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Vibration Control Strategies for Cancelling Floor Vibrations via Inertial Mass Dampers

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Abstract

Nowadays it is quite common to find vibration serviceability problems in light and slender structures due to the developments of building materials and construction techniques. These structures satisfy ultimate limit state criteria but have the potential of attracting complaints coming from excessive human-induced vibrations. These problems can be found in pedestrian bridges, open-plan layout floors, cantilevered grandstands, etc. Among the investigations carried out, active vibration control (AVC) via inertial mass actuators (or active mass dampers, AMD) and passive vibration control (PVC) via tuned mass dampers (TMD) are the most common and efficient solutions.

The main contributions of this thesis are the following. The first one is to propose a practical design guide for AVC using a single-input single-output (SISO) control strategy, which considers the most important practical issues when inertial mass actuators are used, like low-frequency dynamics, stroke saturation, force saturation and spillover problems. A non-dimensional system (actuator and structure) is considered in order to generalize the results and extract simple and effective design rules. In addition, experimental results are presented in order to validate this study.

The second one is to propose a common/general framework/methodology for designing AVC and PVC to control the aforementioned human-induced vibrations. This framework uses a feedback loop for the design of control parameters and accounts for the nature of human excitation and how humans feel the vibration. For an AVC system, the well-known direct velocity feedback strategy is used (for both the design and application); however, for the PVC, the feedback loop is only for designing purposes. The methodology developed considers not only SISO control systems, but also multi-input multi-output (MIMO) control systems, especially useful to cancel vibrations coming from coupled vibration modes with closely-spaced natural frequencies. The strategy finds simultaneously the optimal placements of the TMDs/AMDs and the system parameters (control gain matrix for AVC; damping ratios and tuning frequencies for TMDs of PVC). To make this strategy implementable and suitable to cancel human-induced vibrations, elements such as input-output frequency weighting, output time weighting, band-pass filters, actuator dynamics and non-linearities are considered. Simulations account for all

these elements, providing thus reliable results.

This work also presents an approach to alleviate the computational burden problem when an optimal MIMO AVC is designed. This computational problem is especially important when complex floor structures have to be dealt with, such as floors with a large number of vibration modes (global and local) and with a large number of test points. To tackle this problem, a novel meta-heuristic algorithm is used, which is denoted as Coral Reefs Optimization with Substrate Layer, as a competitive co-evolution algorithm with different exploration procedures within a single population of solutions. This promotes a powerful evolutionary-like algorithm for optimization problems, which has been shown to be very effective for this particular problem of MIMO AVC design.

Finally, this work includes models of structures in order to implement the aforementioned contributions. Thus, finite element models of floors are derived and calibrated through several operational modal analyses and experimental modal analyses. This is demonstrated by application examples: an in-service dining hall floor and a full-scale laboratory floor structure built during this thesis.

Resumen

Hoy en día es bastante habitual encontrar estructuras ligeras y esbeltas que presentan problemas relacionales con el estado límite de servicio de vibraciones. Esto es debido fundamentalmente al uso de nuevos materiales de construcción y al empleo de nuevas técnicas constructivas. Estas estructuras cumplen los estados límites últimos pero no es raro que los usuarios se quejen debido al elevado índice de vibración que suelen presentar. Estos problemas suelen aparecer en pasarelas peatonales, forjados de grandes luces, gradas en voladizo, etc. Entre las estrategias de cancelación más habituales se encuentran el uso de amortiguadores de masa inercial activos (conocidos como AMD por sus siglas en inglés) y amortiguadores de masa inercial pasivos, habitualmente conocidos como TMDs (tuned mass dampers).

Las principales contribuciones de esta tesis son las que se mencionan a continuación. La primera es proponer una guía de diseño para sistemas de control activo basado en estrategias SISO (single-input single-output), una entrada y una salida. En esta guía se contemplan aspectos prácticos que son fundamentales, tales como la dinámica de los actuadores, saturaciones y los típicos problemas de inestabilidad debidos a las altas frecuencias. Se ha realizado un estudio adimensional a partir del cual se han obtenido reglas de diseño sencillas que aseguran la implementabilidad del control. Además, se han obtenido resultados experimentales que validan dichas reglas.

El segundo objetivo es proponer una metodología común/general de diseño tanto de estrategias de control activo como pasivo, basadas ambas en amortiguadores inerciales. Se emplea una realimentación en lazo cerrado para diseñar los parámetros de control teniendo en cuenta tanto la naturaleza de la excitación como la percepción de la vibración por los humanos. Para la estrategia de control activo se emplea una realimentación directa de la velocidad, en este caso tanto para el diseño como para la aplicación, sin embargo, en el caso del sistema pasivo la realimentación es sólo para el diseño. La metodología desarrollada considera sistemas de control SISO y MIMO (multi-input multi-output), siendo estos últimos, especialmente útiles para cancelar vibraciones debidas a varios modos de vibración con frecuencias similares. Con la metodología propuesta se puede diseñar simultáneamente tanto la localización óptima de los TMDs/AMDs como los parámetros de control (matriz de ganancias para el sistema activo;

y amortiguamientos y frecuencias de sintonizado para los pasivos). Esta metodología se ha particularizado en la cancelación de vibraciones inducidas por humanos. Así, ciertos factores se han considerado: la ponderación en frecuencias tanto de la excitación como de la respuesta considerando la naturaleza de las vibraciones inducidas por humanos, ventanas temporales de salida que aseguran la estabilidad de la solución adoptada, filtros paso-bajo y dinámicas y no linealidades del actuador. Las simulaciones que involucran todos estos elementos sin duda son fiables y llevan a diseños que funcionan en la práctica.

En este trabajo también se presenta una solución al problema de la carga computacional involucrada en el proceso de optimización, fundamentalmente cuando se trata de sistemas MIMO aplicados a forjados complejos. Para abordar este problema, se ha utilizado un algoritmo meta-heurístico, llamado “Coral Reefs Optimization with Substrate Layer”, definido como un algoritmo competitivo de co-evolución con diferentes procedimientos de exploración dentro de una única población de soluciones. De esta forma, se emplea un potente algoritmo evolutivo para el problema de optimización, que ha resultado ser muy eficaz.

Finalmente, en el trabajo se incluyen modelos de estructuras con el fin de implementar las contribuciones anteriormente mencionadas. Se han realizado modelos de elementos finitos de forjados que han sido calibrados mediante varios ensayos experimentales, análisis modales operacionales y análisis modales experimentales. Todo ello ha sido demostrado mediante ejemplos de aplicación en un forjado de un comedor y en un forjado de laboratorio a escala real construido durante esta tesis.

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Acronyms

AMD Active Mass Damper

AVC Active Vibration Control

bpm beats per minute

CBO Colliding Bodies Optimization

CL Closed-Loop, Comfort Class

cRIO CompactRIO

CRO Coral Reefs Optimization

CRO-SL Coral Reefs Optimization with Substrate Layer

DE Differential Evolution

DLF Dynamic Loading Factor

DVF Direct Velocity Feedback

DVFC Direct Velocity Feedback Control

ECBO Enhanced Colliding Bodies Optimization

EMA Experimental Modal Analysis

FE Finite Element

FRF	Frequency Response Function
FTT	Feed-Through Term
GM	Gaussian Mutation
HIL	Human-Induced Loading
HMCR	Harmony Memory Considering Rate
HS	Harmony Search
IRC	Integral Resonant Control
LTI	Linear Time Invariant
MAC	Modal Assurance Criterion
MIMO	Multi-Input Multi-Output
MP_x	Multi-Points crossover
MTMD	Multiple Tuned Mass Damper
MTVV	Maximum Transient Vibration Value
NI	National Instruments
OL	Open-Loop
OMA	Operational Modal Analysis
PAR	Pitch Adjusting Rate
PI	Performance Index
PVC	Passive Vibration Control
RMS	Root Mean Square

SDOF	Single-Degree-Of-Freedom
SISO	Single-Input Single-Output
TF	Transfer Function
TP	Test Point
TMD	Tuned Mass Damper
VDV	Vibration Dose Value
2Px	2-Points crossover

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Chapter 1

Introduction

In recent years, it is becoming increasingly frequent to find vibration serviceability problems in new-built buildings [7, 8] and bridges. These problematic structures include offices, classrooms, footbridges [9], concert halls, gyms, shops, operating rooms and stadiums [10]. Continuous developments of building materials and construction techniques, as well as the current trend towards the design of open-plan structures, enable these lightweight and slender structures to satisfy ultimate limit state criteria, but meanwhile might result in excessive human-induced vibrations.

Passive vibration control (PVC) [11] via tuned mass dampers (TMDs) is a quite good option when the resonance problem is within a narrow frequency bandwidth. Besides, TMDs do not require energy input and works with reduced maintenance costs. However, for structures with changing modal parameters or excitation at different frequency ranges, TMDs may be detuned and their performance is drastically affected. Moreover, for low vibration levels, as those usually produced by human excitations, TMDs may show a dead zone, which limit their performance for typical in-service vibrations. In case of non-resonance excitations, their performance is also very limited. For multiple TMDs installed on structures, each TMD works as a perturbation for the others, careful designs have to be undergone then.

Active vibration control (AVC) via active mass dampers (AMDs), mainly inertial mass actuators, has also been proved as an effective tool to control the aforementioned human-induced vibrations [11, 12]. Its ability to target simultaneously multiple modes, its robustness to system changes, and the ability to control vibrations over a much broader frequency range of interest than PVC make it an interesting solution to be explored. Compared with TMDs, inertial mass actuators can control vibrations of much lower levels [13] independently of the waveform; moreover, much smaller moving masses can be used to impart similar damping to a primary structure [14].

1.1 Motivation

This thesis has been carried out within the research project REVES-P (DPI2013-47441-P), titled “Development of novel systems for reducing vibrations in pedestrian structures”, leaded by Dr. Iván Muñoz Díaz. One of the main goals of this project was the development of an AVC system with several inertial mass actuators and sensors distributed along the structure. Thus, this thesis has focused on this goal tackling from the design to the implementation and including both passive and active inertial mass dampers.

PVC and AVC have a good performance for most of the control cases, but only one inertial mass damper (TMD and/or AMD) might not apply to human-induced vibration problems coming from multiple spatially coupled vibration modes with similar natural frequencies. That is, a single TMD cannot control more than one vibration mode, and a single actuator can only control vibration locally. This is better demonstrated in the following example.

A light-weight floor is modelled in ANSYS finite element (FE) program, with the boundary conditions shown in Figure 1.1. The concrete floor has a dimension of $12\text{ m} \times 8\text{ m}$, an effective depth of 18 cm and is supported with five columns at four corners and midpoint. The human-induced vertical vibration serviceability problem might be difficult to solve due to the closely-spaced frequencies (four vibration modes in less than 1 Hz bandwidth) (Figure 1.2). The first six

vibration modes are prone to be excited by human-induced activities: the first (Figure 1.2(a)), second (Figure 1.2(b)), third (Figure 1.2(c)) and fourth mode (Figure 1.2(d)) might be excited by the second harmonics of human walking forces; the fifth (Figure 1.2(e)) and sixth mode (Figure 1.2(f)) might be excited by the third harmonics of human walking forces or the second harmonics of human jumping forces. Furthermore, the placement of control devices is not obvious to decide because vibration needs to be controlled globally (Figure 1.2).

Figure 1.1: A light-weight five-column layout floor simulated in ANSYS.

1.2 Objectives and contributions

To reduce significantly the acceleration response of a lively floor, a control system composed of several inertial mass dampers might be needed. Investigations on designing such control systems are not studied deeply in literature [12]. The main contribution of this thesis is to propose a general/common framework/methodology for designing AVC and PVC systems to control the aforementioned human-induced vibrations.

This framework uses a feedback loop for the design of control parameters and accounts for the nature of human excitation and how humans feel the vibration. For an AVC system, the direct velocity feedback (DVF) is used (for both the design and application); however, for the PVC,

Figure 1.2: Vibration mode shapes and frequencies of the light-weight floor of Figure 1.1.

the feedback loop is only for designing purposes. The methodology developed considers not only single-input single-output (SISO) control systems (i.e., systems composed of one inertial mass damper), but also multi-input multi-output (MIMO) control systems (i.e., systems composed of several inertial mass dampers), especially useful to cancel vibrations coming from closely-spaced natural frequencies.

The strategy finds simultaneously the optimal placements of the TMDs/AMDs and the control parameters (control gain/compensator matrix for AVC; damping ratios and tuning frequencies of TMDs for PVC). To make this strategy implementable and suitable to cancel human-induced vibrations, elements as input-output frequency weighting, output time weighting, band-pass filters, actuator dynamics and non-linearities are considered. Simulations account for all these elements, providing thus reliable results.

Another main contribution of this work is to propose a practical design guide for AVC using a SISO control strategy, which considers the most important practical issues when inertial mass actuators are used, like low-frequency dynamics, stroke saturation, force saturation and spillover problems (instability due to unmodelled high-order dynamics). A non-dimensional system (actuator and structure) is considered in order to generalize the results and extract simple and effective design rules. In addition, experimental results are presented as a validation.

Besides these two contributions, other research objectives are listed:

1. When an optimal MIMO AVC system is designed, a solution to the computational burden problem is proposed, using Coral Reefs Optimization (CRO) algorithm with Substrate Layer (CRO-SL).
2. This work includes vibration control cases of floor structures: from the initial design stage to final practical implementations.

1.3 Thesis organization

The thesis is composed of seven chapters and is organized as follows:

Chapter 1: Introduction

Chapter 1 describes the motivation of this thesis, with objectives, contributions and thesis organization followed.

Chapter 2: Control of human-induced vibrations

In chapter 2 some background knowledge of the related topics of vibration serviceability is presented. Human-induced excitation, human vibration perception are also included, as well as the techniques and devices to control human-induced vibrations such as PVC via TMDs and AVC via AMDs.

Chapter 3: Velocity feedback for controlling human-induced vertical vibrations

Chapter 3 provides interesting and simple design rules for DVF. The relationships between the maximum damping performance as a function of frequency ratios among actuator, structure and band-pass filter are deeply studied. In addition, the actuator stroke and the actuator force are related with the gain margin and actuator damping ratio. The high-pass filter with the ideal integrator can be seen as a band-pass filter. This chapter demonstrates that the cut-off frequency of this band-pass filter must be also considered as a control parameter.

Chapter 4: Common framework for the design of inertial mass dampers

This chapter generalizes the design methodologies for the design of inertial mass dampers (both passive and active) for human-induced vibration. Thus, a general/common framework/methodology taken into account both the nature of the human loading and the perception to vibration is presented. In addition, the design also considers the significant dynamics and non-linearities (both AMDs and TMDs). That is, actuator dynamics and/or non-linearities such as dead zone, stroke and force saturation are (and need to be) considered within the de-

sign process. Band-pass and low-pass filters, which should be introduced to the AVC system works, are also considered.

Chapter 5: Computational optimization using CRO-SL algorithm

In the optimization problems to design the control system of Chapter 4, classical approaches might not achieve good solutions, achieving local optimal, and shows an extremely high computation cost which is unaffordable. Then only a few test points (TPs) and/or actuator/sensor pairs can be considered within these optimization processes. In this chapter, a recently developed co-evolution meta-heuristic, the CRO-SL, is applied to design MIMO AVC for structures subjected to human-induced vibration.

Chapter 6: Designing and experimental implementation of active vibration control

This chapter describes an application example of the aforementioned methodologies: an in-service dining hall floor. The processes from the initial stages related to structure identification to the active control implementation via LabVIEW and CompactRIO (cRIO) of National Instruments (NI) acquisition system are presented in detail.

Chapter 7: Conclusions, publications and future works

Conclusions from the thesis work are drawn, scientific contributions made within this thesis are listed, and some recommendations for future works are finally given.

Chapter 2

Control of human-induced vibrations

A review of the current state-of-the-art in control of human-induced vibration is presented here. In this chapter some background knowledges are presented. Firstly, performance indicators and evaluation methods of the vibration serviceability in standards and codes are presented. Then, human-induced vibration is detailed from human-induced excitation and human vibration perception. Thirdly, the AVC and PVC control devices and system tuning methodologies in recent researches are described.

2.1 Human responses to the vibration

2.1.1 Performance indicators

Normally acceleration is the recommended quantity in standards and codes to evaluate the vibration serviceability. Differently from other quantities, acceleration is an absolute measurement in which no reference is needed. The accelerometer field offers a complete technology at fairly reasonable prices. Thus available instrumentation (Figure 2.1) for measuring acceleration can be used for human-induced vibration measurement.

Figure 2.1: PCB PIEZOTRONICS single axis piezoelectric accelerometer (Model 393A03) used throughout the experimental works.

A frequency-weighted (see W_k in Figure 2.2 included in [4]) acceleration time history is often characterised by a single magnitude for ease of comparisons (usually known as vibration predictor); often the peak value ($a_{w,pk}$) or the root mean square (RMS) value ($a_{w,RMS}$) is used, mathematically defined as follows:

$$a_{w,pk} = \max(|a_w(t)|) \quad (\text{m/s}^2), \quad (2.1)$$

$$a_{w,RMS} = \sqrt{\frac{1}{T} \int_0^T (a_w(t))^2 dt} \quad (\text{m/s}^2), \quad (2.2)$$

in which $a_w(t)$ is the instantaneous frequency-weighted acceleration (obtained from the recorded acceleration $a(t)$), and T is the duration of measurement.

The peak value is usually adopted for the evaluation of footbridges. Table 2.1 shows the comfort classes adopted by one of the most followed guidelines [1]. As it is stated in the table, maximum comfort is achieved for vertical vibration smaller than 0.5 m/s^2 .

For floors, weighted RMS values are adopted in two different versions: basic evaluation and additional evaluation depending on the crest factor (defined as the ratio of the frequency-weighted acceleration peak value to its RMS value $a_{w,pk}/a_{w,RMS}$). A value of 6 is usually used to decide if basic (crest factor smaller than 6) or additional (crest factor greater than 6)

Table 2.1: Defined comfort classes with common acceleration ranges [1]

Comfort class	Degree of comfort	Vertical a_{limit}
CL 1	Maximum	$< 0.50 \text{ m/s}^2$
CL 2	Medium	$0.50 - 1.00 \text{ m/s}^2$
CL 3	Minimum	$1.00 - 2.50 \text{ m/s}^2$
CL 4	Unacceptable discomfort	$> 2.50 \text{ m/s}^2$

evaluation should be adopted. The additional evaluation states the use of the running RMS value (Equation (2.3)) and vibration dose value (VDV) (Equation (2.5)).

Figure 2.2: Frequency weighting function W_k for acceleration output (thicker curve) and its asymptotic definition (thinner curve) [4].

The running RMS evaluation method takes into account occasional shocks and transient vibration by use of a short integration time constant, and is defined by $a_{w,RMS}(t_0)$:

$$a_{w,RMS}(t_0) = \sqrt{\frac{1}{\tau} \int_{t_0-\tau}^{t_0} (a_w(t))^2 dt} \quad (\text{m/s}^2), \quad (2.3)$$

in which τ is the integration time for running averaging (normally $\tau = 1\text{s}$ is recommended), t is the time (integration variable), and t_0 is the time of observation (instantaneous time). The vibration magnitude is defined as a maximum transient vibration value (MTVV), i.e., the

Table 2.2: Recommended values for floor vibration serviceability[2, 3]

floor type	MTVV	$a_{w,pk}$
hospital	0.005	0.007
residential	0.010	0.014
office	0.020	0.028
workshops	0.040	0.057

highest value of $a_{w,RMS}(t_0)$ read during the measurement period:

$$\text{MTVV} = \max[a_{w,RMS}(t_0)], \quad t_0 \in [0, T]. \quad (2.4)$$

Another common performance indicator (or vibration predictor) based on acceleration values is the Response (or Multiplicative) factor (denoted here as R-factor). This is computed as the ratio of the running RMS weighted acceleration to a base curve representative of the human-induced perception. A value of 0.005 m/s² is adopted for vertical vibration in a range of 4 Hz - 8 Hz [15]. The maximum value for a given time history gives the R-factor for that vibration response which can then be compared with recommended limits for the particular environment under consideration; which is more interesting is to obtain the cumulative distribution over time in order to actually obtain the percentage of time an R-factor is exceeded (or not exceeded). Recommended values of MTVV and peak weighted acceleration for floor vibration serviceability can be obtained from [2, 3] (shown in Table 2.2). Note that weighted peak accelerations are obtained multiplying MTVV values by $\sqrt{2}$ (commonly carried out in practice), MTVV values being R-factors multiplied by base value (0.005 m/s²).

