Doctoral Thesis

**Dynamic analysis and control of broadband
frequency harmonic vibrations in lightweight
pedestrian structures**

José Manuel Soria Herrera

Ingeniero de Caminos, Canales y Puertos

Madrid, 2019

UNIVERSIDAD POLITÉCNICA DE MADRID

Escuela Técnica Superior de Ingenieros de
Caminos, Canales y Puertos

**Dynamic analysis and control of broadband
frequency harmonic vibrations in lightweight
pedestrian structures**

by

MEng. Civil Eng. **José Manuel Soria Herrera**

A dissertation submitted in partial fulfillment for the
degree of Doctor of Philosophy

in the

Department of Continuum Mechanics and Theory of Structures

under the supervision of

Dr. Iván Muñoz Díaz

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Madrid, 2019

How to cite this work:

J.M. Soria (2019). *Dynamic analysis and control of broadband frequency harmonic vibrations in lightweight pedestrian structures* (Doctoral thesis). ETSICCP, Universidad Politécnica de Madrid, Spain.

Grupo
Ingeniería
Estructural

The work in this thesis was carried out in the Structural Engineering Group (GIE), consolidated research group of the UPM.
<http://ingstruct.mecanica.upm.es>

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Declaration of Authorship

The author declares that this dissertation is the result of his own research work, specifically performed to get the candidature submitted for the degree of Doctor of Philosophy in Civil Engineering at Universidad Politécnica de Madrid (UPM). The thesis is based entirely on the independent work carried out by the author in UPM between May 2015 and September 2019 under the supervision of Dr Iván M. Díaz and Dr Jaime H. García-Palacios.

Published work of others is always clearly attributed with the corresponding sources given. All the work and ideas recorded are original except for what is referenced in the text. No part of this has been previously submitted for a degree or any other qualification at any university or institution.

A handwritten signature in blue ink. It starts with the initials 'JM' inside a large, loopy circle. To the right of the circle, the name 'Soria' is written in a cursive script, followed by a double vertical line 'H'.

Signed: José Manuel Soria Herrera

Date: 23/09/2019

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Abstract

Lightweight and/or long-span pedestrian structures are usually prone to vibrate excessively. Although current codes may be fulfilled, these structures are not usually comfortable in any way. Control devices can mitigate vibration and significantly improve the comfort as well as increasing the structure's life span. Damping systems for civil structures have been continuously proposed; however, the civil engineering community has not generally accepted these damping systems, although they have shown great potential for cancelling vibrations. This thesis has been carried out within the Public Research Project "*Development of novel systems for reducing vibrations in pedestrian structures*" REVES-P (DPI2013-47441). Thus, this research provides tools for the dynamic analysis of structures with time-varying modal parameters and for addressing the vibration control of broad-band-frequency harmonic vibrations produced by human-induced excitations.

In this thesis, two main subjects have been dealt with. The first one is related to the long-term monitoring and dynamic analysis of an in-service steel-plated stress-ribbon footbridge. A methodology for tracking the vibration modes is developed and the modal parameters are correlated against external factors. It has been shown that this type of structure can be highly sensitive to temperature variations (frequency changes of more than 20 %) with highly seasonal and daily trends. These correlations enable the influence of these uncertainties on modal estimates to be removed, thus facilitating their use as possible damage-sensitive features. Additionally, interesting long-term vibration serviceability predictors are derived and assessed according to current codes.

The second subject is related to semi-active vibration control. When structures show modal properties changing over time, and/or several vibration modes must be cancelled by the same device, passive devices (tuned mass dampers, TMD, are the device usually adopted for footbridges) may detune and thus experience a significant loss of efficiency. Under these circumstances, the use of semi-active devices may be the most promising alternative. This research concentrates on the upgrading of classical TMDs to be totally adaptable to the actual characteristics of the external action or significant changes to the structure's modal properties (so as to be robust to the device changes). As a result, a methodology to numerically test and design passive and semi-active strategies for broad-frequency band vibrations is proposed. The semi-active action is controlled using magneto-rheological (MR) dampers. In this sense, experimental and simulated studies have been carried out to optimize the tuning of semi-active strategies taking into account the existing non-linearities, including an MR model. Existing control laws, based on tuning the TMD phase in real-time, have been adapted in order to make them usable experimentally. Finally, experimental results have shown the potentiality of the proposed semi-active control design methodology.

Resumen

Las estructuras peatonales esbeltas y/o de grandes vanos suelen ser propensas a vibrar en exceso. Aunque las normativas actuales se cumplan, estas estructuras pueden resultar incómodas. Los dispositivos de control pueden reducir las vibraciones y mejorar significativamente el nivel de confort, así como aumentar la vida útil de las estructuras. Se han propuesto numerosos sistemas de amortiguación para estructuras civiles; sin embargo, en este campo no se han adoptado estos sistemas de amortiguación de forma general, aunque éstos son muy eficaces en la cancelación de vibraciones. Esta tesis se ha llevado a cabo dentro del Proyecto de Investigación de financiación Pública “*Desarrollo de nuevos sistemas para reducir las vibraciones en estructuras peatonales*” REVES-P (DPI2013-47441). Así, en esta tesis se proporcionan herramientas para el análisis dinámico de estructuras con parámetros modales que varían en el tiempo y que permiten abordar el control vibraciones producidas por peatones en un amplio espectro en frecuencias.

Esta tesis tiene dos objetivos principales. El primero se relaciona con la monitorización permanente y el análisis dinámico de una pasarela de banda tesa en servicio. Se desarrolla una metodología para el seguimiento de los modos de vibración y la correlación de los parámetros modales con factores externos. Se ha demostrado que la estructura es muy sensible a las variaciones de temperatura (cambios en la frecuencia de más del 20 %) con dependencia estacional y diaria. Estas correlaciones permiten eliminar gran parte de la influencia de estas incertidumbres en las estimaciones modales, facilitando así su posible uso como índice sensible al daño estructural. Además, se han obtenido predictores representativos del servicio de vibraciones en monitorizaciones continuas y se han evaluado éstos de acuerdo con los códigos actuales.

El segundo objetivo está relacionado con el control de vibraciones semi-activo. Cuando se tienen estructuras cuyas propiedades modales cambian con el tiempo y/o se quieren cancelar varios modos de vibración con el mismo dispositivo de control, los sistemas pasivos (amortiguadores de masa sintonizados, TMD, por sus siglas en inglés, habitualmente adoptados para puentes peatonales) se pueden desintonizar, experimentando una pérdida significativa de eficiencia. En estas circunstancias, el uso de dispositivos semi-activos puede ser una solución alternativa prometedora. Esta investigación se centra en la mejora de los TMD clásicos para que sean totalmente adaptables a las acciones externas y/o a cambios significativos en las propiedades modales de la estructura (e incluso a cambios del propio dispositivo). Como resultado, se propone una metodología para el análisis numérico y el diseño de estrategias pasivas y semi-activas para el control de vibraciones en una amplia banda de frecuencias. La acción semi-activa se controla mediante amortiguadores magneto-reológicos (MR). En este sentido, se han llevado a cabo estudios experimentales y simulaciones numéricas para la optimización del sintonizado de diferentes estrategias de control semi-activas teniendo en cuenta las no linealidades existentes, incluido un modelo del amortiguador MR, así como la evaluación de su funcionamiento. Las leyes de control existentes, basadas en el

ajuste de la fase del TMD en tiempo real, se han adaptado/mejorado para hacerlas implementables. Finalmente, a partir de los resultados experimentales se ha demostrado el potencial de la metodología propuesta para el diseño e implementación de un sistema de control semi-activo.

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José Manuel Soria Herrera is Assistant Professor in the Department of Continuum Mechanics and Theory of Structures of the ETSI Caminos, Universidad Politécnica de Madrid (UPM). He teaches courses related to structural dynamics and experimental analysis of structures. Since 2015 he has been a member of the Structural Engineering Group (GIE), a consolidated research group of the UPM. He has also taught several training courses of $\text{\LaTeX}$ and Python for academics through the Instituto de Ciencias de la Educación (ICE).

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He has been involved in research projects supported by public financing and in the publication of several papers in JCR indexed journals. Moreover, he carries out continuously reports and research projects funded by private companies. He is a member of the Spanish Association of Structural Dynamics (AEDE).

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Agradecimientos

Llegado el final de esta intensa etapa, quisiera agradecer a todos aquellos que, de una forma u otra, han contribuido al desarrollo de esta Tesis Doctoral.

Resulta casi imposible expresar en unas pocas líneas el profundo agradecimiento que siento hacia Iván Muñoz Díaz como director de esta Tesis Doctoral. Han sido muchísimas las horas que me ha dedicado, demostrándome estar profundamente comprometido conmigo y haciendo de este trabajo, un trabajo más calidad. Además, me ha demostrado tener una fuerza excepcional en algunos momentos verdaderamente difíciles que le ha tocado superar en estos años, es por ello que a día de hoy es un referente para mí, tanto en lo profesional como investigador y docente como en lo personal. Para mí ha sido todo un privilegio que haya sido director de este trabajo. En segundo lugar, quisiera agradecer a Jaime García Palacios que confiase en mí para aquella oportunidad laboral que me inició profesionalmente en el mundo de la investigación. Su orientación y consejos, tanto en el ámbito de la Tesis como fuera de éste han sido y siguen siendo un pilar fundamental en mi desarrollo profesional. Tanto a él como a su mujer Margarita, gracias por el apoyo y los ánimos que siempre me han dado.

Por otra parte, quisiera dar mi más sincero agradecimiento a Antolín Lorenzana por las muchas lecciones que me ha dado, así como las conversaciones en las que me ha hecho pensar para mejorar este trabajo, sin perder nunca la sonrisa y el buen humor. También quiero agradecer a su pupilo Álvaro Magdaleno toda la ayuda que me ha brindado sin pedir nunca nada a cambio, sé que tengo en él un amigo para siempre, espero que sepa que es algo mutuo. Estoy seguro de que nos espera un futuro lleno de colaboraciones.

A Emiliano Pereira y Javier Cara, agradecerles lo mucho que he aprendido de ellos y el haber estado dispuestos a ayudarme en todo momento. A la gente de la ETS de Ingenieros de Telecomunicaciones de la UPM (Francisco Tirado, Guillermo Jara, Álvaro Araujo y Octavio Nieto) todos los buenos ratos que hemos pasado juntos tanto en su Laboratorio como en los ensayos experimentales realizados.

Quisiera agradecer a todos los profesores con los que he tenido el placer de colaborar, especialmente a Carlos Zanuy, Rafael Fernández, José María Goicolea, Khanh Nguyen, Javier León, Leonardo Todisco y Javier Jiménez, por lo bien que me han orientado siempre. Agradecer también a Luis Plaza y Antonio Madrid esos exámenes de más que han tenido que corregir para darme cancha con la redacción de este trabajo. Agradecer también al Departamento de Mecánica de Medios Continuos y Teoría de Estructuras el apoyo recibido.

No puedo olvidarme de los Maestros de Laboratorio, Isidro García y Miguel Ángel Peña, agradecerles su ayuda en las configuraciones de todos los ensayos experimentales realizados en esta tesis, derrochando profesionalidad y buen humor. Agradecer a Julia Chamorro sus numerosas infusiones que han hecho más llevaderas

aquellas duras tardes de trabajo, así como a Beatriz Gutierrez sus numerosos detalles a lo largo de estos años. Quiero dar las gracias también a mis compañeros del Laboratorio de Estructuras por su apoyo: Carlos de la Concha, Xidong Wang, Gonzalo Sanz-Diez, Cristian Barrera, Carlos Iturregui, Carlos Velarde, y los recién incorporados Rafael Ruíz y Mar Corral. Mención especial para Carlos de la Concha que ha sido un amigo en el que he podido apoyarme en esta recta final cuando mis ánimos estaban más decaídos.

También quiero agradecer al equipo de Pondio Ingenieros la confianza que han depositado en mí siempre, empezando por Juan Calvo, Joaquín Arroyo, Lucía López y José Vicente Martínez, y siguiendo con el resto de compañeros. Han sido muchos los ratos que he pasado junto a ellos y ha sido un placer formar parte de esa gran familia.

Por otra parte, no sólo durante la realización de esta Tesis sino también a lo largo de toda mi vida, he contado con el apoyo incondicional de mis padres. Siempre han sido un ejemplo para mí. Sin ellos no me habría sido posible llegar hasta aquí ni ser la persona que hoy soy. Gracias.

