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Acceleration feedback control of human-induced floor vibrations

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ABSTRACT

Active vibration control (AVC) via a proof-mass actuator is considered to be a suitable technique for the mitigation of vibrations caused by human motions in floor structures. It has been observed that actuator dynamics strongly influence structure dynamics despite considering collocated actuator/sensor control. The well-known property of the interlacing of poles and zeros of a collocated control system is no longer accomplished. Therefore, velocity-based feedback control, which has been previously used by other researchers, might not be a good solution. This work presents a design process for a control scheme based on acceleration feedback control with a phase-lag compensator, which will generally be different from an integrator circuit. This first-order compensator is applied to the output (acceleration) in such a way that the relative stability and potential damping to be introduced are significantly increased accounting for the interaction between floor and actuator dynamics. Additionally, a high-pass filter designed to avoid stroke saturation is applied to the control signal. The AVC system designed according to this procedure has been assessed in simulation and successfully implemented in an in-service open-plan office floor. The actual vibration reductions achieved have been approximately 60% for walking tests and over 90% for a whole-day vibration monitoring.

1. Introduction

Advances in structural technologies have enabled the design of light and slender structures, which have increased susceptibility to vibration. This is compounded by the trend toward open-plan floor structures, which have less inherent damping. Examples of notable vibrations under human-induced excitations have been reported in office buildings, footbridges, shopping malls and sport stadia, amongst other structures. Such vibrations can cause a serviceability problem in terms of disturbing the users, but they rarely affect the fatigue behaviour or safety of structures [1].

Passive and semi-active devices have been proposed to reduce floor vibrations [2,3]. However, due to their passive nature, the vibration cancellation is often of limited effectiveness and they often have to be tuned to damp a single vibration mode. In many cases, several of these devices have to be used to achieve the required vibration reduction. Instead, an active control approach rather than passive devices might be more effective [4]. A state-of-the-art review of technologies (passive, semi-active and active) for mitigation of human-induced vibration can be found in [5]. Furthermore, techniques to cancel floor vibrations (especially passive and semi-active techniques) are reviewed by Ebrahimpour and Sack [6].

An AVC system based on direct velocity feedback control (DVFC) with saturation has been studied analytically and implemented experimentally for the control of floor vibrations induced by humans via a proof-mass actuator [7,8]. This actuator generates inertial forces in the structure without need for a fixed reference. The velocity output, which is obtained by an integrator circuit applied to the measured acceleration response, is multiplied by a constant gain and feeds back to a collocated force actuator. The merits of this method are its robustness to spillover effects due to high-order unmodelled dynamics and that it is unconditionally stable in the absence of actuator and sensor (integrator circuit) dynamics [9]. That is, the resulting root locus map exhibits the well-known interlacing property of poles and zeros of collocated systems [10]. However, when these dynamics are considered, the interlacing property is no longer accomplished. Then, DVFC is not such a desirable solution. Furthermore, the control law is completed by a command limiter (i.e., a saturation nonlinearity in the command signal) that is introduced to avoid actuator force and stroke saturation and to level off the system response in the case of unstable behaviour.

It has been shown that the use of a proof-mass actuator, even though this is positioned at the same location as the sensor, leads to a non-collocated root locus map. The actuator dynamics introduce a pair of high-damped poles that affect drastically

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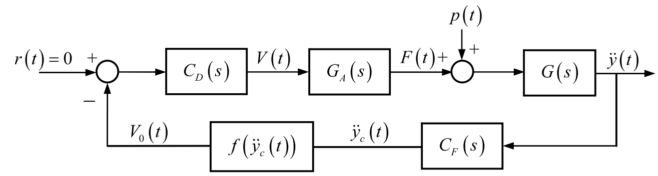
the loci corresponding to the structure dynamics [7,10,11]. This fact might reduce importantly the stability margins and the possible damping to be introduced by the AVC system. Moreover, the closed-loop system could be very sensitive to parameter uncertainties since the control gain should be carefully chosen. For example, in [7], it was shown that a couple of branches in the root locus corresponding to the actuator dynamics go to the right-half plane provoking unstable behaviour in the actuator. The saturation avoids this unstable behaviour but the actuator is involved in a stable limit cycle [12], which is not desirable since it could result in dramatic effects on the system performance and its components. Generally, depending on the interaction between floor and actuator dynamics, one of them will tend to be unstable. Thus, the selection of a suitable compensator to be applied to the actual measured output that ensures high stability margins and enables potentially the introduction of significant damping via closed-loop control is an interesting issue to be dealt with.