Settling time may be used in transient vibration analysis to evaluate the response under impulse loads. Basically it is defined as the time taken for the response to fall and remain within some specified percentage of the maximum peak value of the acceleration response. For the sake of simplicity, in this thesis it is calculated as the time taken for the acceleration response to fall and remain within the range of 0.005 m/s² [16] (representative value from human-induced vibration perception). It can be assumed that vibration under such level can not be perceived

by human.

The VDV, also known as fourth power vibration dose method, is another performance indicator used to characterise the floor response and this explicitly considers the intermittent nature of the excitation, seen as follows:

$$VDV = \int_0^T [(a_w(t))^4 dt]^{0.25} \quad (\text{m/s}^{1.75}). \quad (2.5)$$

The limits placed on VDV values for offices can be found in [2, 3]. Examples of using VDV in the serviceability analysis can be found in Chapters 4 and 5.

2.1.2 Human vibration perception

The vibration that can be perceived by a human depends on the direction of incidence to the human body (Figure 2.3), the frequency content of the vibration (for a given amplitude) and the duration of sustained vibration, among other factors.

The frequency sensitivity for a body position can be considered by attenuating the system response for frequencies where perception is less sensitive. This action is known as frequency weighting and can be undertaken with a suitable filter. Thus, the frequency sensitivity variation for a body position can be taken into account by the frequency weighting functions included in [4] and [5] (Figure 2.3). According to [4], for z-axis vibration of standing and seating, the frequency weighting function (W_k) is a filter with the frequency magnitude shown in Figure 2.2. The W_k function in [4] is used herein as acceleration output frequency weighting in the following chapters.

Human comfort under vibration is also related to the duration of sustained vibration [17]. Thus, persistent vibrations should be penalized in the control design, giving more importance to transient vibration of long duration than those of short duration. This is taken into account by multiplying the system response by an exponential time weighting (i.e., $e^{\alpha t}$), where $\alpha > 0$

Figure 2.3: Directions for vibration according to [4, 5].

adds a constraint in the relative stability of the controlled system. Note that persistent states are penalized more heavily as α is increased. This fact will be explained in Section 4.3.3.

2.2 Human-induced excitation

Human-induced excitation includes activities like jumping, walking, running, dancing and so on. These excitations exhibit mainly periodical time components in a narrow frequency band, which are researched in detail in codes and reports [2, 4, 5, 18].

Usually, due to the periodical nature of rhythmic activities, the force model is assumed as a Fourier series [3, 18] involving the fundamental frequency (pacing frequency) and its multiples. That is, the force model considered is as follows:

$$F_v(t) = Q(1 + \sum_{h=1}^n \alpha_h \sin(2\pi h f t + \phi_h)), \quad (2.6)$$

in which f is the frequency component of repetitive loading, t is the time, h is the harmonic number, and Q is the static weight of a participating person (commonly taken as 700 N). Typical numerical coefficients α_h (or dynamic loading factors, DLFs), and phase angles ϕ_h (or phase lags), are given in Table A.1 (jumping) and Table A.3 (walking) of [18]. Based on this and other researches, Li *et al.* [19] has developed a model to simulate continuous walking forces of both feet, which will be adopted in the simulation of Chapter 4.

In real life, different vibration modes of the structure might be excited with a specific DLF corresponding to a certain frequency range [20]. In [21], [22] and [18], studies were carried out to obtain the DLF values. Based on the DLFs analysis in vertical direction carried out by Brownjohn *et al.* [21], a linear fitting curve has been proposed in this thesis in order to create an input weighting curve to be used within the controller design. This is of great importance when the design process is carried out using impulsive loads. Hence, the probability of exciting a particular mode is considered within the design process. This weighting (Figure 2.4) considers

also the frequency range whose value is smaller than the first harmonics. Furthermore, the weighting is calculated as a sum of all the possible DLFs due to different harmonics. Then after a linear fitting, the following input frequency weighting curve is assumed [6]:

$$g_{IFW}(f) = \begin{cases} 0.02f & 0 \text{ Hz} < f < 1.2 \text{ Hz} \\ 0.37f - 0.42 & 1.2 \text{ Hz} \leq f < 2.4 \text{ Hz} \\ -0.003f + 0.06 & 2.4 \text{ Hz} \leq f < 20 \text{ Hz} \\ 0 & f \geq 20 \text{ Hz} \end{cases}, \quad (2.7)$$

in which f is the frequency. This will be used for the design process in the following chapters, in order to account for the different DLFs depending on the input frequency of human induced loading (HIL).

Figure 2.4: HIL input frequency weighting [6].

2.3 Floor structure dynamic model

For floor structures of different use, the R-factor limits are different. Some areas of a floor will have higher response due to the modes prone to be excited at a particular location and the

modal properties of these modes. The damping ratios and modal participation factors are key properties, and the acceleration response is highly sensitive to them. Generally, areas close to beams and columns will be less responsive than areas in the middle of slabs as these form nodal lines (i.e., limited or no motion) [3].

The modal analysis of the floor vibration can be seen as a decomposition of different natural vibration modes. At a particular location, the motion is formed by the contribution of several vibration modes, and the response of a particular mode can be seen as the one of a single-degree-of-freedom (SDOF) system [23].

For multiple modes or structural points to control, state-space representation ([24]) can be used to simulate the structure model:

$$\begin{aligned}\dot{\mathbf{x}}_s &= \mathbf{A}_s \mathbf{x}_s + \mathbf{B}_{s_w} \mathbf{w}_s, \\ \mathbf{y}_s &= \mathbf{C}_s \mathbf{x}_s + \mathbf{D}_{s_w} \mathbf{w}_s,\end{aligned}\tag{2.8}$$

in which $\mathbf{w}_s$ and $\mathbf{y}_s$ are, respectively, the vectors formed by the perturbation forces and accelerations. In addition, the matrices $\mathbf{A}_s$, $\mathbf{B}_{s_w}$, $\mathbf{C}_s$ and $\mathbf{D}_{s_w}$ are the state-space matrices of the system. $\mathbf{x}_s$, the state vector, will be formed by the displacements and velocities of the spatial coordinates, or by those of the modal coordinates. Then, the system matrices will be different depending on the state-space vector chosen. In Chapter 4, Equation (4.2) shows the state-space model using modal coordinates, which is usually more convenient for the MIMO control design.

2.4 Passive control via tuned mass dampers

PVC reduces the vibration response of a structure by increasing its damping (i.e. energy dissipation) and/or stiffness (i.e., energy storage) through the application of forces generated in response to the movement of the structure. TMD is a PVC device mounted in structures to reduce the amplitude of mechanical vibrations. This is achieved by transferring as much energy as possible from the structure to the TMD by tuning the TMD's frequency and damping ratio.

TMD (Figure 2.5) consists of a mass (m_t), a damper (c_t) and a spring (k_t) attached to primary structure. The frequency tuning is critical to transfer energy from the primary structure to the TMD by relating displacements of the TMD mass. The damping avoids excessive displacements of the TMD mass and makes the TMD operates in a less narrow frequency band, i.e., it is needed to increase the frequency robustness of the device.

In Figure 2.5, it can be observed a CAD model of a TMD. This is mainly formed by masses on a tray supported by four guided springs and two dampers. This TMD has been installed to cancel vibrations of the first vibration mode of a steel-concrete laboratory floor structure, as shown in Figure 2.5(b).

Figure 2.5: TMD: (a) CAD model of a TMD and (b) TMD installed on a laboratory floor.

2.4.1 TMD dynamics

In Figure 2.6, the motion equation of a single TMD installed on a primary structure is as follows (free body diagram of the TMD mass):

$$m_t \ddot{y}_t + c_t \dot{y}_t + k_t y_t - c_t \dot{y}_s - k_t y_s = 0, \quad (2.9)$$

in which y_t and y_s are the absolute displacement of the TMD and the structure node where TMD is installed, respectively. Note that there are non-linearities, such as a dead zone for TMD to start working, that is, TMD cannot move until $|y_s|$ reaches a minimum limit value, which should be considered in the design (see Figure 4.2), and is better explained in Section 4.2.1.

Figure 2.6: Structure and single TMD.

2.4.2 Single TMD tuning algorithm

The TMD design consists of tuning the aforementioned mechanical parameters and locates the position of the TMD to reduce mechanical vibrations. In Den Hartog's classical approach, the optimal TMD parameters are those that minimise the maximum value of the resonant peak at a point for a structure with vanishing damping, with the structure modelled as a SDOF system, representing only the vibration mode to be controlled. Generally, this assumption can be adopted and the TMD is installed where occurs the maximum displacement of the vibration mode to be controlled.

As commented before, the common approach for the TMD tuning is obtained from the assumption that the structure is modelled as a SDOF system with only a particular vibration mode. The two-degree-of-freedom system (structure and TMD) is shown in Figure 2.7, and the motion

equation (free body diagram) of the structure is as follows:

$$m_s \ddot{y}_s + (c_t + c_s) \dot{y}_s + (k_t + k_s) y_s - c_t \dot{y}_t - k_t y_t = f_E, \quad (2.10)$$

in which y_t and y_s are the absolute displacement of the TMD and the structure, respectively; m_s , c_s and k_s are the modal mass, damping coefficient and stiffness related to the mode to be controlled; f_E is the disturbance force applied to the primary structure.

Figure 2.7: Two-degree-of-freedom system (primary structure and TMD).

In Den Hartog's tuning theory [25], the structure displacement is minimized at the resonance frequency and the optimal TMD frequency and damping ($f_{t,DenHartog}$ and $\xi_{t,DenHartog}$, respectively) are shown in the following equations:

$$f_{t,DenHartog} = \left(\frac{1}{1 + \mu} \right) \cdot f_s, \quad (2.11)$$

$$\xi_{t,DenHartog} = \sqrt{\frac{3\mu}{8(1 + \mu)^3}}, \quad (2.12)$$

in which f_s is the frequency of the primary structure's mode to be controlled and μ is the ratio of the TMD mass relative to the primary structure mass (the modal mass of the vibration mode to be controlled, that is, m_t/m_s).

Closed-form solutions can be found in lots of researches [26, 27] (Figure 2.8(a)). Asami *et al.* [26] concluded the approximate analytical solution to H-infinity optimization of TMD design parameters, minimizing the acceleration response of the primary structure:

$$f_{t,Asami} = \sqrt{\frac{1}{1+\mu}} \cdot f_s. \quad (2.13)$$

$$\xi_{t,Asami} = \sqrt{\frac{3\mu}{8(1+\mu)}} \cdot \sqrt{1 + \frac{27\mu}{32}}. \quad (2.14)$$

TMD can be tuned to control the vibration due to a specific vibration mode, however, detuning effects should be considered due to uncertainties in structure dynamics identification. Figure 2.8(b) shows that the frequency tuning is more sensitive than the tuning of damping ratio. Detuning of 10% in the frequency degrades the control performance greatly while a detuning of 50% in the damping does not increase significantly the maximum possible acceleration response.

Figure 2.8: Tuned mass damper: (a) Tuning algorithms (b) Detuning effects.

2.4.3 Multiple TMD tuning algorithm

Traditional TMD tuning only applies to a single TMD system [26]. Designing a multi-TMD (MTMD) system is not straightforward, since each TMD is a perturbation to the others. War-

nitchai and Hoang [28] proposed a method, which can effectively handle a large number of design variables, to simultaneously search for the optimal placement and tuning of TMDs. Li *et al.* [19] proposed an optimization procedure, based on the minimization of maximum RMS acceleration of footbridges, to determine the optimal design parameters of MTMD system.

Other researches, which are based on optimizing a feedback loop for designing MTMD systems, can be found in literatures [29, 30, 31]. This is the idea adopted in this thesis. These contributions assume TMD linear models in order to propose a performance index (PI) in the frequency domain, which is optimized by different algorithms. In addition, these contributions are not orientated to human-induced vibration and the TMD model neglects the nonlinear components. The methodology of considering a TMD as a feedback control system was also used in [32] in order to protect buildings, especially for those buildings that are already constructed but not conveniently protected. According to this, Casado *et al.* [11] designed and implemented a single TMD to cancel human-induced vibration in a real footbridge. Other example is presented in [33], where the feedback design concept [11] and the PI calculation process [12] were combined in order to optimize a MIMO PVC system using TMDs, validated by a simulation design example of a floor structure with closely-spaced frequencies.

2.5 Active control via inertial mass actuators

AVC via inertial mass actuators, has also been proved as an effective tool to control human-induced vibrations [11, 12]. Usually only one accelerometer is used to measure the structure response and a single actuator is used in the control, which is known as SISO AVC [11]. However, there are applications where MIMO AVC has been proposed, (see for example [12], where two accelerometers and two actuators are used to implement a MIMO AVC).

2.5.1 Dynamics of inertial mass actuator

The actuators used in this thesis are inertial mass actuators (also known as proof-mass actuators) that generate forces through acceleration of an inertial mass to the structure on which it is placed. The actuator consists of an inertial (or moving) mass (denoted as m_A) attached to a current-carrying coil moving in a magnetic field created by an array of permanent magnets. The inertial mass is connected to the frame by a suspension system, modelled by a spring stiffness k_A and a viscous damping c_A (Figure 2.9(a)) [23].

The transfer functions (TFs), $G_{A_f}(s)$ (between generated forces and voltage input) and $G_{A_d}(s)$ (between mass displacement and voltage input) are defined as follows:

$$G_{A_f}(s) = \frac{K_A s^2}{(s^2 + 2\xi_A \omega_A s + \omega_A^2)(s + \epsilon)}, \quad G_{A_d}(s) = \frac{1}{s^2 m_A} G_{A_f}(s), \quad (2.15)$$

in which $K_A > 0$ is the actuator gain, ξ_A is the damping ratio, ω_A is the natural frequency and ϵ models the amplifier dynamics. Figure 2.10(a) shows the magnitude of the experimental G_{A_f} in voltage mode. As it is observed, the magnitude presents a roll-off as frequency increases. This roll-off is due to the electrical part and has an influence on the actuator's dynamics when voltage mode is used. This can be modelled by $1/(s + \epsilon)$ in such a way that a third-order model is considered. In Figure 2.10(a), the experimental G_{A_f} is tuned by a third-order and second-order TF.

If current mode for the amplifier is used, the term $(s + \epsilon)$ can be clearly neglected and the actuator damping ratio is much smaller [34], which can be seen from the peak at the resonance frequency, and a second-order model represents clearly the actuator's dynamics. Thus, second-order TF of the actuator (Figure 2.10(b)) can be derived from Equation (2.15):

$$G_A(s) = G_{A_f}(s) \frac{g_A(s + \epsilon)}{K_A} = G_{A_f}(s) \frac{s + \epsilon}{\epsilon} = \frac{g_A s^2}{s^2 + 2\xi_A \omega_A s + \omega_A^2} \quad (2.16)$$

in which $g_A > 0$ is the actuator gain parameter. Note here that $g_A/K_A = 1/\epsilon$.

(a)

(b)

Figure 2.9: Inertial mass actuator: (a) Sketch of typical electrodynamic inertial actuator; (b) APS Dynamic Model 400 Shaker.

(a)

(b)

Figure 2.10: Inertial mass actuator: (a) working in voltage mode; (b) working in current mode.

2.5.2 SISO AVC designing methodologies

Hanagan *et al.* proposed a SISO direct velocity feedback control (DVFC) strategy [35]. Theoretically, DVFC is unconditionally stable in absence of the actuator and/or sensor dynamics. However, in that work, it was shown that the system becomes unstable from a particular gain value, and a saturation in the control voltage had to be introduced in order to keep the control system safe. After [35], several papers have arisen in order to study both the stability of DVFC and its performance, and also, new elements have been included and/or alternative control strategies to enhance both performance and stability have been proposed.

SISO AVC design strategy has been studied widely. Detailed literature review on the SISO AVC strategy can be found in Section 3.1. A SISO AVC system is easier to design and easier to assure stability since the control strategy is defined as collocated (actuator and sensor are placed at the same point physically). Collocated systems exhibit a pattern of alternative resonance/antiresonance (poles/zeros of the TF), and this pattern does not change. This interesting property makes stability analysis an easy task, i.e., the stability can be analysed using classical approaches and it is relatively simple to choose a safe gain that provides good control performance. However, the vibration reduction will only be local and might be ineffective for locally spatially distributed modes with closely-spaced natural frequencies.

If one intends to use several SISO controllers (multi-SISO), it should be taken into account that the control action of one controller is seen by the others as a perturbation. So, multi-SISO control should be seen as a particular case of MIMO control. This particular MIMO is also known as decentralized MIMO control.

2.5.3 MIMO AVC designing methodologies

Compared with SISO AVC system, a MIMO AVC system has the following advantages: being more effective for controlling multiple modes, since large number of disturbance locations and large number of point to be controlled are considered, leading to the ability to reduce

vibration globally. On the other hand, there also exist disadvantages: designing and assuring system stability might not be easy and optimal actuators/sensors placement is not trivial at all. Regarding MIMO AVC, Hanagan *et al.* [36] proposed a PI-based MIMO control using DVFC, which is validated by simulation results. Based on the methodology presented in [36], Pereira *et al.* [12] successfully carried out MIMO AVC using two actuators of a indoor walkway footbridge, which can be seen as a simply-supported structure.

However, in order to carry out reliable and implementable designs, several issues should be considered:

- the actuator bandwidth (i.e., frequency response) significantly affects the stability of the overall control scheme and limits the maximum damping imparted to the structure,
- the actuator stroke saturation, which also limits the maximum damping imparted, could result in dramatic adverse effects on the actuator performance and its hardware,
- the velocity is obtained by integrating the output signal of an accelerometer, necessitating the use of a lossy integrator (a pure integrator + high-pass filter), which affects the stability of the control scheme,
- a low-pass filter may be required to guarantee the finite gain property of the control loop at high frequencies, avoiding spillover problems [37], and
- the frequency bandwidth where humans perceive the vibration [4] may be considered to focus the control effort on the most important vibration modes.

Thus, all these issues are considered in this thesis within a general framework. The methodology can be then applied to complicated structure with multiple coupled modes to control, or a large number of nodes to control, in such a way that the optimal location of actuators might be difficult and computational cost can be extremely unaffordable.

Chapter 3

Velocity feedback for controlling human-induced vertical vibrations

3.1 Introduction

The use of inertial mass actuators to implement AVC is a relatively new research area in civil engineering, so a lot of obstacles have to be overcome before the field can fully mature. Inertial mass actuators can exert forces in one particular direction, which is usually vertical in HIL control implementations. However, the application of this control force, which is similar to adding energy to the structure, might make the closed-control loop unstable and consequences might be devastating. This instability problem usually comes from the actuator dynamics, which are not well considered in the controller design [14, 38]. References [14, 38, 39] conclude that the resonance frequency of the inertial mass actuator should be sufficiently smaller than the natural frequencies of the vibration modes to be controlled.