Por último, quiero agradecer a Mamen, no sé que sería de mí sin ella. Han sido muchas las horas de trabajo que ha aguantado y sin embargo ha estado a mi lado apoyándome en todo momento. Incluso en mis días de estrés y mal humor, ha sabido tranquilizarme y darme la serenidad que necesitaba. En breve me hará el mejor regalo que se me puede hacer, una hija, Clara. Gracias a ellas, la etapa final de esta Tesis no sólo ha sido un camino más fácil, sino un paseo para recordar.

Gracias a todos,

José Manuel Soria

El autor también quiere agradecer el apoyo económico proporcionado por el Ministerio de Economía y Competitividad (Gobierno de España) para la financiación del Proyecto de Investigación REVES-P (DPI2013-47441) y la beca FPI disfrutada.

*“If you want to find the secrets of the universe,
think in terms of energy, frequency and vibration.”*

Nikola Tesla (1856 – 1943)

To my wife Mamen and
my daughter Clara.

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Abbreviations

AMD	A ctive M ass D amper
AVC	A ctive V ibration C ontrol
BI	B ase I solation
CDF	C umulative D istribution F unction
DBG	D isplacement B ased G roundhook
DLF	D ynamic L oading F actor
DOF	D egree O f F reedom
DVFC	D irect V elocity F eedback C ontrol
EM	E xpectation M aximization
FEM	F inite E lement M odel
FFT	F ast F ourier T ransform
FRF	F requency R esponse F unction
HTMD	H ybrid T uned M ass D amper
HVC	H ybrid V ibration C ontrol
KKF	K inematic K alman F ilter
LIF	L eaky- I ntegrator F ilter
LQR	L inear Q uadratic R egulator
LVDT	L inear V ariable D ifferential T ransducer
MAC	M odal A ssurance C riterion
MDOF	M ulti- D egree O f F reedom
MEMS	M icro E lectro- M echanical S ystem
MLR	M ultiple L inear R egression
MR	M agneto- R heological
MTVV	M aximum T ransient V ibration V alue
NI	N ational I nstruments
OMA	O perational M odal A nalysis
PED	P assive E nergy D issipation

PGB	P edro G ómez B osque
PI	P erformance I ndex
PID	P roportional I ntegral D erivative
PSD	P ower S pectral D ensity
PVC	P assive V ibration C ontrol
RMS	R oot M ean S quare
SDOF	S ingle D egree O f F reedom
SHM	S tructural H ealth M onitoring
SLS	S erviceability L imit S tate
SSI	S tochastic S ubspace I dentification
STMD	S emi-active T uned M ass D amper
TF	T ransfer F unction
TMD	T uned M ass D amper
TVA	T uned V ibration A bsorber
ULS	U ltimate L imit S tate
VBG	V elocity B ased G roundhook
VDV	V ibration D ose V alue
VS	V ibration S erviceability
VSA	V ibration S erviceability A ssessment
VSLS	V ibration S erviceability L imit S tate

1

Introduction and objectives

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1.1 Introduction

This Thesis is part of the Research Project “*Development of novel systems for reducing vibrations in pedestrian structures*” REVES-P (DPI2013-47441) funded by the Ministry of Economy and Competitiveness. The said Project was devoted to the development of semi-active and active vibration control strategies and to contribute to the acceptance of these advanced vibration control techniques by the civil engineering community. The PhD candidate got a 4-year research grant within the aforementioned project to carry out this thesis under the supervision of Dr Iván M. Díaz.

1.1.1 Vibration serviceability problems

The continuous development of structural materials and construction techniques are leading to more slender and thus lighter structures, which may be susceptible to human-induced vibrations, compromising the Vibration Serviceability Limit State (VSLS).

Therefore, it is increasingly common to find slender pedestrian structures with vibration and/or comfort problems for their users. These structures interact with human movements such as walking and running and this sometimes produces undesirable effects, which happen particularly when humans' excitation frequencies or their harmonics are close to a structural natural frequency. The natural frequency is that at which the system (structure) would oscillate when vibrating freely after the application of a load. If the damping ratio is low, the resonance phenomenon may occur. [Figure 1.1](#) shows a structure treated as a system, which is very common in Control Engineering. The system has an input or disturbance that is the excitation or load and an output, for example, its acceleration response.

Figure 1.1: Structure treated as an input-output system.

Excessive vibration has constituted a common problem in engineering history [17]. The feeling of fear and discomfort of the structure's occupants due to high levels of vibration is an important consideration for VSLS and safety. The resonance increases the levels of vibration and results in the structures not being able to fulfil the VSLS. Among the strategies to solve this problem, such as stiffening the structure, changing the shape and/or adding mass, the inclusion of damping systems or vibration absorbers can also be found. [Figure 1.2](#) shows a structure with a Passive Vibration Control (PVC) device.

Figure 1.2: Passive controlled structure with a passive control device.

Figure 1.3: Structure with a TVA.

The PVC device coupled to the structure makes a new system formed by “structure + control device” with more damping capacity and improved comfort. If the PVC is an inertial mass control system, such as a Tuned Vibration Absorber (TVA), the controlled system of Figure 1.2 can be represented as in Figure 1.3, in which the TVA is treated as a feedback of the structure. The structure applies forces to the TVA and the TVA reacts by applying forces to the structure. An example of TVA installation in which one of the thesis supervisors participated, together with the Cartif Technological Centre, is the one installed in the Balear Stadium Footbridge (Mallorca, Spain). A TVA was designed and installed to enhance the dynamics response of the bridge and meet current VSLs recommendations. This footbridge had a provisional post-support installed between both carriageways of the highway over-crossed by the bridge (see Figure 1.4a). This support was removed once the TVA was installed¹ (see Figure 1.4b). TVA is also known in the literature as Tuned Dynamic Absorbers and Tuned Mass Damper (TMD). When referring to TVA in civil structures, the term TMD is the most common one adopted, so this is the one adopted in this thesis.

The incorporation of the vibration control device to the structure increases its damping capacity, i.e., the structure comfort is also increased, as it happened in the case of the Balear Stadium Footbridge after installing the aforementioned TMD. However, the modal properties of structures might change over time due to external factors (typically temperature, support conditions, level of occupancy, amongst others) or simply ageing. Lightweight structures are especially sensitive to such factors and/or their dynamic responses may come from several significant vibration modes. Under these circumstances, passive devices may not behave as expected, showing a clear loss of efficacy. Thus, it might be necessary to install more than one PVC or to enable the control system to adapt to the system’s variations. Figure 1.5 shows the scheme of a semi-active control system capable of changing its properties in real-time, resulting in a semi-active controlled structure, as shown in Figure 1.6.

¹The TVA should be fitted into the structure depending on its type. The best-fitting can be achieved when the TVA is included from the structure design stage.

(a) Before installation of TMD. Provisional support to avoid excessive vibrations, in red.

(b) TMD installed on the structure. TMD in green.

Figure 1.4: Example of a practical application of a TMD in an in-service structure.

This thesis focuses on semi-active inertial mass control systems. The semi-active vibration control systems allow increased robustness and efficiency of the passive systems against the aforementioned factors and are able to cope with several vibration modes simultaneously.

1.1.2 Motivation

A paradigmatic case of structures with excessive vibrations induced by pedestrian loading is long-spanned and/or lightweight pedestrian bridges, also known as footbridges. Furthermore, in some of these structures, depending on the structural type and/or external factors, more than one vibration mode may be excited and/or the modal properties of these vibration modes might change significantly over time. A footbridge located in Porto (Portugal) is presented here as an introductory case. The structure connects a leisure centre, known as the Transparent Building, with one of the main parks of the city (see [Figure 1.7](#)). This is a continuous beam-type bridge with two spans of proximately 30 m each and the external

Figure 1.5: Structure with a semi-active control device treated as a feedback system. Red symbol ($\nearrow$) means changing over time.

Figure 1.6: Semi-active controlled structure with semi-active control device. Red symbol ($\nearrow$) means changing over time.

ends behave as simply supports. The deck is 3.5 m wide and has an inclination of 6 %, starting with a lower elevation at the park side and ending with a higher level at the Transparent Building. The cross-section is composed of two main steel girders, type (IPE600), together with secondary steel beams to support a wooden flooring. The total mass of the structure, including all structural and non-structural elements, was estimated to be 380 kg/m [139].

Given the symmetry of the structure, the evaluation of the maximum vibration levels can be performed at mid-span in one of the two spans on any side of the deck. Accordingly, a measure of the vertical accelerations experienced in the middle of the first span is analysed. Figure 1.8a shows the time history acceleration for a recording of almost one hour at this point. The maximum absolute value is 1.15 m/s^2 and, as will be seen in Chapter 2, this acceleration value corresponds to a minimum comfort class. Figure 1.8b shows the frequency spectrum of this test. It can be intuited that there are three representative vibration modes in the structure's response. The first two modes are in a frequency band that is the most excited by walking pedestrians. Additionally, the third mode is in a band that might be excited by pedestrians running. Table 1.1 shows the mean values of the estimation of the modal parameters for these first three vibration modes.

Usually, the dynamic response of the structure is studied from a Finite Element Model (FEM). Although a detailed FEM could be carried out at the design stage,

(a) General view.

(b) Bottom view.

(c) Top view.

Figure 1.7: Footbridge, Porto (Portugal).

uncertainty associated to predicted dynamic properties (also known as modal parameters, f , ζ and m , these being the frequency, the damping ratio and the modal mass, respectively) can be expected.

Apart from the design uncertainties, the modal parameters might change over time due to such external factors as environmental conditions, level of occupancy or the inherent ageing of structures. [Figure 1.9](#) shows the time variations of the natural frequency of the first three vibration modes of the structure for 6-month monitoring² and the distribution densities for these frequencies appear on the lefthand side of [Figure 1.9](#). The variation in the first two modes is much greater than that of the third one. According to the data shown, it can be seen that the modal parameters are not deterministic values and the dynamic response of the structure may differ substantially from the numerically estimated response.

Usually, an equivalent system of the structure at the more sensitive-to-vibration point is extracted i) with only the most representative vibration mode or ii) more accurately, including several vibration modes. Usually, to assess the VSLs, resonant conditions are assumed and a Single-Degree-of-Freedom (SDOF) approach is

²This is an introductory example. Another lightweight footbridge is studied in [Chapter 3](#).

(a) Time domain. Green (maximum comfort) and red (medium/minimum comfort) areas to separate different levels of comfort.

(b) Frequency domain.

Figure 1.8: Experimental test of 1-hour recording at the Porto footbridge.

adopted from design guidelines [73], retaining the fundamental mode at a particular position. In many practical situations, an SDOF model approach is enough to accurately represent the structure's dynamic response and/or the worst scenario. For instance, in beam-type footbridges, if the fundamental natural frequency lies within the excitation range (1.5–4Hz) and the second bending mode is approximately four times the fundamental one, this being clearly outside of the frequency band of interest.

Table 1.1: Mean values of the modal parameters of the Porto footbridge studied [139].

Footbridge parameters	Mode 1	Mode 2	Mode 3
Frequency, f_i (Hz)	1.87	2.02	2.96
Damping ratio, ζ_i (%)	0.86	1.14	0.94
Modal mass, m_i (kg)	12 364	10 346	10 583

1.2 Tuned Mass Dampers

Generally, the control strategies can be classified as: active, semi-active, hybrid and passive. Briefly, active control devices are those that apply controlled forces to the structure (the main drawback is the need for continuous power supply and the risk of instabilities associated with the real-time control system). Regarding passive systems, these can be divided into dissipative and inertial systems.

Dissipative systems are those that are somehow able to dissipate energy (by means of a dissipative effect such as friction, heat, etc). Inertial control systems, such as TMDs, do apply inertial control forces on the structure without consuming energy and so, there is no risk of instability due to their passive nature. The hybrid and semi-active strategies are in the path between the active and the passive. A hybrid system is composed of both a passive and an active device, while semi-active is mainly a passive system continuously tuned to the structure's movement using a smart system whose energy consumption is typically much less than a purely active device. They will be seen in [Chapter 2](#).

The most widespread and commonly used strategies are passive. Within these, TMDs are widely used. [Figure 1.10a](#) shows a clearly growing tendency to use TMDs as a passive vibration control solution in footbridges. Meanwhile, [Figure 1.10b](#) shows that more than 50% of the cases where TMDs are used are on footbridges.

Focusing on passive inertial mass controllers, the TMD consists of a passive damping technology with a purely mechanical operation that has proven to be significantly effective under resonance conditions. Its main advantages are the simplicity of its behaviour and its minimum maintenance costs. Thus, it has been shown to be a cost-effective and reliable solution for reducing vibrations. More concretely, a TMD consists of a mass suspended on springs and dampers and attached to the

Figure 1.9: Tracked frequencies of the first three vibration modes of the footbridge for 6-month monitoring.

