presented in [10,11], the PPF controller can successfully reduce the vibration amplitude at resonant frequency. However, high-level suppression in the primary mode using the PPF approach results in shifting the peak amplitude to left, right, or both sides of the resonant frequency in the closed-loop system. In some cases, the amplitude of the shifted peak is the same as the amplitude of the resonant mode. This implies that the vibration controller has failed to reduce the vibration amplitude in the working frequency domain. Delayed-feedback has been combined with NSC in one study [12], and the cubic velocity feedback [13] in another study. The Receptance method [14] and the Neural Network approach [15,16] have also been used for nonlinear active control of vibration.

The PPF controller was first presented and implemented for linear systems [17]. This approach was later studied in more detail and optimized in other works such as in [18,19]. The problem with the PPF approach is that the closed-loop system has low damping at non-resonant excitations. To overcome this drawback, a first-order compensator was combined with the second-order compensator of the PPF for implementation of either position or acceleration feedbacks in the modified version [20,21]. In terms of increasing the damping of the closed-loop system, the Integral Resonant Control (IRC) has also been implemented for linear systems [22].

In this study, PPF, IRC and Nonlinear Integral Positive Position Feedback (NIPPF) approaches are implemented for nonlinear oscillation attenuation. The PPF controller has previously been used in other studies [11]; however, its solution is utilized here as a reference to assess the performance of the IRC and the NIPPF approaches. The NIPPF consists of a nonlinear second-order positive feedback compensator, which is set parallel with a simple positive integrator. Nonlinearity in the NIPPF controller structure originates from a cubic nonlinear position term in the structure of the second-order compensator. This term increases the potential design flexibility to obtain a response that is not achievable by a linear controller. After explaining the controllers, a comprehensive approximate solution for the closed-loop system is obtained by utilizing the Method of Multiple Scales. Having the solution for the three controllers, the results are graphically demonstrated and discussed. In order to understand the performance of the controllers in more detail, sensitivity analysis on the closed-loop system responses is performed, and the influence of each parameter on the control output is plotted and discussed.

2. Nonlinear system model and controller design

In this section, first the dynamics of the nonlinear system is presented, and then the general form of the controller solution for the NIPPF controller is introduced. The model for the nonlinear dynamics of a cantilever beam is governed by a nonlinear partial differential equation. The primary resonant mode of the cantilever beam is considered not to be involved in an internal resonance with other modes of the system. Single-mode discretization approach results in nonlinear differential equation of [23]:

$$\ddot{x}(t) + \eta_s \dot{x}(t) + \omega_s^2 x(t) + \alpha x^3(t) + \beta x(t) \dot{x}^2(t) + \gamma x(t)^2 \dot{x}(t) - f \cos(\Omega t) = F_c(t), \quad (1)$$

where $x(t)$ is the variable of the main system, *over-dot* is the differentiation with respect to time, and $F_c(t)$ is the control input. f is the amplitude and Ω is the frequency of the external excitation. $\eta_s = 2\mu_s\omega_s$, where μ_s and ω_s are the damping ratio and resonant frequency of the system, respectively. α is the curvature nonlinearity coefficient, β and γ are the inertia nonlinearity coefficients.

The mathematical model for PPF, IRC, and NIPPF are described here to be used in the following sections. First, the model for the PPF controller is:

$$\ddot{y}(t) + \eta_p \dot{y}(t) + \omega_p^2 y(t) = \lambda_p x(t), \quad (2)$$

where $y(t)$ is the state-variable of the PPF controller, $\eta_p = 2\mu_p\omega_p$, where μ_p is the damping ratio, and ω_p is the frequency of the PPF compensator. $\lambda_p > 0$ is the controller gain, and feedback loop is closed by setting $F_c(t) = \kappa_p y(t)$ in Eq. (1), for $\kappa_p > 0$. For the IRC, the lossy integrator is expressed as:

$$\dot{z}(t) + \omega_l z(t) = \lambda_z x(t), \quad (3)$$

where $z(t)$ is the IRC controller's variable, ω_l is the lossy integrator's frequency and $\lambda_z > 0$ is the IRC gain. Based on the definition, the control law for the IRC is $F_c(t) = \kappa_z z(t)$, where $\kappa_z > 0$.

The feedback format of the NIPPF controller is designed such that it absorbs some of the vibration energy by increasing the damping of the system, and compensates the resonant energy using the positive feedback concept. To achieve this goal, a second-order resonant compensator is used alongside an integrator in the feedback loop. Since the nonlinearities of the system shown in Eq. (1) are cubic, a cubic position term is added to the second-order subsection to compensate for nonlinear terms. The second-order compensator does the most parts of the suppression at exact resonant frequency, while the integrator increases the overall damping of the system. The NIPPF controller is described as:

$$\begin{cases} \ddot{u}(t) + \eta_N \dot{u}(t) + \omega_N^2 u(t) + \delta u^3(t) = \lambda_u x(t), \\ \dot{v}(t) = \lambda_v x(t), \end{cases} \quad (4)$$

with the control law of:

$$F_c(t) = \kappa_u u(t) + \kappa_v v(t). \quad (5)$$

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