Different control strategies have been proposed in order to mitigate the aforementioned problems. An important group of strategies try to implement DVFC. For example, in [40], a feed-through term (FFT) between the structure acceleration and actuator force is combined

with an ideal integration of the modified structure output. Other option, which was proposed in [41], is to combine an acceleration feedback with a phase-lag compensator and a high-pass filter. This controller executes an integral action in the bandwidth of interest, which is robust to stroke saturation due to the high-pass filter. The stability margins of the controller proposed in [40] is improved in [42] by an approximate inversion of the actuator dynamics, allowing the application of integral resonant control (IRC) theory [43].

Other control strategies try to custom the actuator dynamics. For example, an inner control loop is proposed in [44]. Other examples are [45, 46], where an inerter added to an AVC system can reduce the natural frequency of the inertial mass actuator, improving the stability of the feedback loop and thus its performance. Finally, more complex control strategies also have been proposed. For example, multi-loop vibration controller, disturbance observer introduced into an outer-loop DVFC and other dynamic compensators are investigated in recent researches [47, 48, 49]. However, these features and new control components undoubtedly complicate the practical implementation and its likelihood of success.

Control laws which are simple and easy to tune are more suitable to be implemented in practice than complex ones. Thus, works like [50] try to propose simple design guides when practical issues must be considered. This chapter considers a generalised system and investigates what sort of performance can be achieved in theory by a perfect DVFC system. In addition, it considers the added complexity of actuator dynamics to demonstrate how this degrades the performance from ideal, assuming a fixed value of cut-off frequency of the band-pass filter (used to implement the DVFC). This band-pass filter hereafter consists of an ideal integration and a high-pass filter.

The chapter extends the results and conclusions of [51] and provides interesting and simple design rules for future implementing. The relationships between the maximum damping performance as a function of frequency ratios among actuator, structure and band-pass filter are deeply studied. In addition, the actuator stroke and the actuator force is related with the gain margin and actuator damping ratio. Thus, it is able to select a proper value of the cut-off

frequency of the band-pass filter which considers the control performance together with the maximum stroke and force of the actuator. In other words, this chapter concludes that the cut-off frequency of the band-pass filter must be also considered as a control parameter, which can improve the control performance for a certain frequency ratio of the first vibration mode of the structure to the actuator. Finally, simulation and experimental results are included in order to illustrate the main conclusions.

This chapter documents the work in [52] and is organized as follows. Section 3.2 briefly explains the ideal DVFC. In Sections 3.3 and 3.3, a methodology to consider and include the practical issues into the AVC is proposed. Section 3.6 details the experimental work, including the dynamic identification of a full-scale laboratory structure, the AVC implementation and frequency response tests. Finally, the main conclusions of this chapter are given in Section 3.7.

3.2 Ideal velocity feedback

A simplified scheme for AVC based on velocity feedback is shown in Figure 3.1. The dynamics of the structure, inertial actuator and the controller are represented by G , G_A and C , respectively; a and d are the acceleration of the structure and the displacement of the actuator's moving mass; f_E and f_C are the excitation force and control force to the structure, respectively.

For this study, the structure is assumed as a SDOF system, considering only the fundamental vibration mode. Thus, the frequency response function (FRF) between the structure's acceleration and the input force is obtained from the following equation:

$$G(j\omega) = \frac{(-1/m_1)\omega^2}{(\omega_1^2 - \omega^2) + j2\xi_1\omega_1\omega}, \quad (3.1)$$

ω being the angular frequency, m_1 , ξ_1 and ω_1 are the modal participation factor, damping ratio and natural frequency of the first vibration mode, respectively.

If $G_A = g_A$ (Equation (2.16)) is constant, i.e., no dynamics from the actuator are introduced

(ideal actuator), and $C(j\omega) = K_C/j\omega$ is a pure integrator (K_C is constant) with infinity magnitude response at zero frequency, the FRF of the closed-loop (CL) system can be obtained from Figure 3.1 as follows:

$$G_{CL}(j\omega) = \frac{A(j\omega)}{F_E(j\omega)} = \frac{-(1/m_1)\omega^2}{(\omega_1^2 - \omega^2) + j(2\xi_1\omega_1 + K_C g_A/m_1)\omega}, \quad (3.2)$$

in which $A(j\omega)$ is the Fourier transform of the structure's acceleration a and $F_E(j\omega)$ is the Fourier transform of the excitation force f_E to the structure. In an ideal DVFC, the added damping to the structure is proportional to $K_C g_A$.

Figure 3.1: General scheme of AVC.

3.3 Practical considerations

Ideal DVFC is not implementable since there exists a lot of practical issues to be considered, such as offset values in signal records, actuator low-frequency response and spillover effect (i.e., instability due to non-modeled high frequency dynamics).

For the sake of simplicity, a second-order model of the inertial mass actuator (an APS Dynamics model 400 electrodynamic actuator) is used (see Equation (2.16) taking $s = j\omega$), whose FRF can be written as [23]:

$$G_A(j\omega) = \frac{-g_A\omega^2}{(\omega_A^2 - \omega^2) + j2\xi_A\omega_A\omega}. \quad (3.3)$$

The ideal integrator must be replaced by a band-pass filter (i.e., a combination of an ideal integration and a high-pass filter) in order to obtain the velocity from acceleration in a frequency range. In addition, a low-pass filter may be required to guarantee stability due to high-order dynamics. That is, finite gain property of the control loop at high frequencies should be assured, in order to avoid spillover problems [37]. The FRF of the filters adopted in this thesis can be written together as follows [51]:

$$C(j\omega) = \frac{jK_C\omega}{(\omega_{HP}^2 - \omega^2) + j2\omega/\sqrt{2}\omega_{HP}} \cdot \frac{\omega_{LP}^2}{(\omega_{LP}^2 - \omega^2) + j2\omega/\sqrt{2} \cdot \omega_{LP}}, \quad (3.4)$$

in which the cut-off frequency of the band-pass and low-pass filters are ω_{HP} and ω_{LP} , respectively. Note that simple second-order Butterworth filter types are adopted here. The band-pass filter mitigates the problem resulted from the signal low-frequency components, alleviating the actuator stroke saturation problem. In addition, the low-pass filter can reduce the high-frequency gain in order to guarantee the stability of the system.

3.4 Non-dimensional study

The objective of this section is to show the best attenuation level that can be achieved with the control scheme of Figure 3.1 as a function of the ratio of the actuator frequency to the structure natural frequency. This is defined as $r_A = \omega_A/\omega_1$, and the ratio of the cut-off frequency of the band-pass filter to the structure natural frequency, defined as $r_{HP} = \omega_{HP}/\omega_1$. In addition, the value of the stroke and force gains (i.e., the magnitude of the TFs between d and f_E , between f_C and f_E , respectively) are analyzed. This analysis shows that there is a tradeoff between the attenuation performance and the maximum stroke and force values.

First of all, the non-dimensional FRF of the structure is obtained from Equation (3.1) as follows:

$$G(j\hat{\omega}) = \frac{(-1/m_1)\hat{\omega}^2}{(1 - \hat{\omega}^2) + j2\xi_1\hat{\omega}}, \quad (3.5)$$

$\hat{\omega}$ (the non-dimensional frequency) being equal to ω/ω_1 . If $\hat{\omega}$ and r_A are considered, the non-dimensional FRF of the actuator defined in Equation (3.3) is as follows:

$$G_A(j\hat{\omega}) = \frac{-g_A\hat{\omega}^2}{(r_A^2 - \hat{\omega}^2) + j2\xi_A r_A \hat{\omega}}. \quad (3.6)$$

The mass displacement can be obtained by integrating twice the mass acceleration, which is obtained by dividing the force by m_A . Thus, the FRF between the actuator mass (m_A) displacement and the control signal is as follows:

$$G_d(j\hat{\omega}) = \frac{G_A(j\hat{\omega})}{m_A\hat{\omega}^2}. \quad (3.7)$$

Finally, the non-dimensional FRF of the controller defined in Equation (3.4) is as follows:

$$C(j\hat{\omega}) = \frac{jK_C\hat{\omega}}{(r_{HP}^2 - \hat{\omega}^2) + j2/\sqrt{2}r_{HP}\hat{\omega}} \cdot \frac{r_{LP}^2}{(r_{LP}^2 - \hat{\omega}^2) + j2/\sqrt{2}r_{LP}\hat{\omega}}, \quad (3.8)$$

r_{LP} being equal to ω_{LP}/ω_1 .

The attenuation performance is defined in terms of the infinity-norm H_∞ (i.e., the maximum of the FRF-magnitude) reduction, which can be written as follows:

$$\text{Attenuation (dB)} = -20\log_{10} \frac{\|G_{a,f_E}\|_\infty}{\|G\|_\infty}, \quad (3.9)$$

in which $\|G_{a,f_E}\|_\infty$ and $\|G\|_\infty$ are the H_∞ norm of the CL and open-loop (OL) systems, respectively. The value of $\|G\|_\infty$ can be approximated, when the flexible structure has a small damping ratio, as follows [53]:

$$\|G\|_\infty = \frac{1/m_1}{2\xi_1\sqrt{1-\xi_1^2}} \cong \frac{1}{2\xi_1 m_1}. \quad (3.10)$$

Then, if Equations (3.5), (3.6) and (3.8) are considered, the resulting FRF of the CL system is:

$$G_{a,f_E}(j\hat{\omega}) = \frac{G(j\hat{\omega})}{1 + G(j\hat{\omega})G_A(j\hat{\omega})C(j\hat{\omega})} = \frac{-(\hat{\omega}^2/m_1)D_{G_A}(j\hat{\omega})D_C(j\hat{\omega})}{D_G(j\hat{\omega})D_{G_A}(j\hat{\omega})D_C(j\hat{\omega}) + j\hat{\omega}^5\hat{K}_C}, \quad (3.11)$$

in which the normalized control gain is defined as $\hat{K}_C = (r_{LP}^2 g_A K_C)/m_1$ and the variables $D_G(j\hat{\omega})$, $D_{G_A}(j\hat{\omega})$ and $D_C(j\hat{\omega})$ are, respectively, the denominators of Equations (3.5), (3.6) and (3.8). Then, if Equations (3.10) and (3.11) are considered, the attenuation level defined by Equation (3.9) becomes:

$$\text{Attenuation (dB)} = -20 \log_{10} \left\| \frac{-(2\xi_1 \hat{\omega}^2)D_{G_A}(j\hat{\omega})D_C(j\hat{\omega})}{D_G(j\hat{\omega})D_{G_A}(j\hat{\omega})D_C(j\hat{\omega}) + j\hat{\omega}^5\hat{K}_C} \right\|_{\infty}. \quad (3.12)$$

The controller design consists of maximizing Equation (3.12) for given values of r_{LP} , r_{HP} and a prescribed gain margin of the CL system. Each design is obtained by calculating the optimum value of $\hat{K}_C$ using the scalar-bounded nonlinear function *fminbnd* of MATLAB. The boundary conditions used in *fminbnd* are $\hat{K}_C = 0$ and $\hat{K}_C$ equal to the gain margin of $G(j\hat{\omega})G_A(j\hat{\omega})C(j\hat{\omega})/\hat{K}_C$ (see Equation (3.11)), which is obtained by the command *margin* of MATLAB, divided by the given gain margin value.

Figure 3.2 shows the attenuation level defined in Equation (3.12) when $\xi_A = 0.3$. The upper limit value of $\hat{K}_C$ is equal to the gain margin of $G(j\hat{\omega})G_A(j\hat{\omega})C(j\hat{\omega})/\hat{K}_C$ divided by one (0 dB gain margin), two (approximately 6 dB gain margin) and four (approximately 12 dB gain margin). Note that there is a set of values of r_{HP} and r_A for which the maximum damping performance is achieved (approximately equal to -32 dB). Then, there are values of r_A for which the attenuation performance is deteriorated (i.e., the maximum damping is not achieved). The range of values of r_A is smaller when the gain margin restriction is increased, a significant attenuation can be obtained for the three gain margin restriction examples. Thus, if an attenuation level of 20 dB is considered as a very good control performance in AVC applications, this can be obtained when $r_A < 0.7$ for a gain margin of 6 dB and when $r_A < 0.6$ for a gain margin of 12 dB.

(a) 0 dB

(b) 6 dB

(c) 12 dB

Figure 3.2: Attenuation (dB) when the actuator damping is equal to 0.3 and the upper limit of $\hat{K}_c$ guarantees a minimum gain margin of 0, 6 and 12 dB.

If a 20 dB attenuation level can be achieved, the next question should be what the most convenient implementation is. In order to illustrate this issue, the stroke and force gain are also plotted. The FRF between the actuator stroke and excitation (see Equation (2.15)) is defined as follows:

$$G_{d,f_E}(j\hat{\omega}) = \frac{-j\hat{\omega}^3 \hat{K}_C / m_A}{D_G(j\hat{\omega})D_{G_A}(j\hat{\omega})D_C(j\hat{\omega}) + j\hat{\omega}^5 \hat{K}_C}. \quad (3.13)$$

Figure 3.3 shows the stroke gain defined as follows:

$$\text{Stroke Gain (dB)} = 20 \log_{10} \|G_{d,f_E}(j\hat{\omega})\|_{\infty}, \quad (3.14)$$

assuming the following actuator parameters: $m_A = 1$ and $\xi_A = 0.3$. Note that the value of $m_A = 1$ is not relevant to show that the actuator stroke gain: i) is reduced when the gain margin is increased, which could be obvious since the damping performance is deteriorated, and ii) depends on r_{HP} , having its minimum value around $r_{HP} = 0.4$ for small values of r_A .

This is better illustrated in Figures 3.4 and 3.5, where the attenuation and the stroke FRFs are plotted for $\xi_A = 0.3$ and $r_A = 0.4$. Note that for a gain margin equal to 0 dB, the increment of r_{HP} reduces the actuator stroke gain (Figure 3.5(a)), and the most important fact is that the maximum actuator displacement occurs at a frequency close to the natural frequency of the structure ($r_{HP} = 0.4$). This is because the CL poles of the actuator have small damping ratios when r_{HP} is not high enough, which might cause problems if there is a perturbation close to actuator frequency. In addition, the problem of stroke gain is less important when the gain margin is 12 dB, where the actuator displacement is approximately constant in the bandwidth of interest (Figure 3.5(b)). However, although the damping performance (attenuation close to 20 dB) is worse than 0 dB-gain margin case, it will impart significant damping for practical implementation.

The actuator force gain should be also studied. The FRF between the actuator force and

(a) 0 dB

(b) 6 dB

(c) 12 dB

Figure 3.3: Stroke gain (dB) when the actuator damping is equal to 0.3 and the upper limit of $\hat{K}_c$ guarantees a minimum gain margin of 0, 6 and 12 dB.

(a) 0 dB

(b) 12 dB

Figure 3.4: Influence of r_{HP} on the attenuation (dB) when the actuator damping is equal to 0.3, $r_A = 0.4$, and the upper limit of $\hat{K}_c$ guarantees a minimum gain margin of 0 and 12 dB.

(a) 0 dB

(b) 12 dB

Figure 3.5: Influence of r_{HP} on the stroke gain (dB) when the actuator damping is equal to 0.3, $r_A = 0.4$, and the upper limit of $\hat{K}_c$ guarantees a minimum gain margin of 0 and 12 dB.

excitation force can be defined as follows:

$$G_{f_C, f_E}(j\hat{\omega}) = \frac{j\hat{\omega}^5 \hat{K}_c}{D_G(j\hat{\omega})D_{G_A}(j\hat{\omega})D_C(j\hat{\omega}) + j\hat{\omega}^5 \hat{K}_c}. \quad (3.15)$$

Figure 3.6 shows the force gain defined as follows:

$$\text{Force Gain (dB)} = 20 \log_{10} \|G_{f_C, f_E}(j\hat{\omega})\|_{\infty}, \quad (3.16)$$

in which $\xi_A = 0.3$. It can be observed that force is reduced in an interval of r_A when the gain margin is increased (around $r_A = 0.4$). However, this issue is better illustrated in Figure 3.7, in which it can be seen that the control force focuses on the natural frequency of the structure. It explains why an attenuation performance around 20 dB can be obtained with a gain margin of 12 dB. Therefore, the conclusion of this study, from the practical implementation point of view, is that a proper gain margin should: i) improve the relative stability of the system, ii) optimize the actuator behaviour into the natural frequency of the structure, and iii) impart enough damping to the structure. In addition, it also shows that a small value of r_{HP} is not always recommended since it increases the stroke gain dangerously at frequencies out of the natural frequency of the structure.

Finally, it is shown herein the influence of the actuator damping into the damping performance. Figure 3.8 shows the attenuation (dB) when the actuator damping is equal to 0.7. It can be seen that the influence of actuator damping deteriorates the attenuation performance for high values of r_A as compared with Figure 3.2. Therefore, the increment of actuator damping may be not advisable if r_A is sufficiently high.

3.5 Design guidelines

Based on the above non-dimensional study, the following design process is proposed:

(a) 0 dB

(b) 6 dB

(c) 12 dB

Figure 3.6: Force gain (dB) when the actuator damping is equal to 0.3 and the upper limit of $\hat{K}_c$ guarantees a minimum gain margin of 0, 6 and 12 dB.

(a) 0 dB

(b) 12 dB

Figure 3.7: Influence of r_{HP} on the force gain (dB) when the actuator damping is equal to 0.3, $r_A = 0.4$, and the upper limit of $\hat{K}_c$ guarantees a minimum gain margin of 0 and 12 dB.

(a) 0 dB

(b) 6 dB

(c) 12 dB

Figure 3.8: Attenuation (dB) when the actuator damping is equal to 0.7 and the upper limit of $\hat{K}_c$ guarantees a minimum gain margin of 0, 6 and 12 dB.

- (i) Identify the structure and actuator dynamics.
- (ii) Calculate the value of r_A .
- (iii) If $r_A \leq 0.6$, fix a gain margin of 12 dB in order to obtain the control gain for a set of values r_{HP} between 0.1 and 0.4.
- (iv) Test the control gain for $r_{HP} = 0.4$ under the most unfavourable perturbation in order to check the maximum level of vibration and actuator mass displacement.
- (v) If the maximum actuator mass displacement is small enough (i.e., sufficiently safe), the value of $r_{HP} = 0.4$ may be reduced.
- (vi) If the maximum vibration level is not achieved, repeat (iii)-(v) for a smaller gain margin.

Note that it is not convenient to reduce the gain margin below 6 dB since the risk of getting the CL system unstable due to the unmodelled dynamics might not be assumed.

3.6 Experimental implementation

To illustrate design guidelines presented in Section 3.5, experimental tests have been carried out on a light-weight structure.

3.6.1 Structure description

The structure used for the implementation is a laboratory slab constructed during the thesis development. This steel-concrete composite slab, with a dimension of 5.8 m $\times$ 1.85 m and a maximum depth of only 14 cm, is simply-supported with a span of 5.5 m. Figure 3.9 shows the elevation of the structure which has a thickness/span ratio of only 1/40. Thus, it is an example of a really slender structure. Figures 3.10 and 3.11 show the components and dimensions of the section. Both positive and negative reinforcements are included, and shear studs have

been placed mainly close to the supports to avoid undesirable cracking and to increase friction between steel and concrete. Finally the constructed structure is shown in Figure 3.12.

Figure 3.9: Elevation of the laboratory floor structure (unit: mm).

Figure 3.10: Typical components of a steel-concrete composite slab.

The supports of the structure can be moved to change the first vibration mode frequency of the structure (Figure 3.13).

3.6.2 Experimental DVFC and result analysis

The experiment is carried out with two APS Dynamics electrodynamic actuators: one is to excite vibration of the structure (model 144, the small one in Figure 3.14), and the other is to

Figure 3.11: Dimensions of the section (unit: mm).

Figure 3.12: A general view of the laboratory floor structure, instrumented with accelerometers underneath.

Figure 3.13: Experimental setup configurations.

control the vibration (model 400, the big one in Figure 3.14); both power amplifiers used to drive the two actuators are APS 145, in voltage mode. They are placed at almost the same point, i.e., at the midpoint of the mid span (Figure 3.14).