Figure 1.10: Information about TMDs installed.

structure. Figure 1.11a shows that the first vibration mode has its maximum sag of the modal shape at mid-span, so a modal model (SDOF) can be made to represent the dynamic behaviour of the uncontrolled structure as shown in Figure 1.11b. A control device can be placed at this point (see Figure 1.12a) and also a simplified model (2-DOF) can be derived, as shown in Figure 1.12b. Figure 1.12 shows schematically how to move from a structural model to an equivalent modal model for TMD analysing.

Figure 1.11: Equivalent SDOF system at mid-span.

Figure 1.13 shows two examples of TMD mechanical designs to reduce vibrations, vertical (1.13a) and lateral (1.13b), developed by VICODA GmbH, a company specialized in vibration control devices. The design of a TMD depends on the oscillation frequency and mass of the structure, the direction of the movements (vertical or lateral) and the available space.

Its correct operation depends on the proper choice of its parameters. This process is called TMD tuning. The stiffness and mass of the device are selected to provide

Figure 1.12: Typically adopted simplified model for a structure with a vibration control device at mid-span.

Figure 1.13: Examples of mechanical design of TMD development by VICODA.

a TMD resonance frequency very close to the structure's resonance frequency. In this case, it can significantly reduce the structure's resonant response. The TMD damping level is selected to provide functioning over a frequency range and the mass is selected according to the desired level of vibration mitigation. Pedestrian-induced vibrations can be effectively controlled with a TMD due to their inherent periodic nature which might produce resonant response of the structure. If there is only one problematic vibration mode and their modal parameters do not vary excessively over time, the TMD will achieve high vibration reduction. However, the TMD performance might significantly degrade if:

- there are variations in modal properties over time due to external factors (temperature, added pedestrian mass, non-linearities dependent on the level of excitation, deterioration, ageing, creep, etc) that become uncertainties associated to the system;
- there is more than one vibration mode that can be excited in the frequency range of human-induced excitation (including walking, running or jumping).

Thus, it might exhibit a lack of TMD performance, since it is sensitive to the tuning and/or it cannot cope with vibrations coming from several modes. For

example, the eigenfrequencies of a structure may not be known to a sufficient level of accuracy when the TMDs are designed and the final parameters are usually determined through in-situ testing. The tuning of the TMD parameters to the first vibration mode (see Table 1.1) of the Porto footbridge as a target is shown in Table 1.2.

Table 1.2: Parameters of the TMD tuned for the first mode of the Porto footbridge.

TMD parameters	Values
Frequency, f_T (Hz)	1.85
Frequency ratio, η (-)	0.9893
Damping ratio, ζ_T (%)	8.65
Mass, m_T (kg)	247.28
Mass ratio, μ (-)	0.02

The motion control of the primary mass m_s is obtained by the pair of forces applied by the spring F_s and the damper F_D that connect the two bodies, given respectively by

$$F_s = k_T (x_T - x_s), \quad (1.1)$$

$$F_D = c_T (\dot{x}_T - \dot{x}_s), \quad (1.2)$$

where $(x_T - x_s)$ represents the relative displacement between the two masses and $(\dot{x}_T - \dot{x}_s)$ the corresponding relative velocity while, k_T and c_T are the spring and damper constants respectively. Both forces generate the total control force, $F_C = F_s + F_D$. The tuning of TMD parameters is much more sensitive to frequency than to damping deviations. This happens due to the greater weight of the spring force relative to the damper force in the control action.

Figure 1.14 shows a time history of the steady-state acceleration of the primary mass m_s when it is subjected to a harmonic load of the type $F = F_0 \sin(2\pi f_s t)$ where F is the dynamic force overtime (t), and F_0 and f_s are the force amplitude and structural natural frequency (first vibration mode) to which the TMD is tuned, respectively. Also the control force F_C is shown in Figure 1.14. For Figure 1.14a (tuned case), a first observation is that the control force is in opposite phase to the excitation force and both forces are delayed $\pm 90^\circ$ relative to the structure acceleration. The phase control theory indicates that the TMD can develop the maximum structural vibration reduction with this phase lag. However, for a frequency detuned case of 10 % (see Figure 1.14b), the opposite phase between the control and excitation force begins to be lost, which causes a significant loss of efficiency of the passive control device since the peak acceleration goes from being 0.97 m/s^2 to 1.57 m/s^2 . It can be concluded that a small detuning (which obviously is really easy to happen) of the TMD frequency leads to a phase angle far from the opposite phase between the excitation and control forces, corresponding to a degradation in the control action. The detuning of the damping ratio is not so conditioning, because the excitation and control force curves remain mostly opposite in phase.

Figure 1.14: Structure acceleration (—), excitation (—) and control(---) forces for the tuned and detuned cases.

In the frequency domain, the transfer function (TF) curves between the response acceleration of the structure and the excitation force show a broader spectrum of the vibration control performance. Figures 1.15 and 1.16 show the TF of the tuned and detuned cases for a single vibration mode (1.15) or several (1.16), respectively, in the frequency domain. Figure 1.15 shows the magnitude and phase of the tuned case (1.15a and 1.15b). TMD increases the structural damping of the structure and thus reduces resonance vibrations. The detuned cases (1.15c and 1.15d, 1.15e and 1.15f) are obtained by moving the frequency of the target vibration mode, keeping the TMD with the initial tuning. Meanwhile, Figure 1.16 shows only the magnitude of the same cases, but in a Multi-Degree-of-Freedom (MDOF) system, using the same TMD (tuned to the first mode) with the purpose of studying the effect in other different vibration modes. In all the figures, the curves relative to the tuned case have been kept as gray-dashed (---) curves to make the comparison easier.

Figure 1.15: Magnitude and phase in frequency domain. SDOF cases. The curves relative to the tuned case have been kept as gray-dashed (---) curves to make the comparison easier

(a) Tuned case.

(b) Detuned (+10 %) case.

(c) Detuned (−10 %) case.

Figure 1.16: Magnitude in frequency domain. MDOF cases. The curves relative to the tuned case have been kept as gray-dashed (---) curves to make the comparison easier

Table 1.3 shows a summary of the TMD performance, in terms of vibration reduction, for all the cases. The values are obtained according to two performance indicators: one related with the maximum value of TF magnitude (H_∞) and another indicator associated to the area under the curve (H_2). These indicators are

well known in the vibration control field and will be explained in more detail in [Chapter 2](#). The loss of efficiency suffered by the passive device outside the tuned case is important, reaching performance losses of up to 25.31 %.

Table 1.3: Performance of TMD for vibration reduction.

Model	Case	H_∞ (%)	Var. (%)	H_2 (%)	Var. (%)
SDOF	Tuned	84.90	–	26.90	–
	Detuned (+10 %)	69.74	15.16	22.40	4.50
	Detuned (–10 %)	64.60	20.30	23.08	3.82
MDOF	Tuned	59.81	25.09	19.59	7.31
	Detuned (+10 %)	63.98	20.92	21.81	5.09
	Detuned (–10 %)	59.59	25.31	15.81	11.09

1.3 Semi-active Tuned Mass Dampers

To mitigate the loss of performance due to detuned problems shown for the passive case (see [Figure 1.15](#) and [Figure 1.16](#)), semi-active control systems are used to broaden the operation frequency band of the control device, thus increasing its robustness against the aforementioned uncertainties. A typical approach has been to make TMD dynamics adaptive through adjustable natural frequency or damping, known as adaptative or self-adjustable TMD [175]. Adaptive TMDs are occasionally confused with semi-active TMD (STMD). A truly semi-active system can alter its adjustable parameter several times during a dominant vibration cycle [118].

Thus, semi-active systems are characterized by the fact that the damping properties and the natural frequency of the TMD can be adapted in real-time (see 1.6). These devices may be especially recommendable for structures with variable modal properties and/or several problematic vibration modes. [Figure 1.17](#) shows a model of an STMD installed on a primary structure, in which the red symbol ($\nearrow$) means that the damper properties are changing over time. The STMD model ([Figure 1.17b](#)) replaces a passive damping element with a controllable damping element ([Figure 1.17a](#)), usually a Magneto-Rheological (MR) damper, which is the key element for the STMD.

This thesis mainly focuses on studying the particular case of a TMD that is upgraded to operate with a semi-active control strategy (STMD) using an MR damper as a smart device (MR-STMD). To obtain the best performance of this semi-active inertial controller, it is necessary to study not only the optimal tuning of the new device, taking into account the nonlinear behaviour of the damper, but also all the aspects necessary for its implementation. Many control laws can be found in the literature; however, this thesis concentrates on those which are simple and clearly-geared-to-implementation.

Figure 1.17: Model of structure with TMD (left) upgraded into a structure with STMD model (right). Red arrow ($\nearrow$) means changing over time.

1.4 Thesis objectives

This thesis deals with the complete path followed from the moment that a structure is perceived as uncomfortable by the users to the final possible solution. It takes into account the problem of identification, tracking modal parameters and vibration control in structures with time-varying modal parameters and/or several problematic vibration modes that need to be cancelled. The aims of this Doctoral Thesis are to:

- Study the variability of the main parameters that identify a structure through real measurements, frequency and damping, in order to quantify the error limits associated with the identification method and the computational approach.
- Develop a method for tracking the modal parameters of a structure and study their variations. This tracking may be carried out for structures whose modal parameters change significantly. This step is prior to the implementation of the most convenient vibration control strategies. Additionally, once the modal parameters have been tracked, the influence of external factors can be removed and these might be considered for a Structural Health Monitoring (SHM) system.
- Study the performance of several easy-to-implement semi-active strategies for inertial controllers applied to one-year experimental estimates of an in-service structure considering also the implementation difficulties of each one.
- Develop a design methodology for the parameters of the semi-active inertial controller in order to optimise the vibration reduction over a broad-frequency band.

- Study an MR damper through experimental tests and identify the parameters of a phenomenological model. The derived model will be included in the optimization process in such a way that the degradation from an ideal damper can be quantified.
- Assess the technical and practical feasibility of implementing STMD in a laboratory structure and study its performance as compared to its passive version.

1.5 Thesis outline

This document has been divided into 6 chapters. The first introduces the research line. An example of a structure with time-varying modal parameters that might exhibit excessive vibration has also been presented. The solution of installing a TMD in the structure has been studied numerically and the problem of detuning has been described, both for the case of an SDOF model with uncertainty and for that of an MDOF model.

The remaining chapters are organized as follows:

- [Chapter 2](#) briefly reviews the fundamentals of vibration control, focusing on vibration serviceability, human vibration perception and human-induced excitations. Research works on structures with time-varying modal parameters are presented. Also, this chapter brings together the theory of and research into passive and semi-active control via inertial controllers. Finally, the chapter ends by describing MR dampers as the smart device used to materialise the semi-active vibration control system.
- [Chapter 3](#) studies an in-service structure in which the modal parameters vary over time. In this chapter, a tracking method is developed.
- [Chapter 4](#) studies two implementable semi-active control strategies for inertial controllers. Furthermore, performance and assessment of the pros and cons of the possible practical implementation are fully described.
- In [Chapter 5](#), an optimization procedure through a Performance Index (PI) to design the STMD parameters using a phase control law is presented. The degradation of the STMD performance when an MR damper model is used (as compared to an ideal viscous damper) is studied. Finally, to complete the procedure, a methodology for the identification of MR dampers using phenomenological models is presented.
- [Chapter 6](#) presents the main conclusions of the thesis and possible future research lines are proposed. The main publications due to the research work carried out within this thesis are also listed.

2

Vibration control

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A state-of-the-art of structure serviceability due to human-induced vibration, vibration control and MR dampers is presented in this Chapter. Also, some background knowledge used in the thesis is introduced.

2.1 Human-induced vibration

According to [55], the designed structure must satisfy a set of safety and serviceability requirements. The former concerns extreme loadings that are likely to occur no more than once during a structure's lifetime. Serviceability is associated with moderate loadings that can occur several times during the structure's lifetime. The structure should ideally be fully operational during service loadings, for example, the structure may suffer inconsequential damage, while the motion experienced by the structure should not exceed the specified comfort limits for humans and the motion sensitive equipment mounted on the structure. One example of a human comfort limit is a restriction on acceleration; humans begin to feel uncomfortable when acceleration reaches a certain value; however that value depends on the type of structure, the position of the person (sitting, standing, lying on the floor), the frequency of the vibration and the discomfort threshold of each particular person.

Advanced material technologies, together with the use of new construction techniques, may lead to slender structures with low fundamental natural frequencies as well as low damping ratios. These structures are sometimes susceptible to human movements such as walking, running, bouncing or jumping. This happens particularly when excitation frequencies, or their harmonics, are close to a structural natural frequency. Because of their slenderness, contemporary footbridges are often highly susceptible to human-induced vibrations. Pedestrians may excite the footbridge deck with a periodic load and they might even synchronize their motion with other pedestrians or the structure itself. This situation can introduce large vibrations into the structure. These vibrations might not compromise the structure's Ultimate Limit State (ULS) but they might lead to excessive vibrations, disturbing the user, who may feel unsafe [146].

