



Contents lists available at ScienceDirect

Mechanical Systems and Signal Processing

journal homepage: www.elsevier.com/locate/ymssp

Narrowband feedback for narrowband control of resonant and non-resonant vibration

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ARTICLE INFO

Article history:

Received 15 May 2015

Received in revised form

26 January 2016

Accepted 30 January 2016

Keywords:

Vibration control

Narrowband feedback

Narrowband control

Second order filters

Loudspeakers

ABSTRACT

This paper presents a simple feedback methodology that uses second order filters to control narrowband resonant and non-resonant vibration of a structural system. In particular, a single degree-of-freedom system is studied throughout the paper. The idea of the methodology is based on the fact that direct feedback is effective for in-phase vibration control. Thus, the position, velocity and acceleration are respectively fed back to control the low, resonant and high frequency vibration of the system. Each of these is passed through a band pass filter of second order that is inserted to extract and feed back the in-phase signal component only. This is called narrowband feedback. It is demonstrated with experiments that narrowband feedback is useful for narrowband control of resonant and non-resonant vibration.

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1. Introduction

Active control in the context of feedback control has generally meant *active damping control* that is aiming to reduce resonant responses of sound and vibration systems. This conception appears reasonable when the systems are all excited by an ideal broadband source (e.g., white noise and an impulse) and are further lightly damped. However, not all responses and systems ought to be controlled are resonant responses nor are necessarily lightly damped. For example, low frequency vibration is persistent in precision machines even if anti-vibration mounts are used [1]. The sound radiation to an acoustic free field is a phenomenon mostly in the high frequency region above the fundamental natural frequency of a moving-coil loudspeaker [2]. Vibration of a rotating or reciprocating machine is often dominated by harmonics of the operating frequency, which can occur at any frequencies including resonance and non-resonance frequencies [3]. Acoustic fields inside car cabins are indeed very highly damped [4]. Active control of such non-resonant sound and vibration in a highly damped system has been rarely reported in the literature. The paper presented here studies this, using a simple single degree-of-freedom (SDOF) vibration system.

There are many feedback methods that have been successful for resonant vibration control. Karnopp et al. [5] proposed direct velocity feedback in 1970s that feeds a velocity response back to the collocated force actuator through gain. They demonstrated with a SDOF system that the controller acts as an electrical damper. Balas [6] suggested that this could also be applied to multiple modal control of a flexible structure as long as the pair of sensor and actuator were truly collocated. Researchers soon realized that Balas's application tended to go unstable at high frequencies and thus instead proposed positive position feedback (PPF) in 1990s [7]. Unlike direct feedback (gain control), PPF is narrowband feedback that uses a low pass filter of second order as the controller so that it works effectively in the tuned frequency region while ineffectively at high frequencies. PPF is particularly suitable for

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controlling the fundamental mode when a strain-type sensor is used. The electrical dynamic absorber (EDA) method was then proposed as a rather general tool for resonant vibration control, which is also narrowband feedback employing a number of second order filters [8–11]. Unlike PPF, however, it is a passivity-based controller (PBC) electrically realizing a mechanical dynamic absorber [9,10]. It is further a robust-PBC that is even more robust than Karnopp's electrical damper [8]. This method is also applicable to multiple modal control in both collocated and non-collocated control configurations, regardless of the types of transducers used [10,11]. Many other methods are also available in the control society, such as, classical compensators [12], state-based optimal methods [13], and intelligent methods like fuzzy control [14]. As such, active damping control for resonant vibration is a topic that has been extensively studied.

In this paper, a simple feedback methodology is presented for narrowband control of not only resonant but non-resonant vibration of a SDOF system. The idea is commonly based on the fact that direct feedback is effective for in-phase vibration control. Thus, the position, velocity and acceleration are respectively fed back to control the low, resonant and high frequency vibration of the system. Each of these is passed through a controller consisting of a band pass filter of second order. These are called narrowband position, velocity and acceleration feedback, respectively. It is well known that narrowband velocity feedback is a way of realizing an EDA for resonant vibration control [8]. The main focus of this paper is thus to investigate narrowband position and acceleration feedback for non-resonant vibration control. It is demonstrated with experiments that these two methods are respectively related to *active stiffness and inertia control*. They are thus useful for controlling the vibration at frequencies well below and well above the resonance frequency, respectively.

2. Theory of systems

Consider the active feedback control of a SDOF vibration system shown in Fig. 1(a), consisting of mass m_s , spring k_s and damper c_s . The SDOF system may represent a simple model of a resilient structure or a single vibration mode of a flexible structure. It is excited by the primary force $f(t)$ and controlled by the force $f_c(t)$ via the negative feedback controller $-C(j\omega)$, where t is time, ω is the angular frequency, and $j = \sqrt{-1}$. The systems before and after control can be represented by the two block diagrams shown in Fig. 1(b), where $P(j\omega)$ indicates the plant, $d(t)$ the disturbance signal, and $e(t)$ the error signal. The two signals $d(t)$ and $e(t)$ are physically the system responses (position, velocity or acceleration) measured by a common vibration sensor before and after control (i.e., disconnecting and connecting the feedback loop), respectively. Thus they cannot be measured simultaneously. Throughout this paper, the time dependence of signals is explicitly indicated (e.g., $e(t)$) while their frequency dependence is abbreviated (e.g., e) for simplicity.

2.1. Direct feedback

The dynamic equation of the SDOF system in Fig. 1(a) can be written as

$$m_s \ddot{w}(t) + c_s \dot{w}(t) + k_s w(t) = f(t) + f_c(t), \quad (1)$$

where $w(t)$, $\dot{w}(t)$ and $\ddot{w}(t)$ are the position, velocity and acceleration of the system in time, respectively. Let the control force be

$$f_c(t) = -[kw(t) + c\dot{w}(t) + m\ddot{w}(t)], \quad (2)$$

where k , c and m are non-negative and are the gains for direct position, velocity and acceleration feedback control, respectively. Combining Eqs. (1) and (2) gives the controlled system written as

$$[m_s + m]\ddot{w}(t) + [c_s + c]\dot{w}(t) + [k_s + k]w(t) = f(t). \quad (3)$$

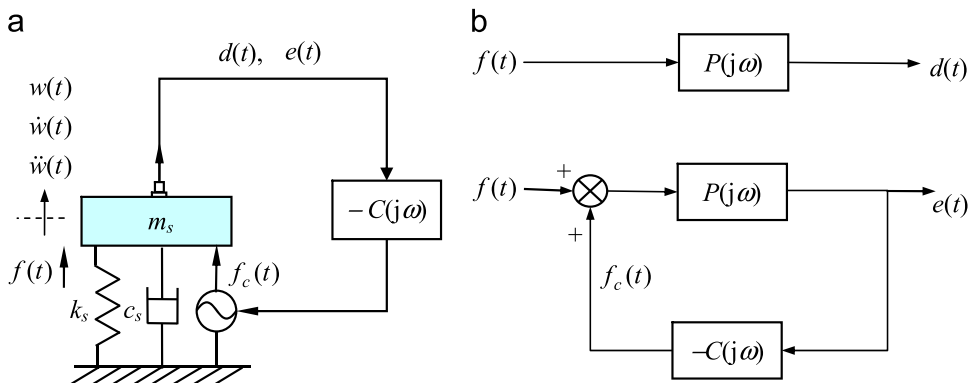


Fig. 1. Active vibration control of a SDOF system $P(j\omega)$ by the control force $f_c(t)$ with the negative feedback controller $-C(j\omega)$: (a) Schematic drawing and (b) the control block diagrams. The responses before and after control are $d(t)$ and $e(t)$, respectively.

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