Figure 3.14: Experimental SISO AVC implementation.

First of all, the experimental identification of the structure is carried out by measuring the force exerted by the actuator and the acceleration of the structure. The excitation (vertical forces) is configured as a chirp signal from 0 Hz to 30 Hz with a duration of 1800 seconds. In Figure 3.15(a) it can be seen the comparison between the FRF measured and the identified model of two different configurations: configuration 1, $m_1 = 1400$ kg, $\omega_1 = 2\pi \cdot 5.5$ rad/s and $\xi_1 = 0.01$; configuration 2, $m_1 = 1400$ kg, $\omega_1 = 2\pi \cdot 6.60$ rad/s and $\xi_1 = 0.01$. As it is indicated in Figure 3.13, two configurations of different spans have been considered. Configuration 1 is for the original span of 5.5 m, and configuration 2 is for a reduced span of 4.9 m. In addition, using the aforementioned chirp signal, by estimating the TF between actuator force (acceleration of the actuator multiplied by m_A) and the chirp signal, the identified modal parameters of APS Dynamics model 400 electrodynamic actuator are obtained (see Figure 3.15(b)) are $\omega_A = 2\pi \cdot 2.7$

rad/s, $\xi_A = 0.25$, $K_A = 20232$ N/V and $\epsilon = 88$. As has been explained previously, the term ϵ models the high-frequency dynamics of the actuator, and the third-order model G_{A_f} in Equation (2.15) can be represented as G_A (Equation (3.3)) multiplied by $\epsilon/(j\omega + \epsilon)$:

$$G_{A_f}(j\omega) = \frac{-K_A\omega^2}{-\omega^2 + 2j\xi_A r_A \omega_1 \omega + r_A^2 \omega_1^2} \cdot \frac{1}{j\omega + \epsilon}, \quad (3.17)$$

in which r_A is equal to 0.41 and 0.49 for configuration 1 and 2, respectively.

(a) Different configurations of floor structure

(b) controller actuator

Figure 3.15: Experimental identifications.

This dynamics must be inverted (i.e., compensated) to apply the theory described in Section 3.2.

Thus, Equation (3.8) is changed by the following:

$$C_\epsilon(j\hat{\omega}) = C(j\hat{\omega}) \cdot \frac{j\hat{\omega} + \epsilon/\omega_1}{\epsilon}. \quad (3.18)$$

Then, the practical guidelines of Section 3.5 are followed, where $r_{LP} = 10$ and a gain margin of 12 dB are considered for configuration 1. Four different values of r_{HP} are taken into account in order to show the differences between them in terms of attenuation. Figure 3.16(a) shows the simulation FRFs corresponding to the values of $r_{HP} \in \{0.1, 0.2, 0.3, 0.4\}$. Figure 3.16(b) shows the comparison between experimental and simulated results when $r_{HP} = 0.2$. It should be remarked that an attenuation of 20 dB can be achieved with a gain margin of 12 dB. Finally, it should be noted that the maximum stroke was kept below 2.8 cm when a chirp signal from 2 Hz to 9 Hz with a duration of 600 seconds was used as excitation. Then, a gain margin of 12 dB might be used.

The same optimal design, where $r_{LP} = 10$ and a gain margin of 6 dB are taken into account for configuration 2. Four different values of r_{HP} are considered in order to show the differences between them in terms of attenuation. Figure 3.17(a) shows the simulated FRFs corresponding to the values of $r_{HP} \in \{0.1, 0.2, 0.3, 0.4\}$. Figure 3.17(b) shows the comparison between experimental and simulation results when $r_{HP} = 0.4$. It should be remarked that an attenuation of 26 dB can be achieved with a gain margin of 6 dB. Finally, it should be noted that the maximum stroke was kept below 2.7 cm when a chirp signal from 2 Hz to 9 Hz in duration of 600 seconds was used as excitation.

3.7 Conclusions

This chapter studies the practical considerations to implement SISO DVFC for human-induced vertical vibrations. First of all, this work assumes the simplification of a second-order system in order to show the maximum damping that can be imparted by a DVFC implemented with

(a) CL system with different values of r_{HP} in simulation, with $r_A = 0.49$ and gain margin equal to 12 dB

(b) comparison between experimental and simulation results with $r_{HP} = 0.2$

Figure 3.16: FRFs of configuration 1.

(a) CL system with different values of r_{HP} in simulation, with $r_A = 0.41$ and gain margin equal to 6 dB

(b) comparison between experimental and simulation results with $r_{HP} = 0.4$

Figure 3.17: FRFs of configuration 2.

a real inertial mass actuator. The non-dimensional study shows that the maximum damping can be achieved when the actuator natural frequency and the low cut-off frequency of the band-pass filter, which is used to implement the DVFC, are small enough. In addition, the frequency ratios are within a more restricted range when higher gain margins are guaranteed. Secondly, the gain of the TFs that relates the force and stroke of the actuator with the excitation force are studied to show the influences of the aforementioned frequency ratios on the risk of force and stroke saturation. Thirdly, a practical guideline to design the band-pass filter for implementing DVFC is proposed. Fourthly, the model simplification, the design conclusions and a practical guideline are validated experimentally by implementing DVFC examples in a real light-weight steel-concrete composite structure. It is shown that for a structure with a fundamental vibration mode, the bandwidth of the filter and the gain margin of the controller are important design parameters to obtain a very good vibration attenuation performance, which is usually considered greater than or equal to 20 dB.

Chapter 4

Common framework for the design of inertial mass dampers

4.1 Introduction

As explained in Section 2.5.2, SISO AVC design strategy has been studied widely. However, MIMO AVC methodologies are not easy to be found in recent researches. Hanagan *et al.* [36] proposed a MIMO AVC design theory which was validated for experimental implementation by Pereira *et al.* [12]. In [6], the design methodology in [36] was enhanced by including input-output frequency weighting (see Figures 2.2 and 2.4). Respect to the design of PVC systems, there are some works that try to tune TMD parameters as a CL system. Examples of that can be found in [31, 33]. This methodology is very useful to optimally design PVC with multiple TMDs. As has been mentioned above, different control methodologies to design AVC and PVC systems can be found. However, although the control objective for both systems is the same, there is not any work that try to design both systems according to a common framework. Thus, this chapter generalizes the previous design methodologies proposed in [6, 31, 33] in order to design inertial mass dampers (i.e., TMDs and/or AMDs) for human-induced vibrations.

Thus, a general/common framework/methodology taken into account both the nature of the human loading and the perception to vibration is presented. In addition, the design also considers the significant dynamics and non-linearities (both AMDs and TMDs), such as band-pass filters, low-pass filters, voltage and stroke saturation, which are extremely important for a successful practical implementation (endorsed by the author's experience).

This chapter is organized as follows. Section 4.2 explains this common feedback-based control framework. In addition, this section also includes the main elements for TMD and AMD systems and the general design process. Section 4.3 shows an application example for a simulated complex floor model (with closely-spaced frequencies) to demonstrate the validation of this common framework. In Section 4.4, a full-scale laboratory floor structure is shown as a MIMO AVC implementation example of real structures. Finally, some concluding remarks are presented in Section 4.5.

4.2 Feedback-based control framework

The general control scheme is shown in Figure 4.1, in which two main parts can be observed. The first one is the flexible structure dynamics and the second one is the inertial mass control system. The flexible structure is modelled by n vibration modes and p nodes (i.e., possible locations for TMDs/AMDs, sensors and perturbations). The standard state-space representation can be as Equation (2.8) with control force inputs $\mathbf{u}_s$ included:

$$\begin{aligned}\dot{\mathbf{x}}_s &= \mathbf{A}_s \mathbf{x}_s + \mathbf{B}_{s_u} \mathbf{u}_s + \mathbf{B}_{s_w} \mathbf{w}_s, \\ \mathbf{y}_s &= \mathbf{C}_s \mathbf{x}_s + \mathbf{D}_{s_u} \mathbf{u}_s + \mathbf{D}_{s_w} \mathbf{w}_s,\end{aligned}\tag{4.1}$$

in which $\mathbf{u}_s$, $\mathbf{w}_s$ and $\mathbf{y}_s$ are, respectively, the vectors formed by the controlled inputs (i.e., the force imparted by the control system), the perturbation forces and acceleration measured at the locations where accelerometers can be placed and/or human can feel the vibration. In addition, the matrices $\mathbf{A}_s$, $\mathbf{B}_{s_u}$, $\mathbf{B}_{s_w}$, $\mathbf{C}_s$, $\mathbf{D}_{s_u}$ and $\mathbf{D}_{s_w}$ define the complete state-space model

Figure 4.1: Simplified control scheme.

of the structure. If Equation (4.1) is defined in modal coordinates, these state-space matrices are as follows [24]:

$$\begin{aligned} \mathbf{A}_s &= \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{\Omega}^2 & -2\mathbf{Z}\mathbf{\Omega} \end{bmatrix}, \quad \mathbf{B}_{s_u} = \mathbf{B}_{s_w} = \begin{bmatrix} \mathbf{0} \\ \mathbf{\Phi} \end{bmatrix}, \\ \mathbf{C}_s &= \begin{bmatrix} -\mathbf{\Omega}^2\mathbf{\Phi}^T & -2\mathbf{Z}\mathbf{\Omega}\mathbf{\Phi}^T \end{bmatrix}, \quad \mathbf{D}_{s_u} = \mathbf{D}_{s_w} = \mathbf{\Phi}, \end{aligned} \quad (4.2)$$

in which $\mathbf{\Omega}$ and $\mathbf{Z}$ are, respectively, diagonal matrices formed by the natural frequencies $([\omega_1, \dots, \omega_n])$ and the damping ratios $([\xi_1, \dots, \xi_n])$; and $\mathbf{\Phi}$ is the modal matrix with dimension $n \times p$. Note that each k^{th} column of the latter matrix is formed by the k^{th} mode shape values at p nodes. The state vector is defined as: $\mathbf{x}_s = [x_{s_1}, \dots, x_{s_n}, \dot{x}_{s_1}, \dots, \dot{x}_{s_n}]$, $[x_{s_1}, \dots, x_{s_n}]$ being the modal coordinates of the structure, and $[\dot{x}_{s_1}, \dots, \dot{x}_{s_n}]$ being their derivatives.

This general control scheme includes three location matrices, which simplifies the perturbation model and the way of finding the optimal location for TMDs/AMDs and sensors. These matrices

are defined as follows (see Figure 4.1):

$$\mathbf{Ku} = \begin{bmatrix} Ku_{11} & \cdots & Ku_{1q} \\ \vdots & \vdots & \vdots \\ Ku_{p1} & \cdots & Ku_{pq} \end{bmatrix}, \quad \mathbf{Kw} = \begin{bmatrix} Kw_{11} & \cdots & Kw_{1p} \\ \vdots & \ddots & \vdots \\ Kw_{p1} & \cdots & Kw_{pp} \end{bmatrix}, \quad \mathbf{Ky} = \begin{bmatrix} Ky_{11} & \cdots & Ky_{1p} \\ \vdots & \vdots & \vdots \\ Ky_{r1} & \cdots & Ky_{rp} \end{bmatrix}, \quad (4.3)$$

in which q and r are the number of actuators and sensors, respectively. Note that: (i) the j^{th} actuator is placed at the l^{th} node when $Ku_{lk} = 1$, (ii) the k^{th} perturbation is placed at the k^{th} node when $Kw_{lk} = 1$ and (iii) the i^{th} sensor is placed at the l^{th} node when $Ky_{il} = 1$. The control system can be a TMD (see Figure 4.2) or an AMD (see Figure 4.3) system. The general methodology proposed in this work consists of designing the location matrices ($\mathbf{Ku}$ and $\mathbf{Ky}$) and TMD dynamics or AMD control law in order to optimize a PI. Note that TMD can be a passive or a semi-active system, whose model may consider non-linearities and/or time variant parameters. It is also shown in Figure 4.4 the linear time invariant (LTI) hypothesis, which can be used in a first design approximation.

It should be remarked at this point that this common framework allows considering initial conditions of the flexible structure (i.e., $\mathbf{x}_s(t)$ in $t = 0$, where t is the time variable) and/or the perturbation ($\mathbf{w}_s$ and $\mathbf{Kw}$) to excite the structure. This excitation should consider the structure common loads and the serviceability (considering the recommended vibration levels for footbridges, offices, concert halls, gyms or operating rooms are different). In addition, it should also be noted that, although this common framework does not include the shaker dynamics and AMD filters into the optimization, they can be considered as a part of the AMD control law, allowing the optimization of the shakers and/or the filters.

4.2.1 TMD system

The TMD can be modelled as a SDOF system (see Figure 4.4), where the force exerted on the structure ($u_{a,j}(t)$) depends on the relative movement between the TMD mass ($m_{t,j}$) and

Figure 4.2: Inertial mass control system: general TMD system (passive and semiactive) and LTI dynamics approximation.

Figure 4.3: Inertial mass control system: AMD system.

the structure, damping and stiffness. The “linear” ($\hat{y}_{a,l}$) displacement input of the j^{th} TMD located at l^{th} node is modelled as follows:

Figure 4.4: Structure and linear TMD system.

$$\hat{y}_{a,l} = 0 \text{ if } |y_{a,l}| < y_{a,min} \quad (4.4)$$

$$\hat{y}_{a,l} = y_{a,l} - y_{a,min} \text{ if } y_{a,l} \geq y_{a,min}$$

$$\hat{y}_{a,l} = y_{a,l} + y_{a,min} \text{ if } y_{a,l} \leq -y_{a,min},$$

in which $y_{a,min}$ is the minimum structure displacement (in absolute value) for TMD to start working. If the TMD linear stiffness and damping coefficient of the j^{th} TMD are defined as $k_{t,j}$ and $c_{t,j}$, respectively, the TF between the force exerted by the j^{th} TMD and the displacement of the l^{th} structure node can be easily demonstrated taking the Laplace transform ($s = j\omega$) of Equation (2.9) [31]:

$$\frac{U_{a,j}(s)}{\hat{Y}_{a,l}(s)} = -\frac{m_{t,j}s^2(c_{t,j}s + k_{t,j})}{m_{t,j}s^2 + c_{t,j}s + k_{t,j}} = -m_{t,j}s^2 \frac{2\xi_{t,j}\omega_{t,j}s + \omega_{t,j}^2}{s^2 + 2\xi_{t,j}\omega_{t,j}s + \omega_{t,j}^2} = H_{T,f_j}(s), \quad (4.5)$$

in which $\omega_{t,j} = \sqrt{k_{t,j}/m_{t,j}}$ and $\xi_{t,j} = c_{t,j}/2\sqrt{k_{t,j}m_{t,j}}$ are the natural frequency and damping ratio of the j^{th} TMD as an isolated system, respectively, and $U_{a,j}(s)$ and $\hat{Y}_{a,l}(s)$ are the Laplace transform of $u_{a,j}$ and $\hat{y}_{a,l}$. The TF between TMD mass displacement (d_j) and the displacement of the l^{th} structure node can be obtained as follows:

$$H_{T,d_j}(s) = \frac{D_j(s)}{\hat{Y}_{a,l}(s)} = \frac{1}{s^2 m_{t,j}} H_{T,f_j}(s), \quad (4.6)$$

in which $D_j(s)$ is the Laplace transform of d_j .

The TMD model defined in Figures 4.2 and 4.4 considers the significant (linear) dynamics and non-linearities (dead zone and stroke saturation). Note that the simulation used to obtain the optimum design parameters gives information about the maximum mass displacement of each TMD (d_j), which can be used to select the appropriate TMD or to decide if the available TMD can be used for this application in this node.

Finally, TMDs can be seen as systems where the sensor and the actuator are placed at the same node (i.e., the location matrices $\mathbf{K}\mathbf{u}$ and $\mathbf{K}\mathbf{y}$ are diagonal). Therefore, the matrices $q \times q$ (or $r \times r$, since $q = r$) of the TMD system (see Figure 4.2) are as follows:

$$\mathbf{K}_{\text{TMD},f}(s) = \begin{bmatrix} H_{T,f_1}(s) & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & H_{T,f_q}(s) \end{bmatrix}, \quad \mathbf{K}_{\text{TMD},d}(s) = \begin{bmatrix} H_{T,d_1}(s) & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & H_{T,d_q}(s) \end{bmatrix}. \quad (4.7)$$

The optimal design tunes some (or all) of the parameters defined in Equation (4.5) and the location matrices $\mathbf{K}\mathbf{u}$ and $\mathbf{K}\mathbf{y}$ (where for this particular case $\mathbf{K}\mathbf{u} = \mathbf{K}\mathbf{y}^T$) also considering the dead zone and the maximum stroke. Usually design parameters for the TMD are the frequency tuning (the stiffness needed is obtained) and damping ratio (viscous damping is derived), same design parameters as single TMD classical tuning shown in Equations (2.11) and (2.14). The mass will usually be predefined for design purposes.

4.2.2 AMD system

The AMD system can be considered as: i) AMD filters, ii) AMD control law and iii) Actuator dynamics (see Figure 4.3). Note that the control action is divided into two parts; one will be designed as control law and the other includes filter to make the control law implementable. The following subsections will describe these components.

AMD filters

From Figure 4.1, the system output ($\mathbf{y}_s$) is the acceleration measured at each node. It is known that many common control laws use the integration of the acceleration (i.e., DVFC). DVFC [36, 16] has been used in which the velocity is estimated by integrating the acceleration measures. However, a “pure” integration cannot be used since: i) DC component of the control signal will unbalance the inertial mass and ii) low-frequency components will lead to a very high sensitivity to stroke saturation. In addition, undesirable instability might appear due to high-frequency spillover effects. Thus a low-pass ([52]) and a high-pass ([37]) filter should be used in order to get DVFC practically implementable. That is, a band-pass filter is introduced into the control design. Note that the band-pass filter should not influence significantly the bandwidth of interests. The cut-off frequencies of the low-pass filters (denoted by ω_{LP}) must be sufficiently high when compared with the maximum frequency of the vibration modes that may be controlled.

The $r \times r$ matrix of the AMD filters (see Figure 4.3) is as follows:

$$\mathbf{G}_F(s) = \begin{bmatrix} G_{F_1}(s) & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & G_{F_r}(s) \end{bmatrix}. \quad (4.8)$$

Actuator dynamics

The $r \times r$ matrices of the actuator linear part are as follows:

$$\mathbf{G}_{A,f}(s) = \begin{bmatrix} G_{A_{f1}}(s) & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & G_{A_{fr}}(s) \end{bmatrix}, \quad \mathbf{G}_{A,d}(s) = \begin{bmatrix} G_{A_{d1}}(s) & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & G_{A_{dr}}(s) \end{bmatrix}, \quad (4.9)$$

in which the TFs $G_{A_{fi}}(s)$ and $G_{A_{di}}(s)$ are defined in Equation (2.15) ($i = 1, \dots, r$).

AMD control law

The AMD control law is defined by the following $q \times r$ matrix

$$\mathbf{K}_{\text{AMD}}(s) = \begin{bmatrix} H_{A_{11}}(s) & H_{A_{12}}(s) & \cdots & H_{A_{1r}}(s) \\ H_{A_{21}}(s) & H_{A_{22}}(s) & \cdots & H_{A_{2r}}(s) \\ \vdots & \vdots & \ddots & \vdots \\ H_{A_{q1}}(s) & H_{A_{q2}}(s) & \cdots & H_{A_{qr}}(s) \end{bmatrix}. \quad (4.10)$$

The matrix $\mathbf{K}_{\text{AMD}}(s)$ allows configuring any control law. For example, if each $H_{A_{ij}}(s)$ is equal to a gain, this matrix represents a DVF if $\mathbf{G}_{\mathbf{F}}(s)$ carries out an integral action for the bandwidth of interest.