2.1.1 Human vibration excitation

Human-induced excitation includes such activities as jumping, walking, running, dancing and so on. These excitations exhibit mainly periodical time force components in the three axes of movement that can be treated as a Fourier series. The main frequency components of the Fourier series are usually within a certain frequency band [1, 95, 96]. The force of a pedestrian corresponds to each footfall and the total force response can be obtained by combining individual foot forces (see Figure 2.1b) [213]. The typical frequency ranges involving the fundamental

frequency (pacing frequency) are defined by Bachmann et al. [16] for different types of human activities and they are shown in Table 2.1.

Activity	Frequency (Hz)
Walking	1.6 – 2.4
Running	2.0 – 3.5
Jumping	1.8 – 3.4
Bouncing	1.5 – 3.0

Table 2.1: Typical frequency ranges for different types of human activity [16].

For example, when walking, periodic force signals are generated according to the longitudinal, lateral and vertical directions. Figures 2.2a and 2.2b (zoom view) show a time history of the vertical pedestrian walking force measured by a 3D instrumented treadmill (see Figure 2.1a) [30]. Figure 2.2c shows the frequency spectrum of the vertical force obtained through the Fast Fourier Transform (FFT). The fundamental harmonic of the signal is the highest one; nonetheless, these higher harmonics might significantly excite structural modes, especially when they have low associated modal masses.

Figure 2.1: Walking force generated by a pedestrian.

Due to the periodic nature of the force, this is modelled by a Fourier series as follows:

$$F_v(t) = Q \left(1 + \sum_{h=1}^n \alpha_h \sin(2\pi h f t + \phi_h) \right), \quad (2.1)$$

in which f is the frequency component of repetitive loading, t is the time, h is the harmonic number, n is total number of harmonics considered, 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

2. Vibration control

(a) Time history of vertical force.

(b) Zoom view of time history of vertical force.

(c) FFT of vertical force. Frequency domain.

Figure 2.2: Vertical pedestrian force measured by a 3D instrumented treadmill [30].

lags) are given, for instance, in Table A.1 (jumping) and Table A.3 (walking) of [95]. Using this force load as a starting point and each different modalities of group or flow of pedestrian, the VSLs can be analysed from the model of the structure. New trends on human-induced loading work on probabilistic-based force models taking into account the uncertain nature of the excitation [162].

2.1.2 Vibration serviceability

The structures do fulfil the ULS and therefore satisfy the criteria for structural integrity, such as compression, tension or bending, but sometimes those requirements do not suffice to fulfil the Serviceability Limit State (SLS), which ensure that a structure remains functional for its intended use and does not cause occupant discomfort. When the structure does not comply with the VSL, a vibration control device is a possible solution to explore.

2.1.2.1 Guidelines and Standards

Eurocode 0 [3] merely specifies that some criteria and requirements should be set in order to try to avoid vibration problems and fulfil traffic comfort criteria. In the case of footbridges, it limits the maximum acceleration in the deck due to the action of pedestrians at 0.7 m/s^2 in the vertical direction and at 0.2 m/s^2 in the lateral direction. This verification should be carried out if the fundamental frequency of the deck is less than 5 Hz for vertical vibrations or 2.5 Hz for lateral vibrations.

The approaches in guidelines as Sétra [188] and HiVoSS [76] are very similar, but they differ in the load cases that must be considered. For example, HiVoSS does not consider the lateral load's second harmonic. It also presents slightly different approach to account for modifications in the structure's dynamic behaviour due to the extra mass of the pedestrians. According to Sétra, the natural frequencies are determined using two hypotheses: i) for the unoccupied structure and ii) the structure plus an added mass of 70 kg for each pedestrian, distributed over the entire surface of the bridge deck. These frequencies limit the upper and lower bounds, respectively, for the range of natural frequencies used to assess the probability of resonance in a given mode. In the case of HiVoSS, the additional modal mass has to be considered once the lower bound exceeds 5 % of the modal mass of the unoccupied bridge deck. The research of Van Nimmen [189] takes into account the expected uncertainty of the structure's predicted natural frequencies, while also considering the influence of the additional mass for both lateral and vertical directions. This reference suggests applying the loading coefficient as presented in Figure 2.3, when the assessment is carried out at the design and rehabilitation stages. To do so, for which the frequency bounds have to be widened by 15 % with respect to the centre frequency of each interval.

In a near future, the plan is to add dynamic models of pedestrian loads to chapter 5.7 of the EN 1991-2 (Eurocode 1: Actions on structures-Part 2: Traffic loads on bridges) [2]. This new chapter relies on insights already used in ISO 10137:2007 [95] and HiVoSS. According to the said guidelines and standards, Table 2.2 shows the comfort values for vertical acceleration in pedestrian bridges.

Figure 2.3: Suggested loading coefficient to be applied in the design stage (---), rehabilitation (—) and (black) original, as defined in Sétra and HiVoSS for vertical (a) and lateral (b) loading [189].

Comfort classes	Degree of comfort	a_{lim} (m/s ²)
CL1	Maximum	< 0.5
CL2	Medium	0.5 – 1.0
CL3	Minimum	1.0 – 2.5
CL4	Intolerable vibrations	> 2.5

Table 2.2: Comfort for vertical acceleration according to Sétra [44] and HiVoSS [76].

2.1.2.2 Comfort predictors

The acceleration, compared to other parameters, has the advantage that it is an absolute measurement in which no reference is needed. Current accelerometer technology offers a wide variety of sensors at fairly reasonable prices. Thus, available instrumentation for measuring acceleration can be used for human-induced vibration measurement.

The vibration that can be perceived by a human depends on the direction of incidence in relation to the human body, the duration of sustained vibration and the frequency content of the said vibration (for a given amplitude), among other factors. The variation in the frequency sensitivity for a body position can be taken into account by enhancing or attenuating the system response for frequencies where perception is more or less sensitive, respectively. The degree to which the response is enhanced or attenuated is referred to as frequency weighting and its functions are applied to make the vibrations for different directions and body positions acceptable. ISO 2631-1:1997 [96] gives details of frequency and direction weighting functions that can be applied and they are all based on the basicentric coordinate system shown in Figure 2.4a. According to ISO 2631-1:1997, for z -axis vibration and standing and seating, the frequency weighting function (W_k) is a filter with the frequency response shown in Figure 2.4b.

Figure 2.4: Frequency weighting depending on user's position [96].

A single magnitude for ease of comparison is extracted (usually known as the vibration predictor: the peak value ($a_{w,pk}$) or the Root Mean Square (RMS) value ($a_{w,RMS}$)) from the frequency-weighted acceleration time history. Mathematically, they are defined as follows:

$$a_{w,pk} = \max(|a_w(t)|) \quad (\text{m/s}^2), \quad (2.2)$$

$$a_{w,RMS} = \sqrt{\frac{1}{T} \int_0^T a_w^2(t) dt} \quad (\text{m/s}^2), \quad (2.3)$$

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

The peak value is usually adopted for the evaluation of footbridges. Table 2.2 shows the comfort classes adopted by Sétra [44] and HiVoSS [76]. As stated in the table, the maximum comfort is achieved for vertical vibrations smaller than 0.5 m/s^2 .

Weighted RMS values are adopted in two different versions: basic 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) evaluation should be adopted. The additional evaluation states the use of the running RMS value defined as follows:

$$a_{w,RMS,\tau}(t_0) = \sqrt{\frac{1}{\tau} \int_{t_0-\tau}^{t_0} a_w^2(\tau) d\tau} \quad (\text{m/s}^2), \quad (2.4)$$

in which τ is the integration time for running averaging (normally $\tau = 1 \text{ s}$ is recommended) and t_0 is the time of observation (instantaneous time). The vibration

predictor is defined as the maximum value of $a_{w,RMS}(t_0)$ and this value is usually known as the Maximum Transient Vibration Value (MTVV):

$$\text{MTVV} = \max(a_{w,RMS,1}(t_0)), \quad t_0 \in [0, T]. \quad (2.5)$$

Another vibration predictor, based on acceleration values, is the Response (or Multiplicative) Factor (denoted here as the R-factor). This is computed as the ratio of the running RMS weighted acceleration to a base curve representative of human-induced perception. For instance, this base curve, which represents an iso-perceptibility curve, takes a value of 0.005m/s^2 from 4 to 8 Hz, which is a representative value of human vibration perception. The Cumulative Distribution Function (CDF) of a random variable is a method to describe the distribution of random variables. The advantage of the CDF is that it can be defined for any kind of random variable (discrete, continuous, and mixed). It is always a non-decreasing function. The empirical CDF over time computes the CDF for a specific time history data of a particular variable [58, 117]. The CDF of the R-factors are usually computed in order to obtain the real percentage of time by which a particular R-factor (or other magnitude) is exceeded (or not exceeded). Figure 2.5 shows these two curves, exceed and non-exceed, computed for a time history acceleration response structure. These CDF curves computed for different variables can also be very useful and interesting for comparing different cases of vibration control, as will be seen in Chapter 4 and Chapter 5.

Figure 2.5: CDF curves.

It is common practice to multiply the MTVV by square $\sqrt{2}$ in order to compare the resulting value with comfort limits in terms of peak acceleration. The latter assumption works quite well in footbridges, since the worst scenario is, usually, mainly governed by the fundamental vibration mode excited by the first harmonic of the human excitation.

Finally, the Vibration Dose Value (VDV) is presented. This is also known as the fourth power vibration dose method, is another performance indicator used to

assess the VSLs. It explicitly considers the intermittent nature of the excitation and it is computed as follows:

$$\text{VDV} = \left[\int_0^T a_w^4(t) dt \right]^{0.25} \quad (\text{m/s}^{1.75}). \quad (2.6)$$

The limits of VDV values set for offices can be found in [1, 178]. Figure 2.7 shows an example of response for an experimental test on a full-scale laboratory structure. The structure is a 5.5 m span mixed concrete-steel slab (see Figure 2.6) with its first vibration mode at approximately 5 Hz. Thus, it is an example of a light-weight structure prone to vibrate under human-excitations. Figure 2.7a and 2.7b show the structure acceleration and the evolution of the VDV, respectively, under several heel-drops. The commented predictors are computed in Figure 2.7c and compared in Figure 2.7d. The peak acceleration, as measured, is not really a representative value of the Vibration Serviceability (VS) of the structure, since very high instantaneous peaks can be achieved which are highly dependable on the sampling frequency of the signal and may come from an out-of-perception frequency range. The predictors described up to now avoid or alleviate this fact.

Figure 2.6: A general view of the laboratory structure, instrumented with accelerometers underneath.

All the predictors mentioned up to this point are time domain indicators focused on the study of VSLs. However, there are frequency response values that focus on the optimization of vibration control systems and which can also give a reference of the level of vibrations reached. Thus, the H_∞ and H_2 values derived from the Frequency Response Function (FRF) are introduced hereof. Using the Laplace transform, it is possible to convert a system's time-domain representation into a frequency-domain input/output representation, also known as the TF, $G(z)$, z being the complex variable. It also transforms the governing differential equation into an algebraic equation which is often easier to analyse. The H_∞ (“H-*infinity*”) of a TF is defined as

$$H_\infty = \max_z |G(z)|, \quad (2.7)$$

where the H_∞ value is the maximum value of the TF amplitude (see [Figure 2.8a](#)). This is the value usually adopted to design TMDs. An optimization with respect to the H_∞ value is recommended for structures that are excited by loads exhibiting mainly periodical time components (i.e., loads generated by such human activities as walking, running, jumping, dancing etc., or by machines).

Meanwhile, the H_2 value is defined as

$$H_2 = \sqrt{\frac{1}{2\pi} \int_{z=-\infty}^{z=+\infty} G^2(z) dz}, \quad (2.8)$$

and is essentially a measure of the area of the square of the TF amplitude (see [Figure 2.8b](#)). For loads having mainly a wide frequency band stochastic character (i.e., wind loads, earthquake loads) and/or in which several vibration modes of the structure are susceptible to be excited, an optimization with respect to H_2 is more recommendable.

The described predictors will be used to evaluate the comfort level or the vibration reduction level during the following chapters of this research.

2.2 Modal parameter uncertainty

[Figure 2.9](#) shows a scheme of the modal parameter identification process, with the different associated sources of uncertainties both, external and internal. The sources of uncertainty are described in [Subsection 2.2.1](#), as well as several study examples in [Subsection 2.2.2](#). The uncertainty of modal parameters due to the selected duration of the measurement and the use different identification methods will be studied and quantified in [Chapter 3](#).