This paper presents a design process of a compensator to be applied to the acceleration output of a structure. It is assumed that the output of the structure is the acceleration, which is usually the actual magnitude measured. This compensator accounts for the interaction between the structure and the actuator dynamics in such a way that it introduces the phase-lag needed to achieve a closed-loop system with desirable properties. Such properties are high damping for the fundamental vibration mode of the structure and high stability margins. Both properties lead to a closed-loop system robust with respect to stability and performance [10]. Acceleration feedback with the phase-lag compensator will be referred as to compensated acceleration feedback control (CAFC) throughout the paper. The proposed design process is completed by: (1) a phase-lead (high-pass property element) compensator which prevents the actuator stroke saturation at low frequencies, and (2) a saturation nonlinearity applied to the control signal to avoid actuator force overloading at any frequency. This phase-lead compensator (direct compensator from this point onwards) must be designed before the design of the phase-lag compensator (feedback compensator from this point onwards) in order to account for the dynamics introduced by the former in the design of the latter. Additionally, the design process is simple since the direct compensator is derived from a frequency domain analysis and the feedback compensator is obtained using the root locus method.

The remainder of this paper is organized as follows. The general control strategy together with floor and actuator dynamics are briefly described in Section 2. The control design procedure is presented in Section 3. Section 4 deals with the experimental implementation of the AVC system in an in-service open-plan office floor. This section contains the system dynamic models, the application of the proposed design procedure, simulation results to assess the feasibility of the design and experimental results involving walking tests and whole-day monitoring tests to quantify the actual vibration reductions. Finally, some conclusions and suggestions for future work are given in Section 5.

2. Control strategy and system dynamics

The main components of the general control strategy adopted in this work are shown in Fig. 1. The output of the system is the structural acceleration since this is usually the most convenient quantity to measure. Because it is rarely possible to measure the system state and due to simplicity reasons, direct output measurement feedback control might be preferable rather than state-space feedback in practical problems [13]. In this figure, G_A is the transfer function of the actuator, G is of the floor structure, C_D is of the direct compensator and C_F is of the feedback compensator. The direct one is merely a phase-lead compensator (high-pass property) designed to avoid actuator stroke saturation for low-frequency components. It is notable that its influence in the global stability will be small since only a local phase-lead is introduced. The feedback one is a phase-lag compensator designed to increase the closed-loop system stability and to make the system more amenable to the introduction of significant damping by a closed-loop control. The control law is completed by a nonlinear element $f(\ddot{y}_c)$ that may be a saturation nonlinearity to account for actuator force overloading [4], an on–off nonlinearity with a dead zone [12] or a variable gain with a switching-off function [14]. In this work, a saturation nonlinearity will be assumed.



$r(t)$	Reference command	$\ddot{y}(t)$:	Acceleration response
$V(t)$:	Control voltage	$\ddot{y}_c(t)$:	Compensated acceleration
$F(t)$:	Actuator force	$V_0(t)$:	Initial control voltage
$p(t)$:	Plant disturbance	$f(\ddot{y}_c)$:	Nonlinear element

$C_D(s)$: Transfer function of the direct compensator
$G_A(s)$: Transfer function of the proof-mass actuator
$G(s)$: Transfer function of the floor structure
$C_F(s)$: Transfer function of the feedback compensator

Fig. 1. General control scheme.

2.1. Floor dynamics

If one considers the collocated case between the acceleration (output) and the force (input) and using the modal analysis approach, the transfer function of the floor dynamics can be represented as an infinitive sum of second-order systems as follows [10]

$$G(s) = \sum_{i=1}^{\infty} \frac{\chi_i s^2}{s^2 + 2\xi_i \omega_i s + \omega_i^2}, \tag{1}$$

where $s = j\omega$, ω is the frequency, χ_i, ξ_i and ω_i are the inverse of the modal mass, damping ratio and natural frequency associated to the ith mode, respectively. For practical application, N vibration modes are considered in the frequency bandwidth of interest. The transfer function $G(s)$ is thus approximated by a truncated one

$$\tilde{G}(s) = \sum_{i=1}^{N} \frac{\chi_i s^2}{s^2 + 2\xi_i \omega_i s + \omega_i^2}. \tag{2}$$

2.2. Proof-mass actuator dynamics

The linear behaviour of a proof-mass actuator can be closely described as a linear third-order model. Unlike previous works [4,12], a low-pass element is added to a linear second-order system in order to account for the low-pass property exhibited by these actuators. The cut-off frequency of this element is not always out of the frequency bandwidth of interest since it is approximately 10 Hz [15]. Such a low-pass behaviour might affect importantly the global stability of the AVC system. Thus, the actuator is proposed to be modelled by

$$\begin{aligned} G_A(s) &= \left(\frac{K_A s^2}{s^2 + 2\xi_A \omega_A s + \omega_A^2}\right)\left(\frac{1}{s+\varepsilon}\right) \\ &= \frac{K_A s^2}{s^3 + (2\xi_A \omega_A + \varepsilon)\, s^2 + \left(2\xi_A \omega_A \varepsilon + \omega_A^2\right) s + \varepsilon \omega_A^2}, \end{aligned} \tag{3}$$