4.2.3 General design process

The proposed design methodology is divided into the following steps:

- Step 1:* Identify the models (flexible structure, inertial mass actuators, electronic equipments, etc.)
- Step 2:* Establish the cut-off frequencies and filter types (Butterworth, Chebyshev, etc.) together with their orders, defining thus the matrix of Equation (4.8).
- Step 3:* Establish the PI. This PI should consider the human vibration perception and excitation characteristics and be related to a magnitude representative of the structure serviceability (recommended vibration levels for footbridges, offices, concert halls, gyms or operating rooms are different).
- Step 4:* Establish the strategy to find the optimal TMD/AMD system together with the location matrices ($\mathbf{K}_{\mathbf{u}}$ and $\mathbf{K}_{\mathbf{y}}$). In this step, the number of actuators and sensors should be taken. Without considering non linear dynamics, it is possible to find a set of possible

locations “*hot points*”, which are candidate for the final placement. This step simplifies and makes computationally affordable the next step.

Step 5: Find the optimal design for the matrices $\mathbf{K}_{\text{TMD}}(s)$ or $\mathbf{K}_{\text{AMD}}(s)$ by using a computational optimization algorithm (gradient-based method, genetic algorithms, etc.). The optimum location matrices ($\mathbf{K}_u$ and $\mathbf{K}_y$) are also results of this step.

4.3 Application example: a simulated complex floor model

The general design process is particularized for the design of two-TMD and two-AMD systems for an example of complex floor model. However, this model is based on an existing dining hall floor structure of a primary-secondary school sited in Madrid (shown in Section 5.2 and Section 6.2). In addition, designs with one TMD and one AMD are also carried out for comparison.

4.3.1 Step 1: Dynamics identification

First of all, Figure 4.5 shows the general arrangement of beams and columns and the layout of the floor structure. A FE model was created in ANSYS Multiphysics [54], using shell elements and 449 nodes. It is an almost rectangular floor with the dimension of 25.5 m $\times$ 20 m $\times$ 0.3 m. As shown in Figure 4.5, the floor is a slab supported by 27 columns. Different connections between the floor and columns are marked with different colors: red ones represent those whose displacements in x, y and z directions are all restricted; cyan ones that are restricted in x and z displacements; blue ones that are restricted in y and z displacements; green ones are connections only restricted in z direction. None of the connections between columns and floor are restricted in rotations. The yellow lines show the meshing grids of the shell elements. The material properties considered are: modulus of elasticity $E = 20 \times 10^9$ N/m², Poisson’s ratio $\nu = 0.15$ and density $\rho = 3000$ kg/m³. The mode shapes, angular frequencies and modal masses

of the first ten vibration modes can be seen in Figure 4.6. The damping ratios of all the modes are assumed as 1%. The layout adopted here is not the actual one shown in Section 6.2. A layout, which is different from the real one but might be realistic, has been adopted here in order to show an example with spatially coupled vibration modes.

Figure 4.5: Floor structure considered.

The actuator considered in this work is an APS Dynamics model 400 electrodynamic shaker (with inertial mass of 30.6 kg), which is shown in Figure 2.9(b). The parameters of Equation (2.15) adopted are: $\omega_A = 13.2$ rad/s (2.1 Hz), $K_A = 12,000$ and $\varepsilon = 47.1$. Finally, the TMD model is assumed LTI as Equation (4.5), where the mass of each TMD is chosen as 1000 kg.

4.3.2 Step 2: Filters design

A second-order Butterworth filter is considered for both band-pass and low-pass filters, and the cut-off frequencies adopted are $\omega_{HP} = 2\pi \cdot 2$ rad/s for the band-pass filter, and $\omega_{LP} = 2\pi \cdot 20$ rad/s for the low-pass filter. The cut-off frequency of the band-pass filter is chosen as 2 Hz.

Figure 4.6: Floor mode shapes, natural frequencies, damping ratios and modal masses.

The value of 2 Hz is selected according to the analysis result of $r_A = 0.41$ for configuration 1 in Section 3.6.2, that is, an approximation of $0.41 \cdot 5.12$ Hz. The TF $G_F(s)$ of each element of matrix defined in Equation (4.8) is as follows:

$$G_F(s) = \frac{s}{s^2 + 2/\sqrt{2} \cdot \omega_{HPS} + \omega_{HP}^2} \cdot \frac{\omega_{LP}^2}{\omega_{LP}^2 s^2 + 2/\sqrt{2} \cdot \omega_{LPS} + \omega_{LP}^2}. \quad (4.11)$$

4.3.3 Step 3: Stablish the PI

The human vibration perception is considered in the controller design by weighting the state vector of the structure $\mathbf{x}_s = [x_{s_1}, \dots, x_{s_n}, \dot{x}_{s_1}, \dots, \dot{x}_{s_n}]^T$ (see Equation (4.1)) as follows:

$$x_{sw_l} = (e^{\alpha t} x_{s_l}(t)) * g_{OFW}(t), l \in [1, \dots, 2n], \quad (4.12)$$

in which $(*)$ denotes the convolution process and $g_{OFW}(t)$ is the impulse response function of a system with the FRF shown in Figure 2.2. Note that the weighted vector $\mathbf{x}_{sw} = [x_{sw_1}, \dots, x_{sw_{2n}}]^T$ is only used to calculate the PI to derive the optimal AMD and/or TMD locations and the design parameters. In other words, the weighting functions are not included in the CL system of Figure 4.1. Note also that all the states are included, displacement and velocity, with indexes from 1 to $2n$.

The design of optimal controllers for unknown disturbances is not trivial since prescribed disturbances are needed within the design process. The solution adopted in this work, similar to that used in [36], is to approximate the influence of zero initial conditions and a spatially distributed, but temporally impulsive, disturbance force by an appropriate initial condition and zero disturbance force. This is achieved by introducing a non-zero initial condition to the velocity states of the structure. Thus, the system perturbation is defined as $\mathbf{x}_s(0) = [x_{s_1}(0), \dots, x_{s_n}(0), \dot{x}_{s_1}(0), \dots, \dot{x}_{s_n}(0)]^T = [0, \dots, 0, \dot{x}_{s_1}(0), \dots, \dot{x}_{s_n}(0)]^T$, where each value of $\dot{x}_{s_k}(0)$

is obtained as follows:

$$\dot{x}_{s_k}(0) = F_0 \phi_{k,max} g_{IFW}(f_k), k \in [1, \dots, n], \quad (4.13)$$

in which the value of F_0 is chosen as 2670 N, which is the normalized maximum force of an idealized heeldrop excitation [16]. In [12], it was proposed to use initial conditions for modal coordinates; however, all the vibration modes of the structure cannot be excited in the same way, taking into account the different DLFs at different frequencies. In order to cope with this fact, the initial conditions are weighted by g_{IFW} (Figure 2.4 and Equation (2.7)), as has been explained in Chapter 2. That is, the input force is weighted according to DLFs of the corresponding frequency range of HIL.

The optimization problem proposed consists of minimizing the following PI:

$$g(\mathbf{z}) = \frac{1}{2} \int_0^{t_f} \mathbf{x}_{sw}^T(\mathbf{z}) \mathbf{Q} \mathbf{x}_{sw}(\mathbf{z}) dt, \quad (4.14)$$

by finding the optimal parameters of $\mathbf{z}$, which includes the parameters of matrices $\mathbf{K}_{TMD}(s)$ (for a TMD system) or $\mathbf{K}_{AMD}(s)$ (for an AMD system), and location matrices $\mathbf{K}_u$ and $\mathbf{K}_y$. The matrix $\mathbf{Q}$ is a $2n \times 2n$ positive definite matrix, which is taken as in [36]:

$$\mathbf{Q} = \begin{bmatrix} \omega_1^2 \phi_{1,max}^2 & \cdots & 0 & 0 & \cdots & 0 \\ \vdots & \ddots & \vdots & \vdots & \ddots & \vdots \\ 0 & \cdots & \omega_n^2 \phi_{n,max}^2 & 0 & \cdots & 0 \\ 0 & \cdots & 0 & \phi_{1,max}^2 & \cdots & 0 \\ \vdots & \ddots & \vdots & \vdots & \ddots & \vdots \\ 0 & \cdots & 0 & 0 & \cdots & \phi_{n,max}^2 \end{bmatrix}, \quad (4.15)$$

$\phi_{k,max}$ being the maximum value of the k^{th} eigenvector ϕ_k . Note that the displacement states are weighted by the natural frequencies, thus making the displacement states comparable to the velocity states. Finally, the value of t_f is the simulation time to obtain the PI, which must

be large enough to achieve the steady state of $g(\mathbf{z})$ (i.e., the weighted state vector $\mathbf{x}_{sw} \cong \mathbf{0}$).

4.3.4 Step 4: Stablish the strategy to find the optimal TMDs/AMDs

This step is to find the value of $\mathbf{z}$ that minimizes the PI defined by Equation (4.14). In order to reduce the computational cost, for structures with a huge number of nodes to consider, a number of possible candidates (“*hot points*”) might be firstly selected according to H-infinity norm and H2 norm calculation [24]. The nodes with the largest values of the following norms are chosen as hot points from all the p nodes of the structure:

$$\begin{aligned} H_{\infty}(l) &= \sum_{m=1}^p \left\| \frac{\mathbf{Y}_{s_m}(j\omega)}{\mathbf{W}_{s_l}(j\omega)} \right\|_{\infty}, l, m \in [1, \dots, p], \\ H_2(l) &= \sum_{m=1}^p \left\| \frac{\mathbf{Y}_{s_m}(j\omega)}{\mathbf{W}_{s_l}(j\omega)} \right\|_2, l, m \in [1, \dots, p], \end{aligned} \quad (4.16)$$

in which $H_{\infty}(l)$ and $H_2(l)$ represents the sum of the p corresponding norms (H-infinity and H2, respectively) of TFs, between acceleration responses of all considered nodes ($\mathbf{Y}_{s_m}$) and the force at the l^{th} node ($\mathbf{W}_{s_l}$). Note that the hot points are selected for those with the highest values in both H_2 norm and H_{∞} norm, given a certain excitation. Thus, these points are the most important locations to place the actuators and/or TMDs. Thus, the 449 nodes considered initially are reduced to 16 points, which drastically reduces the computational cost by 96.44% and 99.88%, for control systems consisting of 1 TMD/AMD and 2 TMDs/AMD, respectively. In this application example, collocated control is assumed for AMD system. Then, the number of sensor and actuators are the same ($q = r$) and they are located at the same place.

Generally speaking, the H_{∞} norm represents the maximum amplitude of the magnitude of the TF. This value is generally known as dynamic amplification factor. That is, H_{∞} norm bounds the output of the system and is usually known to bound the response under harmonic excitations. H_2 norm essentially is the measure of the area under the square of the magnitude of the TF. That is, it is a bound in the response under random (white noise) excitation. In

Figure 4.7, it can be observed that there are 16 points recommendable for both norms.

Figure 4.7: Hot points selection.

4.3.5 Step 5: Find the optimal design for the matrices $\mathbf{K}_{\text{TMD}}(s)$ or $\mathbf{K}_{\text{AMD}}(s)$, $\mathbf{K}_u$ and $\mathbf{K}_y$

This work uses the MATLAB function *fminsearch*, which uses the simple search method of Lagarias *et al.* [55]. This function is used to find the optimal values of $\mathbf{K}_{\text{TMD}}(s)$ or $\mathbf{K}_{\text{AMD}}(s)$ for each possible candidate for locations. Thus, this first step design obtain:

- 16 optimal solutions for SISO TMD and AMD systems.
- 256 optimal solutions for MIMO TMD and AMD systems (two TMDs/AMDs).

Each optimal solution calculates the minimum value of PI (Equation (4.14)) subjected to the stability of the CL system and the restrictions associated with α and damper stroke conditions (i.e., ≤ 79 mm for each AMDs, which is the maximum stroke saturation of the actuator, APS

Dynamics Model 400 Shaker). Note that $\alpha = 0$ imposes strictly stability and $\alpha > 0$ imposes a certain level of stability margin. Typically, the value of alpha is considered as the maximum value of the real part corresponding to OL system poles. Thus, if the selected parameters (by *fminsearch*) of $\mathbf{K}_{\text{TMD}}(s)$ or $\mathbf{K}_{\text{AMD}}(s)$ do not achieve these conditions, the value of $g(\mathbf{z})$ is penalized, avoiding this configuration. The next step is to choose the best among all these optimal solutions.

Using the common framework, 4 control systems are designed: PVC of 1 TMD, PVC of 2 TMDs, AVC of 1 AMD and AVC of 2 AMDs. The system parameters to be designed are listed in Table 4.1. For each cases, system parameters of the best control and the corresponding PIs are shown in Table 4.2. From Table 4.2, the optimum locations for dampers are not obvious, and together with which the system parameters cannot be solved by traditional control design tuning methods. The PI value of MIMO AVC is the smallest compared with others, indicating its best control performance. The following remarks are pointed out:

- Each $H_A(s)$ in $\mathbf{K}_{\text{AMD}}$ (Equation (4.10)) is considered as a control gain here.
- The AVC of 2 AMDs that has resulted is a fully MIMO control with non-zero cross gains.
- The PI is representative of the energy (potential and kinetic) involved into the process for the whole floor.
- The MIMO AVC can reduce 36% more energy of the vibration, than the SISO AVC.
- For single TMD system, $\omega_{t,1} = 2\pi \times 5.5 \text{ Hz} = 34.571 \text{ rad/s}$. That is, it is interestingly tuned at a frequency between the first and second vibration modes.
- For 2-TMDs system, the tuning frequencies are 5.02 and 6.25 Hz, which not only control the first two vibration modes, but also the third vibration mode.

Table 4.1: Tuning parameters

Damping system	Optimised parameters of $\mathbf{K}_{\text{TMD}}(s)$ and/or $\mathbf{K}_{\text{AMD}}(s)$	Number of terms of $\mathbf{K}_{\mathbf{u}}$ and/or $\mathbf{K}_{\mathbf{y}}$ equal to one
1 AMD	$H_{A_1}(s)$	2
2 AMDs	$\begin{bmatrix} H_{A_{11}}(s) & H_{A_{12}}(s) \\ H_{A_{21}}(s) & H_{A_{22}}(s) \end{bmatrix}$	4
1 TMD	$\begin{bmatrix} \omega_{t,1} \\ \xi_{t,1} \end{bmatrix}$	1
2 TMDs	$\begin{bmatrix} \omega_{t,1} & \omega_{t,2} \\ \xi_{t,1} & \xi_{t,2} \end{bmatrix}$	2

Table 4.2: PIs and system parameters of best controls

Damping system	Optimised parameters of $\mathbf{K}_{\text{TMD}}(s)$ and/or $\mathbf{K}_{\text{AMD}}(s)$	Terms of $\mathbf{K}_{\mathbf{u}}$ and/or $\mathbf{K}_{\mathbf{y}}$ equal to one	PI
1 AMD	$[-2228.2]$	$K_{u340,1}$ and $K_{y1,340}$	1.5127×10^{-5}
2 AMDs	$\begin{bmatrix} -3738.3 & -17.8 \\ 1236.0 & -3638.1 \end{bmatrix}$	$K_{u246,1}$, $K_{u350,2}$, $K_{y1,246}$ and $K_{y2,350}$	9.7983×10^{-6}
1 TMD	$\begin{bmatrix} 34.571 \\ 0.115 \end{bmatrix}$	$K_{u360,1}$ (or $K_{y1,360}$)	1.9874×10^{-5}
2 TMDs	$\begin{bmatrix} 39.323 & 31.545 \\ 0.075 & 0.096 \end{bmatrix}$	$K_{u340,1}$, $K_{u355,2}$ (or $K_{y1,340}$, $K_{y2,355}$)	1.4021×10^{-5}
uncontrolled			1.5501×10^{-4}

4.3.6 Evaluation of controller performances

The performance of all the controllers is tested by exciting the structure with jumping input. This perturbation is a representative example of periodic (jumping is a rhythmic human activity, such as dancing or aerobics) loads, which can be generated by humans.

For the jumping excitation, a pedestrian of 800 N-weight is considered and the dynamic load factors (amplitude of each harmonic) of the excitation proposed in [18] are assumed (the periodic excitation is represented as a Fourier series). Five nodes (333, 296, 155, 113 and 95) are chosen to excite (see Figure 4.8) with these frequencies:

- i) 2.56 Hz - Node 333 - 1st vibration mode (5.12 Hz),
- ii) 1.92 Hz - Node 296 - 2nd vibration mode (5.76 Hz),

- iii) 2.22 Hz - Node 155 - 3rd vibration mode (6.65 Hz),
- iv) 2.52 Hz - Node 113 - 4th vibration mode (7.56 Hz),
- v) 2.55 Hz - Node 95 - 5th vibration mode (7.66 Hz).

Figure 4.9 and Figure 4.10 show the simulated time response at node 90 and node 333 for the five cases, respectively. Note that the vibration level of node 90 is practically the same for uncontrolled structure and both PVC systems, since the peak acceleration RMS is 0.0285 m/s^2 , 0.0253 m/s^2 and 0.0243 m/s^2 , respectively. That is, the PVC systems are not able to control the vibration of node 90 for the simulated excitation. For node 333, however, reductions of the peak acceleration RMS are 68.73% and 85.99% for PVC systems, which show excellent control performances. For the same number of dampers, AVC is better than PVC, from the time history response of node 90. Regarding node 333, however, the advantage of AVC is not obvious or even worse than PVC when the control system has only one damper (see Figures 4.10(b) and 4.10(d)).

Figure 4.8: Optimum locations of TMDs (triangle) and AMDs (hexagons); excitation (circles) and response (squares) nodes.

The performance is also evaluated using the VDV, which is used to provide information about

the serviceability of the floor for rhythmic activities [3]. The VDV of all nodes on the floor during 60 seconds of jumping excitation is calculated. Figure 4.11 shows the percentage of floor area where a VDV value (x axis) is exceeded. With the vibration level of the whole floor considered, the order of the performances can be obtained: 1 TMD < 2 TMDs < 1 AMD < 2 AMDs. It can be also seen that MIMO AVC achieves a significant improvement respect to others when the VDV is higher or equal to $0.02 \text{ m/s}^{1.75}$. In addition, MIMO AVC can eliminate the vibration higher than $0.03 \text{ m/s}^{1.75}$. Therefore, it can be concluded that a MIMO AVC may be necessary when a SISO AVC or PVCs cannot guarantee that the vibration level does not exceed a maximum.

4.4 Application example: a full-scale laboratory floor structure

In this section, a MIMO AVC is designed to the laboratory floor presented in Section 3.6. Different from Section 4.3, in which $\mathbf{K}_{\text{AMD}}(s)$ was considered as a control gain and the control dynamics was included into $\mathbf{G}_{\text{F}}(s)$, here $\mathbf{G}_{\text{F}}(s)$ is an all-ones matrix and $\mathbf{K}_{\text{AMD}}(s)$ is not constant and carries out an IRC action. IRC is a simple and robust control scheme working with an integral tracking controller, and has been shown to improve the control performance of HIL problems [42]. Thus $\mathbf{K}_{\text{AMD}}(s)$ is designed assuming IRC strategy, which is implemented.

4.4.1 Step 1: Identifications and FE model calibration

Two actuators have been used to implement the MIMO controllers (APS Dynamics electrodynamic actuators, model 400). The power amplifiers used to drive the two actuators are APS 145 and APS 144, both in voltage mode. Actuators are identified by using an input voltage being chirp signal on rigid floor, by estimating the TF between actuator force (acceleration of the inertial mass multiplied by its value) and the input voltage. The voltage input is configured

Figure 4.9: Time history response of Node 90 under human jumping excitation: (a) without control; (b) 1 TMD; (c) 2 TMDs; (d) 1 AMD; (e) 2 AMDs.

Figure 4.10: Time history response of Node 333 under human jumping excitation: (a) without control; (b) 1 TMD; (c) 2 TMDs; (d) 1 AMD; (e) 2 AMDs.