2.2.1 Sources of uncertainty

The procedure for the estimation of the modal parameters of structures from measured data can be split into three distinct steps: i) data collection, ii) system identification and iii) modal parameter extraction.

When performing vibration tests on large structures, such as civil engineering structures, the data collection step is subjected to important constraints. It is often unpractical and expensive to use artificial excitation, such as shaker tests or controlled drop weight tests, because of their size, mass and/or power consumption. There are two main types of experimental tests, output-only and input-output. In the last two decades, the first have been imposed, since they are the most appropriate in continuous monitoring. The long-term monitoring of vibrations in civil engineering structures is increasingly used to monitor both structure integrity and vibration serviceability.

(a) Complete view.

(b) VDV running.

(c) Zoom view.

(d) Peak comparison.

Figure 2.7: Example of time history acceleration for an experimental test of a full-scale laboratory structure with computed predictors.

Figure 2.8: Example of H_∞ and H_2 graphical representations.

Figure 2.9: Uncertainties in the modal parameter identification process.

Output-only modal analysis, also known as Operational Modal Analysis (OMA), aims to identify the modal properties (natural frequency, damping ratio and mode shape) of a structure using only the (output) vibration response (acceleration, velocity, etc.) [13, 27, 28, 157]. The (input) excitation to the structure is not measured, but is assumed to be stationary white Gaussian noise with zero mean. This allows vibration data to be collected when the structure is in its working or “operational” condition without much intervention. These advantages, the lack of controlled input and minimum impact on the structure service, have contributed to the increasing popularity of OMA techniques in practical applications [31, 83, 97, 204]. With no loading information, the identification uncertainty of the modal parameters from the ambient vibration data is significantly higher than that in impulse or forced vibration tests. Indeed, additional uncertainties are introduced if the input

does not fulfil the stationary white Gaussian noise assumption and the effects of structural/environmental changes [10, 106, 108]. Therefore, uncertainty quantification and quality control of the identified modal properties become especially relevant [35, 82, 107, 152].

A number of examples of bridges and footbridges that are equipped with a monitoring system can be found [60, 81, 89, 138], where the said systems measure and record the vibrations due to traffic or wind. These data can then be used to continuously extract the modal parameters (natural frequencies, damping ratios and modal shapes) that may then be useful for assessing the structural integrity, as structural damage leads to changes in the modal parameters. Thus, vibration-based SHM systems using automated OMA identification have been extensively proposed [68, 126]. One of the main problems of these systems is that they have to cope with changing environmental/operational conditions, which often significantly affect the modal estimation. Hence, the removal of these spurious factors on modal estimates becomes crucial in the implementation of an SHM system that uses modal parameters as damage-sensitive features. These factors may mask meaningful changes for damage detection, due to, for instance, extraordinary events such as earthquakes or very dense traffic [186], especially in structures, such as bridges, that are always exposed to environmental factors.

As for the use of modal parameter variations as possible damage detectors in structures, the modal variations over time due to the influence of some external agents acting on the undamaged structure must be observed [115, 123]. Evaluating the magnitude of this modal variation is important. The most significant environmental factor is usually the temperature [211]. However, there are other factors, such as operational or boundary conditions, which may significantly affect the modal estimates [14, 115]. Several studies have dealt with this problem and have carried out correlation analyses between modal estimates and these external factors [92, 120, 121, 136, 144], as it is crucial to discriminate between modal variation induced by environmental or external factors and those induced by structural damages.

Methods generally consist of automated peak-picking for frequency domain methods [29, 163] or clustering methods for stabilization diagram interpretation [32, 167]. Since all OMA techniques require some user judgment, a non-trivial degree of automation is necessary to bridge the gap between traditional and continuous OMA. Research into automated OMA for civil structures has recently been on the rise. Formulations that use state-space models, designated as Stochastic Subspace Identification (SSI) methods, constitute the parametric approach which is more commonly adopted for civil engineering applications in the time domain. Two of the most popular and accepted SSI implementations are [155]: a covariance-driven (SSI-cov) and a data-driven (SSI-data) implementation [165]. They are also recommended when there are closely-spaced eigenfrequencies [122, 124]. Accordingly, the methods used in Chapter 3 are based on SSI.

The literature concerning the application of these methods focuses mainly on bridges and the effects of environmental and operational conditions on their modal parameters, primarily natural frequency. Environmental studies deal with such factors as temperature [61, 156] and wind [61, 177], while the operational studies consider such factors as vehicle traffic [125, 210] and pedestrians [88, 90, 91]. Understanding these effects is useful in SHM so as to reduce the probability of false positives in damage detection. The structures also experience highly variable levels of excitation, including very low amplitudes, which makes the extraction of consistently reliable modal information much more challenging [169].

Instead, FEM updating is an inverse problem to identify and correct uncertain modelling parameters, leading to better predictions of the dynamic behaviour of a target structure. FEMs are usually updated using the modal parameters [51, 134]. However, when the said parameters show a high variability with changing environmental factors, the updated model might not be as accurate as necessary [21]. Thus, modelling uncertainty, as well as the test–analysis correlation error model, are needed to assess the prediction accuracy of the FEM [185]. It is therefore important to estimate uncertainties in the modal parameter identification so as to establish the degree of confidence of the updated model. This updated model can be useful for guaranteeing a successful structural intervention during structural rehabilitation [57, 98]. That is, non-destructive tests, such as an OMA, together with calibrated models, greatly benefit reliable structural evaluations.

Thus, the modal parameters estimated by OMA include the values of the external characteristics existing during the measurement, hindering the knowledge of their real value. A method that can provide a correction value for the natural frequencies of vibration, removing the environmental effects and decreasing the uncertainty of this value, may be a useful tool, as it would allow the evolution of the structural behaviour over time to be analysed, improving the tracking of the structure's modal parameters and providing a step towards obtaining the possible detection of damage or structural change. It may also be useful to track the modal parameters of structures that are or will be monitored [182]. All these aspects are looked at in more detail in [Chapter 3](#).

2.2.2 Examples of study

Some research examples dealing with modal parameter uncertainties are now described. The Infante D. Henrique Bridge (see [Figure 2.10a](#)), over the river Douro at Porto, Portugal, has an arch span of 280 m between abutments and rises 25 m to the crown; thus exhibiting a shallowness ratio greater than 11/1 [123, 125]. [Figure 2.10b](#) presents the time evolution, from September 2007 to May 2010, of the bridge's first four natural frequencies associated with vertical bending modes. The influence of the annual temperature fluctuations on the values of the natural frequencies can be clearly seen.

(a) General view.

(b) Time evolution of the natural frequencies of the first 4 vertical bending modes.

Figure 2.10: Analysis carried out on the Infante D. Henrique Arch Bridge [125].

For this thesis, pedestrian structures, such as footbridges, are even more interesting, since fluctuations or uncertainties can be the main source of losing effectiveness for a TMD device. The studies carried out by Hu et al. [88, 91] analysed the modal parameter uncertainties associated to the stress-ribbon footbridge located at the FEUP University Campus. Figure 2.11 shows the structure (2.11a) and the evolution of frequencies over time (2.11b and 2.11c). The evaluation of vibration levels, the measurement of temperatures, the analysis of frequency components and automated structural modal identification are carried out in [88, 91] in order to examine and remove the effects of environmental and operational factors on the modal properties, while also detecting early structural changes under operational conditions.

2.3 Vibration control generalities

During the past few decades, structural control has attracted the attention of many researchers. The aim is to reduce excessive vibration using appropriate methods.

(a) FEUP campus footbridge

(b) Variation of estimated modal frequencies.

(c) Variation of the 1st frequency.

Figure 2.11: Example of structure with uncertainties in the modal parameters [91].

This section deals with some relevant aspects of the vibration control of civil structures. According to Connor [55], the structural damping ratio depends on the structural type, the material and the vibration level, among other issues, with typical values ranging between 0.5 % and 5 %. Damping is the process by which physical systems dissipate the energy input from external excitations, especially near resonance conditions. For low damping ratios, the energy dissipated per cycle is small and many cycles are required before the input energy is eventually dissipated. As the damping ratio, ζ , increases, the energy dissipated per cycle also increases. Energy dissipation is related to a number of external and internal mechanisms.

A range of vibration control techniques have been applied to improve different structural vibration performances, such as seismic and wind induced vibration. By and large, structural control systems can be grouped into three broad areas: (a) Base Isolation (BI), (b) PVC, and (c) active and semi-active vibration control [49]. The basic role of PVC devices when incorporated into a structure is to absorb or consume a portion of the input energy, thereby reducing the energy dissipation demand on the primary structural members and minimizing possible structural damage. Active, hybrid and semi-active structural control systems are a natural evolution of passive control technologies. Different combinations of them result in Hybrid (passive + active) or Semi-active (controlled passive) vibration control techniques. Figure 2.12 shows examples of different vibration control system models: (a) passive brace system, (b) active brace system, (c) BI system,

(d) structure with TMD, (e) structure with Active Mass Damper (AMD) and (f) structure with Hybrid TMD (HTMD). In this chapter, the main types of vibration control will be described, while focusing on those research works that use inertial mass controllers, passive (TMD) and semi-active (STMD). BI systems are not dealt with in this thesis, since this research work is focused on human-induced vibrations and BI systems are used to isolate structures from ground movements, such as those produced by earthquakes.

Figure 2.12: Examples of different vibration control system schemes [209].

According to Noormohammadi [146], in PVC methods, the vibration energy is dissipated by the introduction of additional material or devices to the primary structures, which raises their damping and sometimes stiffness. PVC methods do not require an external power source and are generally relatively easy to design [86]. However, they have relatively poor performance, particularly for low-level vibrations where they might not be fully engaged and might not be effective for a range of excitations.

In active control, a control force is applied to the structure through an actuator which is externally driven. Then, the actuator introduces forces to the structures in order to counteract the excitation forces. However, it has some disadvantages, such as possible loss of external power, requirements of the actuators, high power demand in case of large disturbance forces and the risk of unstable behaviour [86, 172].

A semi-active control system can be considered as a passive system where its damping and/or stiffness can be changed in real-time without introducing additional energy to the controlled structure [86, 147, 181]. Semi-active controllers have many advantages. They are cheaper than fully active systems and require low power. Since external energy is not applied directly to the structure, these devices are inherently stable. An important advantage for semi-active systems is their ability to work as purely passive systems in case of external power failure [8, 99, 171, 172, 181]. Thus, this thesis is focused on the design of these systems for highly uncertain structures.

Figure 2.13 shows a structure actively controlled with a feedforward control (open-loop control) and a feedback control (closed-loop control). Feedforward control can be used when the excitation to the system to be controlled (structure) can be measured or is previously known. Feedback control makes use of a measured response of the structure and interacts with it, applying control forces in order to achieve a particular control objective; in vibration control problems, the objective is zero response. Note that, as is shown in Figure 1.3, passive inertial mass controllers physically work as feedback control since they apply control forces depending on the response of the structure. These feedforward and feedback strategies can be used to convert a TMD in a STMD. When the inertial mass controller is semi-active (see Figure 1.5), this is changed/adapted continuously, this which will be carried out through feedforward control to the inertial controller itself.

Figure 2.13: Feedforward and feedback control applied to a structure.

Figure 2.14 shows a scheme of vibration control strategies, according to two types of classification: dissipation and inertial devices. The following sections describe the main vibration control strategies in a general way (passive in Subsection 2.3.1, active in Subsection 2.3.2 and semi-active in Subsection 2.3.3). Then, the TMD (Section 2.4) and STMD (Section 2.5) are dealt with. Figure 2.14 includes the section number of the chapter to which each topic corresponds.

The main difference between the two groups is explained conceptually in Figure 2.15. This illustrates the performance in frequency domain for dissipative (2.15a), passive (2.15b) and semi-active inertial (2.15c) inertial dampers, showing the uncontrolled and controlled frequency domain magnitude of the response. The grey shaded area indicates the performance band. The dissipative case is less effective reducing vibrations but acts in a broad-band frequency. Meanwhile, the passive inertial device is much more effective but in a narrow-band frequency. The

Figure 2.14: Scheme of vibration control strategies.

semi-active inertial strategy may have the advantage of both strategies and avoid the main drawback of active inertial strategy, energy consumption.

Figure 2.15: Examples of uncontrolled and controlled structure's response curves in frequency domain for different types of vibration control strategies.