Figure 4.11: Percentage of floor area where VDV is exceeded.

with a magnitude of 1 V and a sampling frequency of 1000 Hz, within the frequency range of [0.1 Hz, 50 Hz]. Each test lasts 3000 seconds. The third-order model defined in Equation (2.15) is adopted here, and the identified modal parameters for each actuators are the following:

- actuator 1: $\omega_A = 2\pi \cdot 1.47$ rad/s, $\xi_A = 0.55$, $K_A = 20954$ N/V and $\epsilon = 144.5$.
- actuator 2: $\omega_A = 2\pi \cdot 1.5$ rad/s, $\xi_A = 0.45$, $K_A = 10933$ N/V and $\epsilon = 37.7$.

Experimental identification of the structure is carried out by measuring the force exerted by the actuator and the acceleration of the structure. The excitation (vertical forces) is configured as a random signal from 0.1 Hz to 50 Hz with a duration of 600 seconds and a sampling frequency of 1000 Hz. The input voltage is set as 2V. The FRFs are estimated using the MATLAB function *tfestimate*, which carries out the H_1 estimation technique [56]. The FRFs are estimated with window segments of 20s and overlapping adjoining segments 50%. From the FRFs, the well-known and simple peak-picking modal parameters extraction method is used. The estimated modal parameters are shown in Table 4.3.

The information obtained in the experimental analysis has been used to calibrate the FE model. First, the design parameters are chosen. Second, a reasonable range of variation for the design

parameters has been adopted. Third, the influence of each design parameter on the modal response of the structure has been evaluated. Afterwards, the modes identified in the measure and in the FE model are compared using the modal assurance criterion (MAC) [57]. Finally, the minimization of the following objective function is used:

$$OF = \min \left\{ \sum_{i=1} \alpha_i |f_{EXP_i} - f_{FEM_i}| \right\}, i = 1, 2, \dots, n, \quad (4.17)$$

in which f_{FEM_i} is the frequency of mode i in the FE model, f_{EXP_i} the experimental one, α weights the importance applied to each mode, and n is the number of modes considered.

Using an iterative process between ANSYS and MATLAB, and the previous experience of the authors [56, 58], the optimum is reached. The design variables in the optimal solution have been updated in the final FE model before to carry on with the design of the vibration control system.

The FE model was created in ANSYS using shell element, shown in Figure 4.12. For this case, as shown in Figures 4.13 and 4.14, the cracking area (L_{crk}) of the floor and the related decrease in the elastic modulus ($E_2 < E_1$) of the concrete are the variables considered in Section 4.4.1, as well as the contact lengths ($L_{crk} < L_{eff}$) of both simply-supported boundaries. Note here that two spring elements (see Figure 4.14) are also included at both supports to simulate the behaviour of the first torsional mode. These variables are within reasonable ranges:

- $L_{crk} \in [0.1 L_{eff}, 0.5 L_{eff}]$.
- $E_1 \in [18 \times 10^9 \text{Pa}, 22 \times 10^9 \text{Pa}]$.
- $E_2 \in [0.7 E_1, 1.0 E_1]$.
- $d_{touch} \in [0.2 d_{eff}, 1.0 d_{eff}]$.

In Section 4.4.1, the first three vibration modes have different importance factors: $\alpha_1 = 1.0$, $\alpha_2 = 0.9$, $\alpha_3 = 0.75$. The first three vibration modes in calibrated ANSYS model are shown

Figure 4.12: FE model of the structure in ANSYS.

Figure 4.13: Consideration of the cracking concrete in FE modelling.

Figure 4.14: Simulated supporting condition in ANSYS.

in Figure 4.15, and comparison of frequencies between simulation and experiment is listed in Table 4.3.

Table 4.3: Modal properties of the first three natural vibration modes of the floor structure

Mode number i (bending or torsional)	Simulated frequency f_{FEM_i} (Hz)	Experimental frequency f_{EXP_i} (Hz)	Damping ratio (%)
1 (bending)	5.081	4.982	0.64
2 (torsional)	13.561	13.747	0.80
3 (bending)	20.742	21.038	0.94

State-space matrices are established using 45 nodes of the deck (Figure 4.16) of the floor structure, which is updated according to modal parameters of the first three vibration modes. From the obtained model, N16 and N22 are the resulting optimal locations. However, since the actuator masses influence mainly the modal masses, the controllers are redesigned according to the experimental FRFs shown in Figure 4.17.

Thus, the experimental FRFs at N16 and N22 are identified to design the controller. The system model identification considers six vibration modes, whose modal parameters are the following (see Equation (4.2)):

(a) 5.081 Hz, first bending

(b) 13.561 Hz, first torsional

(c) 20.742 Hz, second bending

Figure 4.15: First three vibration modes simulated in ANSYS.

Figure 4.16: Node numbers and coordinates.

- Natural frequencies (Ω): $\omega_1 = 2\pi \cdot 5.0$ rad/s, $\omega_2 = 2\pi \cdot 13.7$ rad/s, $\omega_3 = 2\pi \cdot 21.0$ rad/s, $\omega_4 = 2\pi \cdot 22.8$ rad/s, $\omega_5 = 2\pi \cdot 25.6$ rad/s and $\omega_6 = 2\pi \cdot 27.9$ rad/s.
- Damping ratios (Z): $\xi_1 = 0.009$, $\xi_2 = 0.015$, $\xi_3 = 0.020$, $\xi_4 = 0.020$, $\xi_5 = 0.010$ and $\xi_6 = 0.010$
- Mode shape (Φ):
 - Node N16: $\phi_{1,1} = 0.0177$, $\phi_{2,1} = 0.0020$, $\phi_{3,1} = 0.0138$, $\phi_{4,1} = -0.0204$, $\phi_{5,1} = 0.0017$ and $\phi_{6,1} = 0.0012$.
 - Node N22: $\phi_{1,2} = 0.0194$, $\phi_{2,2} = 0.0284$, $\phi_{3,2} = 0.0014$, $\phi_{4,2} = 0.0020$, $\phi_{5,2} = -0.0034$ and $\phi_{6,2} = -0.0036$.

4.4.2 Step 2: Filters design

Different from the simulation example in Section 4.3, neither band-pass nor low-pass filters are included in the design.

Figure 4.17: Magnitude (in dB) plot of the identified model (black) with the measured data (grey), 1-N16 and 2-N22.

4.4.3 Step 3: Stablish the PI

The IRC controller proposed in [59] is herein considered as the following $q \times q$ diagonal matrix:

$$\mathbf{K}_{\text{AMD}}(s) = \begin{bmatrix} H_{A_{11}}(s) & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & H_{A_{qq}}(s) \end{bmatrix}, \quad (4.18)$$

in which $H_{A_{ii}}(s) = \gamma_i / (s - d_i \gamma_i)$ is the IRC configured for i node, γ_i and d_i being integral damping gains and FTTs respectively. Note that this particular case of MIMO IRC only considers the acceleration measured at node where the actuator is placed. The stability conditions which should be fulfilled, consider the actuator dynamics and the high-frequency model of the system, being the resulting controller robust to spillover. The PI is the maximum value of the FRFs between acceleration from the sensor and force applied by the actuator. In order words, the PI is the maximum value of the FRFs shown in Figure 4.17 but in CL system. Due to the light weight of the structure, FRFs of the structure change with the position of the actuator on the floor. However, the information of the FE model is used to propose a good location to cancel the most important vibration modes. Thus, the actuators are located at nodes N16 and N22.

The proposed IRC controller is designed by minimizing the maximum of the controlled system, from 0 Hz to 15 Hz. In order words, the proposed controller tries to maximize the damping of the first two vibration modes showed in Figure 4.15. Note that the second bending mode (around 21 Hz) of the structure has a zero input frequency weighting value (see Figure 2.4) and is also low in the output frequency weighting value (see Figure 2.2). So $q = 2$ for the $q \times q$ diagonal controller matrix is used, that is, a MIMO AVC system of two actuators to control the first two vibration modes. Thus this strategy can control the most important vibration modes from human-induced excitation and human vibration perception.

4.4.4 Step 4: Stablish the strategy to find the optimal AMDs

As it was mentioned in Section 4.4.3, the strategy to find the optimal AMDs consist of finding the best IRC parameters to minimize the maximum of the FRFs corresponding to N16 and N22 from 0 Hz to 15 Hz.

4.4.5 Step 5: Find the optimal design for the matrices $\mathbf{K}_{\text{AMD}}(s)$

The maximum of the FRFs are obtained by using the identified models in Section 4.4.1. In order to obtain the best IRC parameters, the minimization is carried out by the MATLAB function *fminsearch*, which finds a local minimum of unconstrained multivariable function using derivative-free method. If the CL system is unstable, the function is penalized to avoid these IRC parameters configuration. The resulting controller is as follows (see Equation (4.18)):

$$\mathbf{K}_{\text{AMD}}(s) = \begin{bmatrix} H_{A_{11}}(s) & 0 \\ 0 & H_{A_{22}}(s) \end{bmatrix} = \begin{bmatrix} -749.1/(s + 0.10) & 0 \\ 0 & -353.86/(s + 0.61) \end{bmatrix}. \quad (4.19)$$

4.4.6 Implementation and control performance of MIMO AVC

The controls are carried out using LabVIEW Real-Time Module and NI acquisition system: cRIO-9066 equipped with analog output card NI-9203 and IEPE conditioning cards NI-9234 to measure the structure response. Figure 4.18 shows the LabVIEW block diagram for a two actuator/sensor control system. Two main loops can be observed, one (top) is for running the control (priority loop) at 1000 Hz and the other for showing and recording data. Figure 4.19 describes a sketch of the block diagram. I_1 and I_2 are for configuring the acceleration inputs, and each input is filtered according to the control design, and gained if needed; here the control gain is an all-ones matrix, i.e., $g_{11} = g_{12} = g_{21} = g_{22} = 1$. Finally saturation blocks are included to reduce stroke and force saturation risks. O_1 and O_2 are for configuring the output cards and generating the analog voltage going to the amplifiers which drive the actuators. In

Figure 4.18: LabVIEW block diagram of MIMO AVC.

Figure 4.19: MIMO AVC implementation flowchart.

Figure 4.20, it can be seen the cRIO, amplifiers and a computer recording the results coming from the second loop of the block diagram.

In order to obtain the CL experimental results, impulse hammer (PCB PIEZOTRONICS model 086D50) is used to excite the structure near nodes 16 and 22 (Figure 4.21). Each test includes three impacts to the floor, with an interval of 20 seconds. The experimental responses obtained with the impulse hammer are compared with OL responses in Figure 4.22. Note that the first mode is attenuated more than 30 dB at nodes 16 and 22 and more than 20 dB at node 22. In addition, a peak around 2.2 Hz can be observed, which corresponds with the inertial mass

Figure 4.20: Experimental setups.

Figure 4.21: AVC of the laboratory floor structure.

actuator. This peak is proportional to the instability at low frequencies (i.e., the poles of actuator can be unstable if the values of γ_1 and γ_2 are increased).

Figure 4.22: Magnitude (in dB) plot of the OL (black) and CL (grey) measured data.

4.5 Conclusions

This chapter proposes a generalized framework and a general design process to optimize the control performance of AVC and PVC systems for human-induced vibrations. Thus, any vibration control system design can be made from the same point of view, simplifying it and facilitating the comparison between different vibration control systems by establishing common benchmarks, which can be used to compare actuators, control laws, optimization algorithms etc. in future works. In order to simplify the general design process, it is divided into five common steps. The first one is the structure model identification. The model used in this work defines

the force as input (exerted by humans) and the acceleration as output (variable employed to define the limit state of vibration). The second one is to define the filters needed to implement AVC or to defined the constant system parameters for TMDs. It should be highlighted that these filters allow implementing the control law when accelerometers and shakers are used as sensors and actuators, respectively, for AMDs. The third is to define the PI to obtain the optimal design. In this step, this work provides a guideline of how to consider the interaction between human and the structure; in other words, how human excites the structure and how human feels the vibration. The fourth is to define the strategy to obtain the best results when the number of TMDs and AMD are predefined. Finally, the optimal parameters are obtained by the utilization of an optimization algorithm.

The work concluded with two application examples. The first one is a simulated floor structure with many closely-spaced frequencies, where optimal SISO and MIMO TMDs/AMDs are designed and tested, thus showing the aforementioned advantages of the proposed common framework. The second one is an experimental implementation of a MIMO IRC on a full-scale laboratory floor.

Chapter 5

Computational optimization using CRO-SL algorithm

5.1 Introduction

The methodology in Chapter 4 have been validated in both simulation and experimental implementation. For some structures, the procedure to choose some candidate points (*“hot points”*) might be a good option to reduce effectively the computational cost of classical optimization. However, problems in structural optimization are often characterized by search spaces of extremely high dimensionality and nonlinear objective functions [60]. In these optimization problems, classical approaches do not lead, in general, to good solutions, or in many occasions they are just not applicable, due to the unmanageable search space structure or its huge size, which implies an extremely high computation cost (i.e, the computation time to obtain a local solution may be months). Then, only a few of TPs and/or actuator/sensor pairs can be considered within these optimization processes. In this context, modern optimization meta-heuristics methods have been successfully applied to an important number of structural optimization problems [61]. Meta-heuristic algorithms have been shown as a possibility to obtain an approximate solution to a given problem which cannot be tackled with exact algorithms.

Modern optimization meta-heuristics have been lately the core of a huge research work, focused on improving the performance and search capabilities of this class of optimization techniques. The application of these new optimization algorithms in almost all fields of science, engineering and technology has been massive, generating even much interest on their study and improvement. Many of these meta-heuristics approaches have a *bio-inspired* origin such as evolutionary algorithms (Genetic Algorithms [62], Evolutionary Strategies [63], Evolutionary Programming [64], Differential Evolution [65], among others). These schemes are based on concepts borrowed from natural evolution and survival of the fittest individuals in nature. Similarly, Ant Colonies Optimization [66] are based on the social behavior of ants, Particle Swarm Optimization approaches [67] imitate the behavior of birds flocks or fish schools, and Artificial Immune System algorithms [68] are focused on imitating the behavior of the immune system in animals. Many alternative meta-heuristics have arisen in the last years, such as Artificial Bee Colony [69], which imitates the bees behaviour when locating and bring food to the hive.

There are different meta-heuristics that have been specifically applied to structural engineering problems. Genetic and evolutionary algorithms have been applied to the discrete optimization of structures in [70]. There have been other works that applied genetic algorithms in structural optimization problems [71, 72, 73, 74]. In the last few years, alternative modern meta-heuristics based on physics process have been applied to structural optimization problems, such as the Big-Bang Big-Crunch algorithm [75], and the colliding bodies optimization algorithm [76].

In this chapter, the recently developed co-evolution meta-heuristic, the CRO-SL (Coral Reefs Optimization with Substrate Layer) [77], is applied to design MIMO AVC for structures subjected to human-induced vibration. This optimization algorithm is particularly interesting when vibrations on complex floor structures with several closely-frequency spaced vibration modes have to be cancelled. Thus, this algorithm promotes a powerful evolutionary-like search, ideal for solving high-burden optimization problems, which will be shown to be very effective in this particular problem of MIMO AVC design. The algorithm's performance has been evaluated and compared with several reference algorithms in a FE model of a complex floor structure.

This chapter documents the work in [78], which was carried out in collaboration with the research group GHEODE (Grupo de Heurísticos Modernos de Optimización y Diseño de Redes of the Universidad de Alcalá), led by Prof. Sancho Salcedo. This work has two main parts: the first one is to configure the CRO-SL proposed in [77] to a AVC problem, which was mainly carried out by GHEODE; the second one is the application example, which consists of modelling a real structure in order to define a PI to be optimized by CRO-SL. Thus, the developed MATLAB files can calculate any PI to obtain the actuator/sensor locations and the controller. These files have considered the aforementioned practical control issues such as actuator's low-frequency dynamics, stroke saturation, force saturation and spillover problems. The second part was carried out by the PhD candidate. The structure of the remainder of this chapter is as follows: Section 5.2 describes the structural model of the application example used in Section 5.4. Section 5.3 presents the main characteristics of the original CRO, including the different operators and the algorithm's dynamics. In addition, Section 5.3 also describes the proposed CRO-SL version, including the definition of *substrate layer*, and, in this case, how it represents the co-evolution of different searching mechanisms with the rules of the CRO. Section 5.4 presents the aforementioned application example, where the algorithm's performance is evaluated and compared with a reference algorithm. Section 5.5 closes the chapter by giving some conclusions and remarks.

5.2 Problem definition

The optimization problem (Equation (4.14)) is tackled in this chapter. This is particularized for MIMO AVC assuming DVFC, that is, $\mathbf{K}_{\text{AMD}}(s)$ is a gain matrix to be designed and filter matrix $\mathbf{G}_{\mathbf{F}}(s)$ is established practically to carry out the integration rule. Additionally, the position matrices $\mathbf{K}_{\mathbf{y}}$ and $\mathbf{K}_{\mathbf{u}}$ are also part of the optimization procedure. Thus, the step 4 (Stablish the strategy to find the optimal AMDs) can consider a large number of nodes and/or more sensors/actuators. This section explains how to use the CRO-SL as a solver of minimizing an example of PI.

The structure considered is a dining hall floor (Figure 5.1), whose FE model is made in ANSYS. Note that in Section 4.3 a created FE model is shown, whose columns are rearranged based on the real floor structure explained in this section. The mode shapes, natural frequencies, damping ratios and modal masses of the first ten vibration modes can be seen in Figure 5.2. The damping ratios are obtained according to a preliminary approximation of Rayleigh Damping to make all the ratios close to 1% (Figure 5.2). This FE model is made according to only technical drawings (Figure 6.2), and can be seen as the first approach to identify the structure dynamics, though only modelling part of the whole structure (Figure 6.9). This model will be calibrated using operational modal analysis (OMA) and experimental modal analysis (EMA) in Section 6.2.

Figure 5.1: Floor structure considered for the optimization problem.

5.3 Proposed Co-evolution Coral Reefs Optimization with Substrate Layer

This section presents the CRO-SL used in this chapter for tackling the MIMO AVC design. First, the basic CRO algorithm is presented, which will be modified with a *substrate layer*

Figure 5.2: Floor mode shapes, natural frequencies, damping ratios and modal masses.

in order to obtain a competitive co-evolution algorithm with different exploration procedures involved.

5.3.1 Basic CRO

The CRO [79], [80] is an evolutionary-type algorithm based on the behaviour of the processes occurring in a coral reef. Let $\mathcal{R}$ be the reef represented by an $R_1 \times R_2$ grid, where each position (i, j) of $\mathcal{R}$ is able to allocate a coral or a colony of *corals* ($\mathcal{C}_{i,j}$), i.e., tentative solutions to the current optimization problem at hand. Therefore, a coral encodes a candidate solution to the optimization problem. This solution is encoded as a real-valued vector that represents the parameters of the AVC ($\mathbf{z}$ in Equation (4.14)). The quality of a solution is evaluated using the objective function given in Section 4.3.

The CRO algorithm first initializes some random positions of $\mathcal{R}$ with random corals, and leaves some other positions empty. These holes in the reef are available to host new corals that will be able to freely settle and grow in later phases of the algorithm. The rate between free/occupied positions in $\mathcal{R}$ at the beginning of the algorithm is a parameter of the CRO algorithm, denoted as ρ_0 with $0 < \rho_0 < 1$.

The second phase simulates the processes of reproduction and reef formation. The different reproduction mechanisms available in nature are recreated by sequentially applying different operators:

1. **External sexual reproduction or Broadcast Spawning.** Broadcast spawning consists of the following steps at each iteration k of the algorithm:

- 1.a. A random fraction of the existing corals is selected uniformly, turning these corals into broadcast spawners. The fraction of broadcast spawners with respect to the overall amount of existing corals in the reef will be denoted as F_b .

1.b. Several coral larvae are formed. To generate each new larva, two broadcast spawners are selected and a crossover operator or any other exploration strategy is applied. Note that once two corals have been selected to be the parents of a larva, they are not chosen anymore at iteration k for reproduction purposes. Corals' selection can be done randomly, uniformly, or using any fitness proportionate selection approach (e.g. roulette wheel).

2. Internal sexual reproduction or Brooding. Hermaphrodite corals mainly reproduce by brooding. This reproduction is modelled by means of any kind of mutation mechanism and takes place on a fraction of corals of $1 - F_b$. A percentage P_i of the coral is mutated.

3. Larvae setting. Once the larvae are formed either through external or internal reproduction, they will try to set and grow in the reef. Each larva will randomly try to set in a position (i, j) of the reef and, if the location is free, it will set. If the location is already occupied, the new larva will set only if its health function (fitness) is better than that of the existing coral. Moreover, the CRO algorithm defines a parameter $\mathcal{N}_{att}$, that determines the maximum number of tries a larva can attempt at each iteration h .

4. Asexual reproduction. Corals reproduce asexually by budding or fragmentation. The CRO models this mechanism in the following way: the whole set of corals in the reef are sorted according to their level of health value (given by $f(\mathcal{C}_{ij})$). Then, a small fraction (denoted as F_a) of the best available corals are duplicated and mutated (with probability P_a) to provide variability, and try to settle in a different part of the reef as in Step 3.

5. Depredation. Corals may die during the reef's formation. Therefore, at the end of each reproduction iteration k , a small number of corals in the reef can be depredated, thus liberating space in the reef for next coral generation (iteration $h + 1$). The depredation operator is applied with a very small probability (P_d) to a fraction (F_d) of the corals in

the reef with worse health.

The flowchart diagram of the CRO algorithm, with the different CRO phases (reef initialization and reef formation), along with all the operators described above, is as follows:

Require: Valid values for the parameters controlling the CRO algorithm

Ensure: A single feasible individual with optimal value of its *fitness*

- 1: Initialize the algorithm
 - 2: **for** each iteration of the simulation **do**
 - 3: Update values of influential variables: predation probability, etc.
 - 4: Sexual reproduction processes (broadcast spawning and brooding)
 - 5: Settlement of new corals
 - 6: Predation process
 - 7: Evaluate the new population in the coral reef
 - 8: **end for**
 - 9: Return the best individual (final solution) from the reef
-

5.3.2 CRO with substrate layer

The original CRO algorithm described above is based on the main processes of coral reproduction and reef formation that occur in nature. However, there are many more interactions in real reef ecosystems that can be also modelled and incorporated into the CRO approach to improve its performance in optimization and search problems. For example, different recent studies have shown that successful recruitment in coral reefs (i.e., successful settlement and subsequent survival of larvae) strongly depends on the type of substrate on which they fall after the reproduction process [81]. This specific characteristic of coral reefs was first included in the CRO in [82], in order to solve different instances of the Model Type Selection Problem for energy applications. This new version of the CRO was named CRO-SL (Coral Reefs Optimization algorithm with Substrate Layer). In [77], a new version of the CRO-SL was presented, where several substrate layers which provide a different search procedure each, were defined in the CRO. Following this version of the algorithm, the CRO-SL general approach for competitive co-evolution, where each substrate layer represents different processes (different models,

Figure 5.3: Example of different substrate layers in a CRO-SL algorithm.

operators, parameters, constraints, repairing functions, etc.). The use of CRO-SL as a competitive co-evolution algorithm, was successfully tested in different applications and problems such as micro-grid design [83], vibration cancellation in buildings [31] or in the evaluation of novel non-linear search procedures [84]. In this section, the main ideas of the CRO-SL as co-evolution search algorithm are described, which will be applied to solve the problem of AVC design.

The inclusion of substrate layers in the CRO can be done in a straightforward way: the artificial reef considered in the CRO is redefined in such a way that each cell of the square grid Ψ representing the reef is now defined by 3 indexes (i, j, m) , where i and j stand for the cell location in the grid, and index $m \in T$ defines each substrate layer, by indicating which structure (search operator in this case) is associated with the cell (i, j) of the reef. Each coral in the reef is then processed in a different way (with a different search operator) depending on the specific substrate layer in which it falls after the reproduction process. Note that this modification of the basic algorithm does not imply any change in the corals' encoding. Figure 5.3 shows an example of the CRO-SL reefs structure, with five different substrate layers. In this example, each substrate is assigned to a different exploration process, Harmony Search (HS), Differential Evolution (DE), Gaussian mutation, 1-point crossover and 2-points crossover.

The CRO-SL can be seen as a generalization of the original CRO, that does not modify the dynamics of the algorithm, which can still follow the basic steps given in the previous subsection.

The only difference between the CRO-SL and the basic CRO is the specific implementation of the broadcast spawning procedure, which now depends on the specific substrate to which each coral (solution) is associated. Note that the CRO-SL is in fact a competitive co-evolution searching procedure: the substrate layers of the CRO-SL promote the co-evolution process between different search operators, without the necessity of defining different populations. The competitiveness of the approach is given by the fact that CRO dynamics is based on a procedure of larvae settlement which involves competition among corals.

5.4 Computational evaluation and comparisons

The examples carried out to evaluate the proposed CRO-SL in this context consist of two approaches of AVC, a SISO and a MIMO cases, where MIMO is configured with $q = r = 2$ (each pair is placed at same location, as explained in Section 4.3.4).

The CRO-SL parameters used in the experiments are shown in Table 5.1.

Table 5.1: Parameters values used in the CRO-SL.

Parameter	Description	value
Reef	Reef size	120
F_b	Frequency of broadcast spawning	97%
Substrates	HS, DE, 2Px, GM, MPx	5
$\mathcal{N}_{att}$	Number of tries for larvae settlement	3
F_a	Percentage of asexual reproduction	5%
F_d	Fraction of corals for depredation	5%
P_d	Probability of depredation	5%
α	Maximum number of iterations	1000

Five different substrates layers have been used within the CRO-SL in the experiments carried out:

1. HS: Mutation from the Harmony Search algorithm. Harmony Search [85] is a population based meta-heuristic that mimics the improvisation of a music orchestra while its composing a melody. This method integrates concepts such as harmony aesthetics or

- note pitch as an analogy for the optimization process, resulting in a good exploratory algorithm. This property is also applicable to the operator based on its behaviour. HS controls how new larvae are generated in one of the following ways: i) with a probability $\text{HMCr} \in [0, 1]$ (Harmony Memory Considering Rate), the value of a component of the new larva is drawn uniformly from the same values of the component in the other corals. ii) with a probability $\text{PAR} \in [0, 1]$ (Pitch Adjusting Rate), subtle adjustments are applied to the values of the current larva, replaced by any of its neighbouring values (upper or lower, with equal probability).
2. DE: Mutation from Differential Evolution algorithm (with $F = 0.6$). This operator is based on the evolutionary algorithm with the same name [86], a method with powerful global search capabilities. DE introduces a differential mechanism for exploring the search space. Hence, new larvae are generated by perturbing the population members using vector differences of individuals. Perturbations are introduced by applying $x'_i = x_i^1 + F(x_i^2 - x_i^3)$ for each encoded parameter on a random basis. x' corresponds to the output larva, x^t are the considered parents (chosen uniformly among the population), and F determines the evolution factor weighting the perturbation amplitude.
 3. 2Px: Classical 2-points crossover. The crossover operator is the most classical exploration mechanism in genetic and evolutionary algorithms [62]. It consists of coupling to individuals at random, choosing two points for the crossover, and interchanging the genetic material in-between both points. In the CRO-SL, one individual to be crossed is from the 2Px substrate, whereas the couple can be chosen from any part of the reef.
 4. MPx: Multi-points crossover. Similar to the 2-points crossover, but in this case a number k of crossover points are selected, and a binary template decides whether parts of the individuals are interchanged.
 5. GM: Gaussian Mutation, with a σ value linearly decreasing during the run, from $0.2 \cdot (A - B)$ to $0.02 \cdot (A - B)$, where $[B, A]$ is the domain search. Specifically, the Gaussian

probability density function is:

$$f_{G(0,\sigma^2)}(x) = \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{x^2}{2\sigma^2}}.$$

The reason of adapting the value of σ along the generations is to provide a stronger mutation in the beginning of the optimization, while fine tuning with smaller displacements nearing the end. The mutated larva is thus calculated as: $x'_i = x_i + \delta N_i(0,1)$, where $N_i(0,1)$ is a random number following the Gaussian distribution.

5.4.1 Results

Using the framework proposed in Chapter 4, the design process is as follows:

Step 1: Dynamics identification. This is carried out in Section 5.2.

Step 2: Filters design. The adopted cut-off frequencies are: $\omega_{HP} = 12.6$ rad/s (2 Hz), $\omega_{LP} = 125.6$ rad/s (20 Hz).

Step 3: Stablish the PI. This is previously explained in Section 4.3.3. Stroke and voltage saturation of the actuator APS Dynamics Model 400 Shaker are selected as 7.9 mm and 2 V, respectively; and α is 0.3547 (see Equation (4.12)).

Step 4: Stablish the strategy to find the optimal AMDs. Different from Section 4.3.4, “*hot points*” are not selected here, instead all the nodes in FE model are considered as possible location for AMDs.

Step 5: Find the optimal design for the matrices $\mathbf{K}_{AMD}(s)$, $\mathbf{K}_u$ and $\mathbf{K}_y$. CRO-SL is utilized to obtain the optimal design parameters.

Table 5.2 shows the results obtained by the proposed CRO-SL, compared to different alternative algorithms. Specifically, all the algorithms which form the substrate layers in the CRO-SL approach have been tried on their own: the CRO with a single substrate has been run, with the

same number of function evaluations than in the case of five substrates. This will show how the competitive co-evolution process promoted by the CRO-SL is able to obtain accurate solutions for the AVC design and location problem. In addition, a comparison with a high-performance recently proposed meta-heuristics for structures optimization, the Enhanced Colliding Bodies Optimization (ECBO) [87] is included. This approach is an improved version of the Colliding Bodies Optimization (CBO) [76], which includes memory and a specific mechanism to scape from local optimal. The computer code for the ECBO has been released by the authors [88], so we have used that implementation with small adaptations to the problem at hand.

In Table 5.2 it can be seen how the CRO-SL (five substrates) obtain the best performance, both in the SISO and MIMO cases, with two actuators/sensors ($p = q = 2$). In the SISO case, the differences among different methods are small, since it is the simplest case. In fact, the CRO-SL and CRO with DE substrate and the ECBO algorithm obtain a similar value of the PI. In this case, the HS substrate is the next algorithm in terms of performance, whereas the Gaussian mutation and the two crossover operators (2-points and multi-point) are the poorest in terms of the PI. In the case of the MIMO, the differences are much more significant. The CRO-SL with five substrates in co-evolution clearly obtains the best performance. The ECBO algorithm also performs well in this problem, as it obtains the second best result overall. In this case, the DE, is the third best approach among the tested algorithms. The HS is also the next better substrate in this version of the problems, and again the Gaussian and crossover operators do not obtain competitive results on their own in this case.

Figure 5.4 shows the ratio of times that every substrate generates the best larva per generation in the CRO-SL approach. It is possible to see how the 2-points crossover and the MPx crossover are the two exploration operators that contribute the most to the CRO-SL search. The contribution of the DE substrate is also significant, and it grows during the search. Note that the HS and Gaussian substrates are the ones which contribute the less to the search, especially, the Gaussian substrate does not seem to be effective in this problem, and barely contributes to obtain the best solutions in each iteration of the CRO-SL.

Table 5.2: Comparison of the results obtained in the two case-studies taken into account (SISO and MIMO) with different algorithms, in terms of the fitness function considered (Equation (4.14)).

	SISO		MIMO	
	Min	Mean	Min	Mean
CRO-SL	$1.8923 \cdot 10^{-5}$	$1.8923 \cdot 10^{-5}$	$1.282 \cdot 10^{-5}$	$1.2835 \cdot 10^{-5}$
HS	$2.0326 \cdot 10^{-5}$	$2.039 \cdot 10^{-5}$	$1.681 \cdot 10^{-5}$	$1.832 \cdot 10^{-5}$
DE	$1.8923 \cdot 10^{-5}$	$1.8923 \cdot 10^{-5}$	$1.4199 \cdot 10^{-5}$	$1.5431 \cdot 10^{-5}$
2Px	$3.2173 \cdot 10^{-5}$	$3.2205 \cdot 10^{-5}$	$2.3401 \cdot 10^{-5}$	$2.3491 \cdot 10^{-5}$
GM	$2.5513 \cdot 10^{-5}$	$2.5698 \cdot 10^{-5}$	$2.2238 \cdot 10^{-5}$	$2.2549 \cdot 10^{-5}$
MPx	$3.2155 \cdot 10^{-5}$	$3.2304 \cdot 10^{-5}$	$2.3375 \cdot 10^{-5}$	$2.3577 \cdot 10^{-5}$
ECBO	$1.8923 \cdot 10^{-5}$	$1.8923 \cdot 10^{-5}$	$1.382 \cdot 10^{-5}$	$1.3832 \cdot 10^{-5}$

Figure 5.4: Ratio of times that each substrate produces the best larva in each iteration of the CRO-SL algorithm, for the AVC-MIMO design.

Following, the best solutions obtained by the CRO-SL algorithm in the SISO and MIMO cases is analyzed in detail. The best result obtained by the CRO-SL for the SISO case (also reached by ECBO and DE) is the following:

$$\begin{aligned} \mathbf{K}_{\text{AMD}} &= -4758.6, \\ \mathbf{K}_{u_{329},1} &= 1, \mathbf{K}_{y_{1,329}} = 1, \\ g(\mathbf{z}) &= 1.8923 \times 10^{-5}. \end{aligned} \tag{5.1}$$

This design is useful to reduce the vibration of the 1st, 2nd, 5th, 7th and 9th modes. Note that the frequency weighting function W_k of Figure 2.2 shows that the modes with natural frequency higher than 10 Hz are less important from the human perception point of view. Then, this SISO control is mainly designed for 1st, 2nd and 5th mode. The location of the actuator/sensor pair is shown in Figure 5.5 (triangle). This point is a location where these three vibration modes can be measured and controlled.

Figure 5.5: Optimum location of actuator/sensor pair for SISO (triangle) and MIMO (hexagons) designs. Excitation (circles) and response (squares) nodes.

The best result obtained by the CRO-SL for the MIMO case (best solution over all comparison algorithms) is the following:

$$\begin{aligned}
\mathbf{K}_{\text{AMD}} &= \begin{bmatrix} -1272.5 & -1367.8 \\ \cong 0 & -3753.2 \end{bmatrix}, \\
\mathbf{K}_{u83,1} &= 1, \mathbf{K}_{y1,83} = 1, \\
\mathbf{K}_{u328,2} &= 1, \mathbf{K}_{y2,328} = 1, \\
g(\mathbf{z}) &= 1.282 \times 10^{-5}.
\end{aligned} \tag{5.2}$$

The two actuator/sensor pairs are shown in Figure 5.5 (hexagons). The first actuator/sensor location is practically the same as SISO design (node 328). The second actuator/sensor pair is useful to reduce the vibration of the 3rd, 4th, 6th, 8th and 10th mode. However, if the human perception is considered, the actuator placed at node 83 is mainly focused on 3rd and 4th mode. Therefore, this MIMO design can reduce the vibration of the first five vibration modes, all which can influence the human comfort. It should be also remarked that the optimum matrix gain $\mathbf{K}_{\text{AMD}}$ configures the first actuator as a two inputs - one output system, whereas the second actuator is configured as a decentralized collocated control system. Therefore, although the actuator/sensor locations may be obvious, since MIMO AVC reduces the vibration level of the first five vibration modes, the configuration of $\mathbf{K}_{\text{AMD}}$ together with the problem of actuator/sensor locations cannot be solved with traditional control design tuning methods. Finally, the value of PI is reduced 32.25% with respect to the SISO case.

The performance of both controllers (SISO and MIMO) is also tested by exciting the structure with jumping and heel-drop inputs. These two perturbations are representative examples of periodic (jumping is a rhythmic human activity, such as dancing or aerobics) and impulsive (heel-droop allows measuring the damping imparted by controllers, evaluating the transient response evaluation of floor structures and checking stability properties) loads, which can be generated by humans.

The performance of both controllers tested using a heel-drop excitation allows monitoring the

peak acceleration and settling time. The force of the heel-drop is modelled by a ramp with initial value of 2670 N and 50 ms of duration. The settling time is defined as the time taken for the response to fall and remain within some specified percentage of the maximum peak value of the acceleration response. For the sake of simplicity, here it is calculated as the time taken for the acceleration response to fall and remain within the range of 0.005 m/s^2 . The same nodes are chosen (excitation nodes are 82, 88, 327, 355 and 360, monitoring nodes are 333 and 90).

Figures 5.6 to 5.8 show the simulated time response of node 333 for no controlled case, the SISO AVC of Section 5.4.1 and the MIMO AVC of Section 5.4.1. Note that the vibration reduction level is practically the same for SISO and MIMO AVC at node 333 since the peak acceleration is 0.132 m/s^2 for SISO and 0.130 m/s^2 for MIMO, while the settling time is 1.985 s for SISO and 1.865 s for MIMO.

Figure 5.6: Simulated time response without AVC of node 333 for a normalized heel-drop excitation at nodes 82, 88, 327, 355 and 360.

Figures 5.9 to 5.11 show the simulated time response of node 90 for: no controlled case, the SISO AVC of Section 5.4.1 and the MIMO AVC of Section 5.4.1. Note that, unlike node 333, the vibration reduction level is significant if SISO and MIMO AVC are compared since the peak acceleration is 0.129 m/s^2 for SISO and 0.104 m/s^2 for MIMO, while the settling time is 4.625 s for SISO and 2.979 s for MIMO. Therefore, the MIMO AVC also improve substantially, like for jumping excitation example, the floor area defined by the node numbers between 1 and 175

Figure 5.7: Simulated time response with SISO AVC of node 333 for a normalized heel-drop excitation at nodes 82, 88, 327, 355 and 360.

Figure 5.8: Simulated time response with MIMO AVC of node 333 for a normalized heel-drop excitation at nodes 82, 88, 327, 355 and 360.

(see Figure 5.5).

For the jumping excitation, a pedestrian of weight 800 N was considered and the dynamic load factors (amplitude of each harmonic) of the excitation proposed in [18] were assumed (the periodic excitation is represented as a Fourier serie). Five nodes (82, 88, 327, 355 and 360) are chosen to excite with (see Figure 5.5) with these frequencies:

- i) 2.8 Hz - Node 355 - 1st vibration mode (5.6 Hz),

Figure 5.9: Simulated time response without AVC of node 90 for a normalized heel-drop excitation at nodes 82, 88, 327, 355 and 360.

Figure 5.10: Simulated time response with SISO AVC of node 90 for a normalized heel-drop excitation at nodes 82, 88, 327, 355 and 360.

- ii) 2.5 Hz - Node 360 - 2nd vibration mode (7.5 Hz),
- iii) 3.0 Hz - Node 88 - 3rd vibration mode (9.1 Hz),
- iv) 3.4 Hz - Node 82 - 4th vibration mode (10.2 Hz),
- v) 3.6 Hz - Node 327 - 5th vibration mode (10.7 Hz).