2.3.1 Passive Vibration Control

In PVC, the reduction of vibration can be achieved by adding viscoelastic materials and/or devices to dissipate vibration energy are required; however, no external energy is needed. PVC has been used to mitigate vibrations under a range of excitations, such as seismic, wind, machinery and human induced vibration [56, 180].

There are various mechanisms by which passive systems may operate, such as metal yielding, frictional sliding, stage transformation in metals and viscoelastic deformation in solid or liquid materials and fluid orifices. Regarding the type of passive systems, these can be divided into dissipative and inertial. Dissipative systems are those that do not apply forces to the structure but which are able to dissipate energy. Inertial control systems do apply inertial control forces on the structure but do not consume energy, there is no risk of instability due to their passive nature and they have a relatively lower cost [6, 147, 148, 172, 174, 181, 201]. However, passive control methods typically have a number of drawbacks, such as lack of damping capacity and they are subject to various trade offs when excited across a large frequency bandwidth [8, 174]. Figure 1.2 from Chapter 1 shows a structure with PVC. Some examples of passive controllers are TMDs, base isolation, viscoelastic dampers, metallic yield dampers, friction dampers, viscous fluid dampers and tuned liquid dampers [181].

TMDs have been widely used to mitigate vibration in civil, mechanical and aerospace structures and much theoretical and practical research has been done in this area [147, 171, 181]. TMDs have been found to be an effective device to reduce human induced vibration caused by pedestrian activities on footbridges [170].

Apart from the inherent limitations of passive systems, their performance depends highly on the inertial mass value and, if the mass is too small, the effect of a TMD is limited. Also, since the dynamic properties of structures commonly vary over time due to external factors, the TMDs may not remain optimally tuned and their performance may be reduced. Furthermore, TMDs are generally efficient when the primary structure has low damping [6, 18, 148, 170, 171, 201].

2.3.2 Active and Hybrid Vibration Control

The inability to dynamically change a passive control system to compensate for an unexpected load tends to result in an over-conservative design. Considering this limitation, there is a potential for improving performance by real-time modifying the load and system properties [55]. The performance of Active Vibration Control (AVC) is based on force generation using an external source of energy. In this method, a control force is exerted on the structure through an actuator which aims to counteract the vibration caused by disturbance forces [174]. In this method, the disturbance (input) force and/or the response of the structure is monitored

continuously and the output information is used to calculate and generate the appropriate control force [71, 93, 187].

Generally, an active scheme consists of sensors and actuators in combination with a control unit [11, 174, 207]. The actuator force changes in real-time according to the control law adopted and the response of the structure [133]. The active control approach is the combination of several engineering disciplines, such as electronic engineering, computer engineering, control engineering and materials engineering [181]. AMD is a typical example of inertial mass active control device [146].

AMDs have the advantage that i) they are more effective than passive inertial systems for the same inertial mass, ii) the control target can be selected in a wide frequency band and, when robustly designed, iii) they are not sensitive to changes in the modal parameters of the structure or in the excitation load. On the other hand, they are systems more expensive than passive or semi-active and might become unstable (control law should be carefully designed); they stop working in a power supply failure, as could happen in the case of earthquakes [176, 179].

Active and passive control methods have some advantages and drawbacks. In an attempt to have the advantages of both, hybrid vibration control methods have been developed [174]. A hybrid controller is a combination of passive and active control systems [45], which is an attractive option for structural vibration control [9, 54, 141, 142].

Figure 2.16 shows a commercial inertial mass actuator (APS model 400) which has been used in this thesis as an exciter to apply controlled forces to the structures for identification purposes. This shaker has been extensively used as an AMD for controlling human-induced vibrations [69–71].

Up to now, some examples of AVCs in pedestrian structures have been reported. For instance, the research work carried out by Bleicher et al. [23] at TU Berlin describes the development of an AVC system for a light and flexible stress ribbon laboratory footbridge of 13.05 m span. To reduce pedestrian-induced vibrations, on-off pneumatic actuators are introduced into the handrail in order to change the stress of the ribbon in real-time. Figure 2.17 shows the laboratory stress-ribbon footbridge (2.17a) and pneumatic actuators for active control (2.17b). The main drawback is the nonlinear and discontinuous behaviour of these actuators.

A few of examples of AVC for human-induced vibration in footbridges and floors using AMD have been carried out over the last decade, both as a single-input single-output version and a multi-input multi-output version [39, 158, 191, 193]. The research work of Casado et al. [39] is concerned with the design and experimental implementation of a TMD (see Figure 2.18b) and an AMD (see Figure 2.18c) to mitigate human-induced vibrations on an in-service footbridge (see Figure 2.18a). Although the structure had one dominant vibration mode (which was well separated from the others) prone to be excited by human loading, the TMD mass had to be fine-tuned periodically, since structural natural frequencies change with structural ageing. Furthermore, environmental conditions (mainly temperature

Figure 2.16: Inertial mass actuator. APS Dynamic Model 400.

and wind) and pedestrian density might change the resonance response of the structure, thus affecting the TMD performance. According to [Casado et al.](#), the AVC hardware and installation might be estimated to cost between three and four times more than the TMD solution.

Meanwhile, the research carried out by [Wang et al. \[193\]](#) provides practical guidelines for the tuning of Direct Velocity Feedback Control (DVFC) considering the damping performance, inertial mass actuator limitations, such as stroke and force saturation, as well as the stability margins of the closed-loop system. The work carried out by [Pereira et al. \[158\]](#) consists in an AVC Multi-Input Multi-Output (MIMO) design methodology for controlling human-induced vibration based on DVFC in an indoor footbridge sited in the University of Exeter. [Figure 2.19](#) shows the indoor footbridge ([2.19a](#)) and the spectrum response for the uncontrolled and controlled cases ([2.19b](#)).

These kinds of control strategies might be most suitable for indoor structures such as office floors or indoor footbridges, used only in working hours or intermittently where the electrical supply can be provided more easily.

2.3.3 Semi-active Vibration Control

Semi-active systems improve their counterpart passive version by introducing certain adaptability capacities. The concept was introduced for the first time by [Karnopp et al. \[104\]](#) in 1974, who proposed to modify the force of a fluid damper

Figure 2.17: Laboratory stress-ribbon footbridge (a) using pneumatic actuators (b) for active control at TU Berlin [23].

by controlling the opening of a valve for automotive applications. Hence, the target was to obtain better isolation of the vehicle from the roughness of the road through car suspension systems. The first proposal for semi-active control of civil structures can be found in [87]. In that work, the concept of semi-active control was extended to civil buildings, proposing a TMD connected to the main structure with a semi-active viscous damper. The proposed device was a variable-orifice damper [41].

Semi-active control has many benefits, including less external power requirements in comparison with active control; as such, it might operate with affordable batteries. They are able to cope with system uncertainties and off-tuning issues. In contrast with active control, semi-active control is inherently stable [6, 99, 101, 148, 170, 172, 181]. However, semi-active control has some disadvantages as well. Its operating frequency bandwidth is not as wide as the active control [19, 72, 99, 101, 148]. It also requires harmonic excitations and resonant behaviour when is used to updating a TMD [19, 85, 85, 131].

Figure 2.20 shows a structure controlled by a semi-active TMD. The TMD can be controlled by a feedforward loop, the command to the TMD is directly obtained from the interaction magnitudes between both systems (relative velocity and structure displacement) as well as by a feedback loop, which may regulate the output of the system by measuring the force transmitted to the structure.

(a) Valladolid Science Museum Footbridge.

(b) TMD.

(c) AMD.

Figure 2.18: Passive and active vibration control on an in-service footbridge [39].

2.4 Passive control via Tuned Mass Dampers

The concept of TMD without an integrated damping device (undamped tuned vibration absorber) was introduced by the naval engineer Hermann Frahm in 1908. He created a device to reduce the rolling motion of ships as well as ship hull vibrations, and he published a patent titled “*Device for damping vibrations of bodies*” in 1911 [80] which has already expired. Since then, and throughout the twentieth century, they have evolved and adapted to the fields of aeronautical, mechanical and structural engineering in order to mitigate the vibrations due to different sources of excitation.

A TMD is a conventional passive control method to decrease the response of such flexible structures as skyscrapers or long bridges. It consists of a mass, spring and damper, which interact with the structure to which they are attached to dissipate vibration energy and to reduce the structure’s dynamic response. A TMD acts as a secondary moving mass attached to the main structure and it is tuned to operate

(a) Forum Footbridge. AMDs are marked with red area.

(b) Experimental results. Power Spectral Density response.

Figure 2.19: MIMO vibration control application in an indoor footbridge [158].

at a specific frequency, which is generally the one of the dominant structural vibration mode. Thus, when that frequency is excited, the inertial mass of the TMD vibrates in the opposite phase to the main system, imparting control forces to the structure and hence the vibration energy of the structure is transferred to the auxiliary mass, to be dissipated through the TMD damping element (see Figures 1.12b, 1.13 and 2.12d). This is why TMDs are effective mostly in resonant cases, where the frequency of the external force is close to the frequency of the structure [55, 148, 171, 201, 207]. The effectiveness of the TMD depends on a) the accuracy of its tuning, b) the inertial mass value as compared to the modal mass of the target mode, that is, its mass ratio, and c) the level of internal damping of the TMD. Apart from these main aspects, the TMD performance is highly dependent on the level of the excitation and the structure response since when the response is not purely resonant, the TMD will degrade rapidly its performance. Viscous dampers are commonly used as the energy dissipation element inside the

Figure 2.20: Control scheme of a structure with semi-active vibration control.

TMDs. A viscous damper is merely a piston submerged in a pot with viscous fluid flowing between chambers. The advantage of dash-pots is their simple design (see Figure 2.18b).

In spite of this, it still took a few years for the theory on TMDs and their tuning to begin to see the light. It was not until the publication of a study by [Den Hartog and Ormondroyd](#) in 1928 [66], followed by a detailed discussion of optimal tuning and damping parameters in Den Hartog's book on Mechanical Vibrations [67], that a systematic analysis of TMD parameters and their influence on structures was carried out. The initial theory was applicable to an undamped SDOF system subjected to a sinusoidal force excitation. Den Hartog used the fixed-points method in the undamped case to obtain an approximate solution of the optimal tuning parameters of the inertial absorber. An extension of the theory to damped SDOF systems has been investigated by numerous researchers. Since then, much research has been done to better understand the interaction between TMD and various structural types subjected to a wide range of excitation. One of the most important works in this field is the one developed by G.B. Warburton in the early 1980s, based on the works of [Warburton and Ayorinde](#) [194, 195]. In practical applications, the primary system can be assumed to have vanishing structural damping, so many attempts have been made to find exact, closed-form solutions of the optimal parameters. Recently, analytical approximations obtained by perturbation techniques were constructed by [Asami and Nishihara](#) [12]. In these works, analytical solutions to the optimal tuning of TMDs were presented for several types of excitation and with different minimization objectives. It was in the research work of [Bachmann and Weber](#) [15] in 1995, where a TMD appeared for the first time as a vibration solution applied to a footbridge.

More recently, thanks to the evolution of computer equipment, the structural models continue to become more complicated by improving and sophisticating the methodologies of design and optimization of TMDs. Thus, there are works such as [Miranda](#) [132], which presents an energy approach to the problem, [Krenk and Høgsberg](#) [116], which applies white noise as excitation to obtain the properties

of TMDs from a statistical minimization, and [Hoang et al. \[84\]](#) and [Adam and Furtmüller \[5\]](#), who perform two of the numerous comparative parametric studies with a certain completeness that have come to light during the first decade of the 21st century, among many others.

2.4.1 Theoretical design

The design of the mass damper involves specifying the mass, stiffness and damping coefficient (see [Figure 1.12b](#)). In Den Hartog's classical approach [\[67\]](#), the optimal TMD parameters are those that minimise the maximum value of the resonant peak at a point for a structure with vanishing damping ($\zeta_s \simeq 0$), with the structure modelled as an SDOF system, representing only the vibration mode to be controlled. Generally, this assumption can be adopted and the TMD is installed where the maximum displacement of the vibration mode to be controlled occurs.

In Den Hartog's tuning theory [\[67\]](#), TMD parameters are studied so as to minimize the structure displacement at the resonance frequency. The optimal TMD frequency and damping ($f_{T,\text{DenHartog}}$ and $\zeta_{T,\text{DenHartog}}$, respectively) are shown in the following equations:

$$f_{T,\text{DenHartog}} = \left(\frac{1}{1 + \mu} \right) \cdot f_s = \eta_{\text{DenHartog}} \cdot f_s, \quad (2.9)$$

$$\zeta_{T,\text{DenHartog}} = \sqrt{\frac{3\mu}{8(1 + \mu)}}, \quad (2.10)$$

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, $\mu = m_T/m_s$) and $\eta = f_T/f_s$ is the frequency ratio. It is the usually used tuning rule and/or it is used as the initial solution for optimization-based methods.