In order to illustrate the influence of the controllers, the serviceability of the floor for rhythmic activities [3] are evaluated. The VDV of all nodes on the floor during 60 seconds of jumping

Figure 5.11: Simulated time response with MIMO AVC of node 90 for a normalized heel-drop excitation at nodes 82, 88, 327, 355 and 360.

excitation is calculated for all nodes. Figure 5.12 shows the percentage of floor area where VDV is exceeded. It can be seen that MIMO achieves a significant improvement respect to SISO when the VDV is higher or equal to 0.02 m/s^{1.75}. This improvement is due to the fact that SISO cannot cancel the vibration of 3rd and 4th vibration modes. In addition, MIMO AVC can eliminate the vibration higher than 0.03 m/s^{1.75}. Therefore, it can be concluded that a MIMO AVC may be necessary when a SISO AVC cannot guarantee that the vibration level does not exceed a maximum.

Figure 5.12: Percentage of floor area where VDV is exceeded.

5.5 Conclusions

In this chapter, the recently proposed CRO-SL has been applied to a problem of AVC design for human-induced vibration mitigation in civil structures. The CRO-SL is a competitive co-evolution meta-heuristic algorithm in which different exploration procedures (i.e., different types of searching mechanisms) are jointly evolved within a single population of potential solutions to the problem. The CRO-SL algorithm gives the possibility of jointly exploring the performance of different meta-heuristics within a single population. In this case state-of-the-art operators in the CRO-SL have been included, such as two-points crossover mutations, Harmony Search, Differential Evolution or Gaussian mutations. In addition, a comparison with a recently proposed meta-heuristic for structures design have been included, the Enhanced Colliding Bodies Optimization algorithm. In terms of structural design, the main contribution of this chapter is to have the possibility of designing a MIMO AVC for complex floor structures with several closely frequency space vibration modes, where the number of TPs and sensor/actuator pairs is not a problem to obtain a global optimum solution in a affordable computation time. In addition, this chapter also shows that a MIMO AVC improves substantially the vibration reduction compared with a single-input and single output AVC for the proposed application example, which is a real complex floor structure. Future works will be focused on including new high-performance substrates in the CRO-SL, such as the ECBO compared, and also on the application of this work to experimentally reduce human-induced vibration in real complex floors by using inertial mass actuators.

Chapter 6

Designing and experimental implementation of active vibration control

The control design and the optimization problem proposed have been explained in Chapters 3 to 5; this chapter mainly focuses on the application of these methodologies experimentally.

6.1 Introduction

This section shows the MIMO AVC implementation on a real complex structure, a dining hall floor which is considered by its users as a lively structure that might exhibit excessive human-induced vibration. The complete process has been carried out to make a MIMO AVC implemented on this structure: from the initial stages related to structure identification to the active control implementation via LabVIEW and cRIO acquisition system of NI. Experimental results are shown to demonstrate the viability of MIMO AVC for a complex floor structure and its advantages compared with SISO AVC.

The steps to implement the AVC are also listed here. First of all, the floor structure and actuator models are identified. Based on these experimental tests, a state-space representation is obtained from an updated FE model of the floor structure, and the TFs of the actuators can be deduced (Section 2.5.1). Then, the optimal placement of the actuators and control parameters for the MIMO control are decided using the proposed common framework (Chapter 4). Thirdly, the MIMO control is performed using the LabVIEW Real-Time Module and the corresponding hardware that has been described in Section 4.4.6.

6.2 Dining hall floor

6.2.1 Floor structure description

In this section, the in-service open plan dining hall floor of a primary-secondary school sited in Pozuelo (Madrid, Spain) has been investigated (see Figure 6.1). This structure has a dimension of 25.5 m $\times$ 20 m and is simulated through a preliminary FE model described in Section 5.2. The plan of the floor is shown in Figure 6.2, which corresponds to the top floor of a three-storey building (basement, ground floor and first floor). The floor considered is highlighted in Figure 6.2. Three sides are external walls, with large windows separated with columns and load bearing walls. In the other side there are more rooms and some columns that must be taken into account. The floor can be seen as a three-bay structure with underneath partitions that mainly influence the damping [3].

Figure 6.1: Panoramic picture of the dinning hall.

The floor is daily used during lunch time (Figure 6.3) and also sometimes to carry out different

(a) First floor (dining hall)

(b) Ground floor

Figure 6.2: Architectural drawing of the dining hall.

events. The floor is considered by its occupants to be quite lively, but not sufficiently lively to attract complaints. Special attention was paid to the bay near the main entrance (Bay 1) because it was perceived to be more lively than the other two bays.

Figure 6.3: Students having lunch in the dining hall, near the kitchen.

This section follows with a description of the FE model of the structure, the OMA and EMA carried out from on-site measures, the model calibration, the MIMO AVC designing and the implementation.

The tests carried out have been undertaken in two campaigns:

- Campaign 1: Several tests were done in order to identify the dynamics of the floor. The results have been used to calibrate the FE model and to design optimal positions of the actuators for MIMO AVC.
- Campaign 2: Control tests were performed for the positions of actuators decided from campaign 1.

6.2.2 FE model description

The floor model is sketched in Figure 6.4. The columns and load bearing walls are considered as boundary conditions. Most of the information has been taken from architectural drawings (Figure 6.2) and visual inspection of the structural elements whenever they are accessible. However, there are some geometrical uncertainties, as the floor slab thickness, that are difficult to determine, as well as the material properties and the stiffness introduced by the underneath partitions.

Figure 6.4: FE model with boundary conditions.

The model has been created in ANSYS considering shell elements (SHELL 63) for the slab. Columns are introduced as a boundary condition constrained in the vertical direction (Z), rotations are allowed and displacement in X or Y direction is allowed according to its corresponding edge of the floor. Load bearing walls are also considered as boundary conditions with

Table 6.1: Description of the tests of the first campaign

Test	Setup	Excitation
1	1	Actuator at TP3 random signal, 2Hz-20Hz, 10min
2	1	Actuator at TP3 random signal, 4Hz-15Hz, 10min
3	1	No signal, 10min
4	1	No signal, All night
5	1	Actuator at TP3 random signal, 0Hz-30Hz, 20min
6	1	Actuator at TP3 chirp signal, 5Hz-12Hz, 10min
7	1	Heeldrops and jumpings at TP3 or TP18, 5min
8	2	Actuator at TP6 random signal, 0Hz-30Hz, 20min
9	2	Actuator at TP3 random signal, 0Hz-30Hz, 20min
10	3	Actuator at TP3 chirp signal, 20min
11	3	Actuator at TP6 chirp signal, 20min

the movement constrained in vertical and in the direction parallel to the maximum length of the wall.

6.2.3 Experimental analysis and model calibration

In order to characterize the dynamic behaviour of the structure, some measurements were planned and done on the structure. These results are used to calibrate the FE model. Three setups with 18 accelerometers have been planned, placed at the TPs shown in Figure 6.5. Note that in setups 2 and setup 3, the roving accelerometers corresponds to TPs numbered as 2xx and 3xx, respectively. Also an electrodynamic shaker has been used in some of the experiments. The actuator acceleration signal has been measured together with the floor accelerations. Data have been acquired using a NI cRIO quasi-distributed equipment and high sensitivity (approximately 10 V/g) PCB accelerometers measuring in the vertical direction. A total of 11 tests were carried out and are summarized in Table 6.1. As shown in Figure 6.5, the TPs in setup 1 cover all the three bays, considering the dynamics of the whole structure generally, and setups 2 and 3 mainly focus on bays 2 and 3, respectively.

Tests 1, 2, 5, 8 and 9 were carried out with the actuator mass moving randomly. Thus, since the input signal is measured and is a white noise, these tests can be used for both EMA and

(a) setup 1

(b) setup 2

(c) setup 3

Figure 6.5: TPs configuration in three setups.

OMA. In tests 3 and 4, there are not any measured input, the floor is only subjected to ambient noise, so that they can be used for OMA. Tests 6, 10 and 11 were carried out driving a chirp signal, so that they might be used for FRF estimations.

The modal identification has been done with the Stochastic Subspace Identification method [89], and the identified modes (mainly bay 1 of the floor) are shown in Figure 6.6.

The use of one actuator in some of the tests with a moving mass of 30.6 kg and accelerometer attached on it allows to obtain the input force. Therefore, a FRF-based analysis can be done and the modal mass associated to each mode can also be estimated. This extra information is very helpful in the calibration process of the FE model. Figure 6.7 shows the FRF obtained in test 6.

The measured force can be also used as an input for a transient analysis on the FE model to compare the resulting time histories at the same locations measured in the real tests. The same force can be used in a harmonic analysis of the FE model.

The FE model is calibrated using the same process as explained in Section 4.4.1. The material properties adopted are: modulus of elasticity $E = 27 \times 10^9 \text{ N/m}^2$, Poisson's ratio $\nu = 0.15$ and density $\rho = 3000 \text{ kg/m}^3$; the thickness of the floor slab is 0.33 m. The calibrated vibration modes are shown in Figure 6.8, which are in good correspondence with experimental results (see Figure 6.6) from both frequencies and mode shapes.

6.3 AVC results

6.3.1 Designing based on in-situ tests

As shown in the OMA results and the FE model mode shapes (Figure 6.8), the human-induced vibration problem focuses mainly on bay 1 (the part near the main entrance), which are separated from the other part by two lines of columns (Figure 6.4).

(a) $f = 6.63\text{Hz}$, $\xi = 2.50\%$. (b) $f = 7.44\text{Hz}$, $\xi = 2.20\%$. (c) $f = 10.12\text{Hz}$, $\xi = 1.58\%$.

Figure 6.6: Identified vibration modes of bay 1 from Test 6.

Figure 6.7: FRF of TP3 (magnitude in dB referred to 1 m/Ns²).

The optimal placement of the two actuators are obtained from campaign 1, which can control mainly the vibration of the first three modes. TP A and TP B, shown in Figure 6.9.

The MIMO AVC uses a decentralized IRC (as used in Section 4.4.3) and the controller design is carried out by estimating the FRFs shown in Figure 6.10, which is obtained using actuator forces of random signal (2 Hz-15 Hz), resulting the following controller:

$$\mathbf{K}_{\text{AMD}}(s) = \begin{bmatrix} H_{A_{11}}(s) & 0 \\ 0 & H_{A_{22}}(s) \end{bmatrix} = \begin{bmatrix} -4000/(s + 0.1) & 0 \\ 0 & -3000/(s + 0.1) \end{bmatrix}. \quad (6.1)$$

The parameters have been tuned to guarantee a good damping performance and a sufficient stability margin. Moreover, a Butterworth low-pass filter with cut-off frequency of 30 Hz (see Section 4.2.2) is used together with $\mathbf{K}_{\text{AMD}}(s)$. Note that this filter can avoid spillover effects in practical applications [37].

6.3.2 Implementations and response tests

Figure 6.11 shows the MIMO AVC implemented, using the same actuators and devices as in Section 4.4. The control performance are evaluated under HIL, and the time history responses

(a) 6.64 Hz

(b) 7.52 Hz

(c) 9.98 Hz

Figure 6.8: Vibration modes in calibrated ANSYS FE model.

Figure 6.9: TPs for MIMO AVC implementation of the dining hall floor.

Figure 6.10: Magnitude (in dB) plot of the measured data, 1-TP A and 2-TP B.

of two people (67 kg and 75 kg) jumping (see Figure 6.12) near TP A at 180 bpm (beats per minute) are shown in Figure 6.13. Note that this periodical excitation can excite the first vibration mode with the first harmonic of the excitation. From the peak R-factor and VDV in Figure 6.13, the MIMO has a better control performance for TP A than TP B.

Figure 6.11: Two-actuator MIMO AVC implementation of the dining hall floor.

Figure 6.12: Response test of MIMO AVC under human jumping excitation.

(a)

(b)

(c)

(d)

Figure 6.13: Time history response of TPs A and B under human jumping excitation: (a) TP A without control; (b) TP A with MIMO AVC; (c) TP B without control; (d) TP B with MIMO AVC.

6.4 Conclusions

This section presents an application of MIMO AVC using two actuators for human-induced vibration problems, with a decentralized IRC and low-pass filter included.

In the implementation example, the contribution is illustrated by experimental results. The time responses under human jumping forces shows the viability of MIMO AVC to control human-induced vibration coming from a lively floor.

Chapter 7

Conclusions, publications and future works

7.1 Conclusions

The problem of cancelling human-induced vibrations in “lively” structures has been dealt with throughout this thesis. Starting from a state-of-the-art of the SISO controller design, it has been demonstrated that all these developments can be interpreted as band-pass filters that, in a way, carry out velocity feedback. Thus an interesting non-dimensional study taking reasonable assumptions, such as modelling the structure by its fundamental mode or taking second-order model for the actuator dynamics has been carried out. Simple design rules have been derived in order to get the controller implementable in a way. All these rules for the implementation have been summarized in a practical design guidelines in Chapter 3, which is validated by experimental implementation on a full-scale laboratory structure.

When the vibration cancellation has to be achieved for a complex floor with plenty of vibration modes spatially coupled and with similar natural frequencies, SISO controllers might not work properly. Several independent SISO controllers cannot be applied since the control action at

a particular controlled point is considered as a perturbation to another controlled point and the control action might be destroyed. Hence, MIMO strategies should be used if one wants to cope with these dynamically complicated floors. Thus in Chapter 4, it has been developed a novel design methodology which merges the design of active and passive inertial dampers in a common framework in the context of human-induced vibrations. That is, the methodology considers both, the nature of the excitation (DLFs are different at different frequencies) and how humans perceive vibration (depending on the frequency).

It is well known that the force transmitted by a TMD can be interpreted as a feedback control force. Hence, both strategies can be designed under the same framework. A PI is minimized in order to obtain the best location for a number of dampers and the design parameters. In case of passive devices, the tuning parameters (frequencies and damping ratios) play the role of controllers to be designed.

The PI proposed in Chapter 4 involves minimizing a non-linear function. Thus, the computational burden of this optimization increases drastically with the number of possible locations. Firstly, a simple and straightforward method for selecting a number of good candidates is proposed in Chapter 4. That is, the linear problem is considered in order to reduce drastically the possible locations and the optimization algorithm can be used with an affordable computational cost. Interestingly, the recently developed modern co-evolution meta-heuristic, CRO-SL optimization method, has been applied to solve the optimization problem in Chapter 5. It has been shown that a global optimum solution can be achieved within an affordable computational time.

Finally, apart from several simulated examples, two experimental applications of MIMO control has been presented: AVC of a full-scale steel-concrete composite floor built in the laboratory (Chapter 4) and AVC of a complex in-service floor used as dining hall (Chapter 6). All the results obtained has been supports the contributions of this thesis. This thesis has contributed to the development of vibration control strategies for human-induced vibration and will be foundation for future researches in which the development of actuator with suitable properties

for this application should be tackled with.

7.2 Publications

It is worthy here to mention that two papers have been already published and three are under review or going to be submitted. All of them will be published in high-quality journals, assuming thus the contributions of the thesis are properly divulged. Additionally several interesting conference papers have contributed to impact of the thesis.

7.2.1 Journal papers

- C. Camacho-Gómez, **X. Wang**, E. Pereira*, I.M. Díaz, S. Salcedo-Sanz. Active vibration control design using the Coral Reefs Optimization with Substrate Layer algorithm. *Engineering Structures*, 157: 14-26, 2018.
- **X. Wang***, E. Pereira, I.M. Díaz, J.H. García-Palacios. Velocity feedback for controlling vertical vibrations of pedestrian-bridge crossing. Practical guidelines. *Smart Structures and Systems*, 22(1): 95-103, 2018.
- E. Pereira*, **X. Wang**, I.M. Díaz. A common framework for the design of active and passive inertial mass dampers for human-induced vibrations. *Structural Control and Health Monitoring*, 2018. (Under review)
- **X. Wang**, I.M. Díaz, S.S. Aphale, E. Pereira*. Stability approach to implement Integral Resonant Control in active vibration control of pedestrian structures. *IEEE-ASME Transaction on Mechatronics*, 2018. (Under Review)
- **X. Wang**, I.M. Díaz, J.H. García-Palacios, E. Pereira*. Implementing a multi-input multi-output active control for an in-service dining hall floor. *Vibration*, 2018. (Manuscript preparation)

7.2.2 Conference proceedings

- **X. Wang**, I.M. Díaz, E. Pereira, Q. Xu. Optimal TMD-based control strategy for human-induced vibrations. *3rd International Conference on Mechanical Models in Structural Engineering*, Seville, Spain: 24-26 June 2015.
- **X. Wang**, I.M. Díaz, E. Pereira, MIMO control design including input-output frequency weighting for human-induced vibrations. *6th European Conference on Structural Control*, Sheffield, UK: 11-13 July 2016.
- **X. Wang**, E. Pereira, I.M. Díaz, J.H. García-Palacios. Step-by-step guide for MIMO active vibration control: from the design to the implementation. *1st Conference on Structural Dynamics*, Madrid, Spain: 20-21 June 2018.
- E. Pereira, **X. Wang**, C. Alén, J. García-Palacios, I.M. Díaz. Vibration control of a dining hall: MIMO and SISO design. *8th ECCOMAS Thematic Conference on Smart Structures and Materials*, Madrid, Spain: 5-8 June 2017.
- J.H. García-Palacios, I.M. Díaz, **X. Wang**, J.C. Déniz, J.M. Soria, C. Martín de la Concha. Dynamic analysis of a lively dinning hall floor: testing and modelling. *1st Conference on Structural Dynamics*, Madrid, Spain: 20-21 June 2018.
- C.A. Barrera, J.M. Soria, **X. Wang**, J.H. García-Palacios, I.M. Díaz. On the search of TMD configurations for vibration mode with changing modal parameters. *1st Conference on Structural Dynamics*, Madrid, Spain: 20-21 June 2018.
- I.M. Díaz, E. Pereira, **X. Wang**, J.H. García-Palacios. Active vibration control of human-induced vibrations: from SISO to MIMO. *1st conference on Structural Dynamics*, Madrid, Spain: 20-21 June 2018.
- J.M. Soria, I.M. Díaz, J.H. García-Palacios, C. Zanuy, **X. Wang**. Tuning a phase-controlled smart TMD for broad-band-frequency-varying vibration modes. *1st Conference on Structural Dynamics*, Madrid, Spain: 20-21 June 2018.

- J.M. Soria, I.M. Díaz, E. Pereira, J.H. García-Palacios, **X. Wang**. Exploring vibration control strategies for a footbridge with time-varying modal parameters. *Journal of Physics: Conference Series*, 744: 012170, 2016. *13th International Conference on Motion and Vibration Control*, Southampton, UK: 3-6 July 2016.

7.3 Future works

The application examples of this thesis have been carried out on two real structures (a full-scale steel-concrete composite floor built in the laboratory and a complex in-service floor used as dining hall). The problems associated to these structures mainly motivate the future works.

The laboratory structure is a very light-weight floor, whose dynamics change as the shaker and/or occupant moves on the structure. Thus, an LTI model cannot be considered to optimally design a MIMO controller based on a PI. Therefore, a structure model, which can be updated with the location of actuator and/or occupant must be obtained. In addition, an optimally designed robust controller, which considers the dynamic changes, must be also proposed. The use of this model can be a good benchmark to test novel global optimization algorithms like CRO.

Though it has been demonstrated that the technologies studied in the thesis can be applied to the dining hall floor, its dynamics should be studied more deeply so that the FE model can be better calibrated. Therefore, the model obtained in Chapter 6 may be improved before designing an optimal MIMO AVC considering the methodology proposed in Chapter 4 and using optimization algorithm like CRO-SL, utilized in Chapter 5. These results will be submitted as a journal paper.

Besides these contributions above, the following future works will also be carried out:

- The experimental work of MIMO AVC is carried out in a dining hall floor, whose performance has been evaluated by some human-induced activities. In the future, a long-term

serviceability monitoring and evaluation of the control performance might be investigated.

- It is interesting to design a PVC system for this structure, and compare with the AVC system. Moreover, some hybrid control systems might be designed for this structure; because there is one important vibration mode spatially distributed from the others, for which a single TMD might work well. Thus, a control system consisting of a TMD and two actuators is proposed.

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