The closed-form approximations derived by [Asami and Nishihara \[12\]](#) include different inputs (excitation force or ground motion of the primary structure) and different outputs, such as acceleration, velocity or relative displacement. The analytical solution from the H_∞ optimization of TMD design parameters with vanishing structure damping, minimizing the acceleration response of the primary structure, are:

$$f_{T,\text{Asami}} = \sqrt{\frac{1}{1 + \mu}} \cdot f_s = \eta_{\text{Asami}} \cdot f_s, \quad (2.11)$$

$$\zeta_{T,\text{Asami}} = \sqrt{\frac{3\mu}{8(1 + \mu)}} \cdot \sqrt{1 + \frac{27\mu}{32}}. \quad (2.12)$$

The stiffness and damping ratio for the TMD are then obtained from

$$k_T = \omega_T^2 m_T, \quad (2.13)$$

$$c_T = 2\zeta_T m_T \omega_T, \quad (2.14)$$

respectively. TMDs are tuned to control the vibration due to a specific vibration mode. However, detuning effects should be considered due to the possible uncertainties associated with this mode. [Figure 2.21](#) shows the maximum value of the FRF amplitude (H_∞ value) for a range of TMD parameters. A significant reduction in the response of the primary system under the target vibration mode is observed when the TMD parameters are properly selected/tuned. The performance of the TMD is more sensitive to the frequency tuning than the damping tuning. TMDs are sized so that their inertial mass is large enough to achieve the expected vibration cancellation performance. Increasing the inertial mass of TMD results in a greater vibration cancellation capacity and increasing the TMD damping results in a greater frequency-band of effectiveness. However, the vibration cancellation capacity at the target frequency is reduced. That is, the TMD damping reduces the relative displacement of the inertial mass of the TMD (the motion of the TMD mass relative to the structure). [Figure 2.22](#) shows the vibration reduction (%) obtained for the [Figure 2.21](#) example, in the area of interest, from 0.9 to 1.1 for the frequency ratio (–) and from 0 to 10 % for the TMD damping ratio. It should be noted that, for very low damping values, the reduction goes from almost 100 % to 20 %. To avoid these irregular performance due to slight frequency detuning, the damping is increased and reductions close to 50 % are reached, losing the optimum previously achieved, but increasing the bandwidth of functioning. However, less effective vibration reduction is obtained under different frequency harmonics, impulsive or random excitation is also obtained due to its inability to reach a resonant condition and therefore dissipate energy under these different excitations.

2.4.2 Examples of vertical TMD in footbridges

TMD can be set to cancel vertical or lateral vibration. This thesis is concerned with vertical TMDs for human-induced vibrations. Some examples of TMD installation are briefly described hereof.

The case of the Millennium Bridge (see [Figure 2.23a](#)) is one of the best known due to the serious consequences that the vibrations had on this pedestrian structure, causing its closure to the public for two years. It is a steel suspension footbridge located in London. The design of the structure was the result of a collaboration between Arup Group, Foster and Partners, and the sculptor Anthony Caro. The bridge has two piers and is compounded of three main spans of 81 m, 144 m, and 108 m with a total structure length of 325 m. On the opening day, the structure showed excessive vibrations, mainly laterally. A retrofitting, including vertical and horizontal TMDs and lateral structure stiffening was undertaken. Focusing on the

Figure 2.21: FRF amplitude of a primary structure with a TMD, changing its damping ratio along the frequency.

Figure 2.22: Vibration reduction reached in frequency domain for [Figure 2.21](#) example.

vertical TMDs, 50 units were installed (see [Figure 2.23b](#)) with a mass between 1 000 and 2 000 kg, designed by the company GERB (see [Figure 2.23c](#) and [2.23d](#)), in order to mitigate oscillations belonging to frequencies between 1.2 to 2.2 Hz.

The GERB company also installed eight TMDs in the Solferino footbridge, located in the city of Paris (France). This footbridge is a double-deck structure constituted by two double arches (see [Figure 2.24a](#)). The footbridge was designed

2. Vibration control

Figure 2.23: TMDs in Millenium Bridge (2001). GERB company.

by the architect Marc Mimram and built by the Eiffel company. The main structure is 140 m length with a main span of 106 m. The steel wight is approximately 990 tn. Due to the excessive vibrations, several solutions were analysed to reduce them. It was considered impossible to stiffen the structure for aesthetical reasons but also for technical ones. Indeed, the use of horizontal stiffeners would reduce the vibration amplitudes, but they not change the acceleration levels, and consequently, would not improve the pedestrian comfort [59]. Finally, an alternative was to increase the modal damping by adding TMDs (see Figure 2.24b). Thus, the GERB company installed eight vertical TMDs, 4 of 2 500 kg for a vibration mode at 1.94 Hz and another 4 of 1 900 kg for other vibration mode at 2.22 Hz.

Another example is a footbridge located in the city of Bilbao (Spain). The structure has a total length of 142.25 meters and the main span of 84 meters (see Figure 2.25a). Several vibration modes of the structure laid within the common human-induced frequencies. Three couples of TMDs were installed by the company MAURER to mitigate vibration modes with frequencies of 1.85, 2.32 and 2.78 Hz, respectively. They all had the same basic design, but different vibration masses: 550, 725 and 1 200 kg, respectively (see Figure 2.25b). The same company installed two TMDs of 1 000 and 1 660 kg in a cable-stayed footbridge of 117.5 m

Figure 2.24: TMDs in Solferino footbridge (1999). GERB company.

length sited in Forchheim (Germany) (see [Figure 2.26a](#) and [2.26b](#)) to mitigate the vibration modes at 1.26 and 2.70 Hz, respectively.

Figure 2.25: TMDs in Abandoibarra Footbridge (1997). MAURER company.

Another well-known structure with a TMD installed is the Pedro e Inês Footbridge at Coimbra (Portugal). The footbridge has a total length of 275 m and is formed by a parabolic central arch with a span of 110 m and two half lateral steel arches, supporting a composite steel-concrete deck (see [Figure 2.27a](#)). Numerical and experimental studies developed by the Laboratory of Vibration and Structural Monitoring (ViBest) of FEUP showed that this slender footbridge would be prone to excessive lateral and vertical vibrations caused by groups or streams of pedestrians [90]. ViBest installed six vertical TMDs, each one with a mass of 2465 kg (see [Figure 2.27b](#) and [2.27c](#)), adding a total of 14790 kg (mass ratio of 7.3%) [34]. In this structure, the mass of pedestrians corresponding to a density of 1.3 people/m² leads to a decrease in the natural frequency at the end of construction from 0.86 (bridge unloaded) to 0.79 Hz. Thus, the frequency tuning of the TMDs was 0.83 Hz, associated with the presence of 0.5 people/m².

The Taylor Devices company installed eighteen TMDs for three new footbridges in Las Vegas (Nevada), each one with 44, 48 and 50 meters span, respectively.

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(a) General view.

(b) TMD view.

Figure 2.26: TMDs in Forchheim Footbridge (2003). MAURER company.

(a) General view.

(b) TMD view 1.

(c) TMD view 2.

Figure 2.27: TMDs in Pedro e Inês Footbridge (2007) [34].

The site is on the famous Las Vegas “Strip” and new construction was expected to increase pedestrian traffic across the intersections associated with each bridge. The resulting design produced first and second vertical frequencies ranging from 1.48 Hz to 1.9 Hz for the three bridges. Each TMD required a moving inertial mass of 1 360 kg (see [Figure 2.28a](#) and [2.28b](#)).

After examining the examples of TMDs in footbridges, it is even more evident that, in order to deal with the uncertainties in the modal parameters as well as with

Figure 2.28: TMDs of one of the footbridges in Las Vegas (2007) (Taylor Devices company).

the existence of several problematic modes, in most cases, it is necessary to install more than one TMD, making the passive solution more expensive. These are where the semi-active vibration control with inertial controller (STMD) becomes more feasible as it is able to reduce the number of TMDs needed.

2.5 Semi-active Tuned Mass Dampers

Over the past few decades, modern control techniques have been applied to various TMD structures to improve the usable operating range and to make TMDs less sensitive to parameter variations. Most of the proposed STMD structures rely on damping control, as it is easier to implement than stiffness or mass control. Thus, the variation of TMD dynamics is made to be so fast that a TMD can alter its properties several times during its dominant vibration cycle, and the resulting TMD is known as STMD [26, 46, 50, 79, 114, 148, 170, 172, 173, 197, 198].

Generally speaking, an STMD is a TMD with changeable damping and/or stiffness elements through a real-time controller [78, 127]. An important aspect of STMD is the ability to be tuned to a wider range of frequencies in comparison with passive TMDs [172]. Many studies show that STMDs are more effective than conventional TMDs in the presence of off-tuning issues [110, 135, 197]. By their very definition, STMDs do not have the disadvantages that AMDs do; they require incomparably less energy and are characterised by inherent passivity, which directly implies their inherent stability. STMDs typically comprise a mass, a passive spring and a controllable damper, for instance, a variable-orifice damper or MR damper. Significant progress in the development of STMDs has taken place in recent years and this can be largely attributed to the availability of MR dampers, which have gained recognition in structural control due to their beneficial features i.e., a short response time and large dynamic force range [112, 128].

In the practical implementation of these devices, if the control law requires the dominant frequency of vibration to be known in order to set its behaviour, it can be determined, for instance, on the basis of the relative motion of the STMD. In the case of the harmonic vibration of an assumed unknown frequency, or in the case of non-harmonic vibration, the semi-active control force can be calculated from a variety of control algorithms. For example, the clipped Linear Quadratic Regulator (LQR) [159] or the clipped Linear Quadratic Gaussian control [168], various groundhook control approaches [111, 190], bang-bang control [206], optimal fuzzy logic control [105], PID (Proportional, Integral and Derivative) control [65], or the recently developed on-off phase control algorithms [50, 79, 139] can be found, amongst others. Groundhook and skyhook are actually modalities of phase control. Hereinafter, the most significant works using simple laws based on phase control are described, since this has been the approach in this thesis.

Koo et al. [109, 110] tested four algorithms to control an STMD with a linear variable damping: 1) DBG (Displacement Based Groundhook) control, an idea initially proposed by Hrovat et al. [87]. In this type of control, the damping coefficient of the semi-active damper is obtained from the product of the structure displacement multiplied by the relative velocity of the TMD. Two versions of this algorithm are presented: a) an on-off DBG: The damper takes its maximum or minimum value depending on whether the product signal is positive or negative, respectively; b) a Continuous DBG: The variation of the damping coefficient is proportional to the structure displacement; 2) VBG (Velocity Based Groundhook) control, in which the damping coefficient of the semi-active damper is obtained from the product of the structure velocity multiplied by the relative velocity of the TMD. Also, two versions of this algorithm are presented: a) an on-off VBG: The damper takes its maximum or minimum value depending on whether the product signal is negative or positive, respectively; b) a Continuous VBG: The variation of the damping coefficient is proportional to the velocity value of the structure. The authors verified a low efficiency of the VBG algorithm in relation to the passive system. However, the DBG algorithm guarantees a higher efficacy than VBG with the ability to improve substantially the vibration response obtained with the passive system. They also concluded that the continuous and on-off DBG algorithms presented very similar results. The explanation for this is that the optimal gain determined was very high, causing the continuous algorithm to have a similar behaviour to the on-off version.

Liedes [118] proposed a mixed algorithm between DBG and VBG, which was called DaVBG (Displacement and Velocity Based Groundhook), in which feedback is given based on a combination of the displacement and velocity of the structure. This method uses a weighted sum of displacement and velocity signals as a reference signal. Indeed, the feedback is nonlinear, which means that the optimal feedback matrix varies depending on the response frequency of the structure. This algorithm is based on the fact that the DBG algorithm is the one that contributes to reducing the high-frequency response. Liedes concluded that, in theory, it is possible to improve the DBG algorithm using this combination. It has been found, however, that the complexity of the practical implementation of

this system increases substantially, requiring a more complex observation system with a real-time vibration frequency detection system. A displacement sensor located between the primary structure and the fixed base is not feasible. However, the displacement of the primary structure could possibly be integrated from the acceleration of the primary systems. The integration process typically involves signal processing, which may induce errors in the signal. Consequently, further examination and development is needed.

In 2013, Chung et al. [50] presented an STMD ruled by a phase control algorithm, thus reducing the structural response through a control action that guarantees that the velocity of the TMD and the displacement of the primary structure are in opposing phases. Different researchers [79] reached the same STMD dynamics by using different control formulations, to be precise: the displacement-based ground-hook presented in 2002 by Setareh [172] and the clipped-optimal LQR presented in 2002 by Occhiuzzi et al. [148].

Ferreira [77] studied three high potential semi-active systems and only STMD proved to be able to largely overcome the efficiency of a TMD for the control of harmonic motion, mainly due to its ability to reduce system sensitivity to detuning. Ferreira et al. [79] also studied the dynamics of the phase control law applied to STMDs, including other practical application features, such as the analysis of the time delaying effect and locking damping. They also proposed an optimum tuning to reduce not only the structural response, but also the TMD stroke, which is a common limiting factor in civil engineering structures. A tuning method for this system should be studied in detail, since its behaviour is different from the TMD, whose tuning process is well established in the literature.

The above-mentioned algorithms differ in terms of calculation complexity, effectiveness, the purpose of generated semi-active control force and feedback signals used. Despite using different formulations and even different sensor inputs (acceleration, velocity and displacements) the practical control effect in the system dynamics is the same. This research work has focused on those strategies of semi-active control that can be materialised from a controllable smart damper (damping control), usually an MR damper.

2.5.1 Examples of vertical STMD in bridges

One of the most iconic examples of the application of an STMD on a bridge was carried out by Weber [196] on the Volgograd Bridge (Russia) in 2011. The Volgograd Bridge (see Figure 2.29a), which is 7 km long, is one of the longest bridges in Europe. It was inaugurated in October 2009, after 13 years of construction. This structure became world-famous in May 2010, when the bridge deck began to suffer strong oscillations, in the order of ± 40 cm. The bridge has several vibration modes that can be excited and these characteristics suggested that STMD solution would be the best to study it. The study was carried out by the MAURER

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(a) General view of the bridge.

(b) Frequency controlled tests of STMD.

(c) Damper view.

(d) Final functional check.

Figure 2.29: STMDs in the Volgograd Bridge (2011) [196].

company and the Universität der Bundeswehr München. Field measurements and wind tunnel tests demonstrated that the first three vertical bending vibration modes at 0.45, 0.56 and 0.68 Hz should be mitigated. The problem was dealt with by three groups of 4 STMD with an inertial mass of 5 200 kg each (a total number of 12) installed in 3 different sections of the bridge. Weber proposed an MR damper that emulates stiffness and damping variations that reduces the TMD sensitivity to detuning (see Figure 2.31b). According to MAURER, it would require 36 passive TMDs to reach the efficiency level verified with the 12 STMD installed.

In the pedestrian bridge of Forchheim show in Figure 2.26, a study concluded that the structure was susceptible to modifications of its dynamic properties due to the mass variation on the bridge. It was decided to upgrade the passive device into STMD to control a wider frequency range and thereby increase its efficiency. The control algorithm uses the clipped-optimal LQR law [148]. Occhiuzzi et al. [148] proposed a semi-active algorithm in which the feedback is determined by the clipped-optimal LQR formulation, having concluded that the optimal LQR feedback converges to the DBG algorithm. This is explained by the fact that the DBG algorithm is the one that optimally corrects the phase opposition between the TMD and the structure. The semi-active control device demonstrated a higher vibration reduction than the passive TMD, especially when the structure presents variations in its dynamic properties [170].

The work developed by Moutinho et al. [140] summarizes the study, the experimental analysis and the installation of a passive TMD in the FEUP Footbridge shown above in Figure 2.11a, as well as its upgrading to a semi-active system. Because the structure has several critical vibration modes with natural frequencies close to 2 Hz and also with uncertainties associated to these modal parameters (see Figure 2.11b and 2.11c), a single device cannot act efficiently on all of them, which limits the control efficiency. Upgrading the TMD to an STMD was the option selected to increase the robustness, given the self-tuning capabilities of these devices, as well as the possibility of performing multimode control. As a continuous dynamic monitoring system had been installed in 2009, it was possible to evaluate the structure's vibration levels measured over the previous few years, including the periods prior to and after the installation of the control solutions. A tendency was noted in the reduction of the peak accelerations of some sections of the deck after the installation of the passive TMD, which were even more attenuated with the activation of the semi-active system.

Many semi-active vibration control laws have been proposed; however, most are not geared to implementation and/or important practical issues are neglected. This thesis concentrates on two control laws with clearly developed thinking in their practical implementation. The selected laws are those used in the works of Weber [196] and Moutinho [139], respectively. The semi-active strategy used in the Forchheim Footbridge example converges to the DBG phase control algorithm, from which the Moutinho law comes, and that is why it is not studied in detail here. The law presented by Moutinho [139] will be called “*Semi-active control 1*”

Figure 2.30: STMD in FEUP Footbridge (2017) [140].

and is an on-off phase control (see Figure 2.31a). This law stands out for its simplicity. On the other hand, the law presented by Weber [196], “*Semi-active control 2*”, is a strategy in which the control force is continuously modified (see Figure 2.31b). Both laws are explained in more depth in the following sections and are studied numerically in Chapter 4.

Figure 2.31: Scheme of the two semi-active strategies studied. Red symbol ($\nearrow$) means changing over time.

2.5.2 Semi-active control 1

The phase control presented in [50, 180] and adapted by Moutinho [139] is described here. This is clearly geared to practical implementation due to the measured real-time parameters employed: the structure acceleration instead of displacement and the inertial mass velocity instead of the relative velocity. This control law will thus be effective when structure velocity is negligible with respect to the inertial mass velocity, which usually happens in civil engineering structures under resonant events. The lowest structural response is achieved when the velocity of the inertial mass and structure acceleration have opposing phases, so the semi-active device objective is to bring the inertial mass motion as close as possible to this phase. This is equivalent to a phase lag of 90° between the structure acceleration and the control force. Figure 2.31a shows an STMD (hereafter denoted as STMD 1), whose damping can be changed continuously. Thus, the adopted control law that originally assumes a controllable, ideal viscoelastic damper is defined as follows:

$$\begin{cases} \ddot{x}_s \cdot \dot{x}_T \leq 0 & \Rightarrow c_T = c_{min} & \text{(normal functioning),} \\ \ddot{x}_s \cdot \dot{x}_T > 0 & \Rightarrow c_T = c_{max} & \text{(blocking functioning),} \end{cases} \quad (2.15)$$

in which c_{max} is the maximum damping coefficient to block the TMD inertial mass movement, c_{min} is the optimal damping obtained from Equation 2.14, $\ddot{x}_s$ is the structure acceleration (recorded by an accelerometer) and $\dot{x}_T$ is the absolute velocity of the inertial mass (which might be obtained from the integration¹ of an accelerometer signal installed on the inertial mass controller).

2.5.3 Semi-active control 2

Figure 2.31b depicts the STMD strategy proposed by Weber [196] and denoted as STMD 2 hereafter. The key feature of this control law is that the energy dissipation and natural frequency of the STMD, respectively, are adjusted by changing the damping force continuously according to Den Hartog's formula in real-time to f_w instead of f_s , f_w being the actual instantaneous frequency of the vibration response, independently of whether this frequency corresponds to the resonant frequency of the targeted vibration mode or represents an excitation frequency due to forced excitation out of resonance. The damper works in parallel with a virtual passive spring, designed according to Den Hartog's formula (see Figure 2.31b) and denoted by k_T .

The damper emulates the sum of a controllable stiffness force (positive or negative), F_{κ_D} , and a controllable dissipative force, F_{c_D} , as follows

$$F = F_{\kappa_D} + F_{c_D}. \quad (2.16)$$

¹This and other aspects of the implementation are described in greater depth in Chapter 5.

2. Vibration control

It is a real-time adaptive control law that aims to tune the controller to the vibration frequency at every instant. However, these forces are clipped to zero (ideally or at the minimum frictional force, $F_{\min}$) in order to make its operation dissipative. It yields to the semi-active force, F_D , as follows

$$F_D = \begin{cases} F, & \text{sgn}(\dot{x}_s - \dot{x}_T) \cdot F \geq 0 \\ F_{\min}, & \text{sgn}(\dot{x}_s - \dot{x}_T) \cdot F < 0 \end{cases}, \quad (2.17)$$

in which $\text{sgn}(\dot{x}_s - \dot{x}_T)$ is the sign of the relative velocity between the structure and TMD mass. The actual frequency, f_w , is estimated from the zero-crossings of the relative displacement, $x_s - x_T$, through the half period time, T_{half} . The actual frequency is then estimated as follows:

$$f_w = \frac{1}{2 T_{\text{half}}}. \quad (2.18)$$

Thus, the stiffness and energy dissipation of the STMD 2 are tuned to the actual frequency of vibration f_w by the modified mass ratio $\tilde{\mu}$, which depends on the frequency shift between f_w and f_s as follows

$$\tilde{\mu} = \mu \left(\frac{f_w}{f_s} \right), \quad (2.19)$$

and then the damper properties are derived as

$$k_D = k_s \frac{\tilde{\mu}}{(1 + \tilde{\mu})^2} - k_T, \quad (2.20)$$

$$\zeta_D = \sqrt{\frac{3 \tilde{\mu}}{8 (\tilde{\mu} + 1)}}, \quad (2.21)$$

$$c_D = 2 \zeta_D m_T (2\pi f_w). \quad (2.22)$$

Finally, the control forces to compute (Equation 2.16) and apply the semi-active control law (Equation 2.17) are,

$$F_{K_D} = k_D (x_s - x_T), \quad (2.23)$$

$$F_{C_D} = \text{sgn}(\dot{x}_s - \dot{x}_T) \left\{ \frac{\pi}{4} c_D (2\pi f_w) (x_s - x_T) \right\}. \quad (2.24)$$

It should be mentioned that this algorithm requires the relative displacement to be measured, as well as knowing the sign of the relative velocity. This might be undertaken using a displacement transducer between the structure and the TMD mass.

2.6 Magneto-rheological dampers

2.6.1 Magneto-rheological fluids

Vibration control requires systems that can modify their properties faster than the dynamics of the system to be controlled. The control hardware is increasingly faster, since actuators with a reaction time as small as possible is required. For this reason, traditional materials and fluids are being replaced by smart materials and fluids. The most common example of smart fluids is the MR fluids, which were discovered and applied for the first time in the 1940s by Rabinow [161] at the US National Center of Standards in 1948. These fluids modify their rheological behaviour when they are crossed by a magnetic field [102, 149, 212].

They are a colloidal suspension, where the continuous phase is mineral oil and the dispersed phase is formed by high purity iron particles, whose size is just a few micrometres and in a volumetric fraction from 20 % to 50 %. When the particles are traversed by a magnetic field, the formation of dipoles is induced, which in turn allows the particles to form chains parallel to the field lines (see Figure 2.32), which restricts the movement of the fluid and increases the viscosity. Figure 2.32 shows MR fluid particles (Lord Corporation) in the liquid phase (2.32a) and the solid or semi-solid phase (2.32b). An applied input current varies the magnetic field to increase or reduce the MR fluid's viscosity in terms of magnetic flux density. The greater the intensity of the applied magnetic field is, the greater the energy necessary to break these chains and, consequently, reduce the viscosity is. This rupture delimits two characteristic regions of these fluids, the pre-creep region where the chains are formed and the post-creep region where the chains have already been broken. Several studies have characterized both regions [24, 52, 153], but also other aspects such as the shape of the particles, which affects the durability and abrasion [63], or the type of mineral oil, which also influences their behaviour, as proved by Lim et al. [119] and Park et al. [154].

Figure 2.32: MR fluid particles (Lord Corporation).

Carlson and Jolly [38] treated the wide variety of applications that this type of fluids has. Because its response time is less than 50 ms and energy consumption is low, its main use is in shock absorbers [208] (as dampers are usually called when they are applied in vehicles), which are present in a wide range of applications.

One of the main applications of MR dampers, as well as its main source of development, is for their use in the suspension of vehicles to improve damping against irregular road surfaces (see Figure 2.33). The traditional hydraulic damper is substituted by magnetic suspension, shock absorbers, smart suspensions etc., which are terms used to refer to MR damper-based suspension. In an MR suspension, controllable MR fluid replaces the traditional hydraulic fluid in each MR damper or, as it is known in this field, shock absorber. As sensors monitor road and vehicle conditions, a controller modifies the damping characteristics up to one thousand times per second. First introduced in 2002, the system now appears in more than a dozen car models. They are also in a wide range of vibration control applications in civil engineering as elements of dissipation in buildings and bridges, mainly for earthquake excitation [74, 86, 103, 183].

Figure 2.33: Automotive suspension systems (Lord Corporation).

Many publications related to the modelling of MR dampers can be found in the literature [102, 149, 212], indicating that MR dampers still have many issues to be solved, either in terms of precision or in terms of the complexity of the model. In general, characterization, modelling and validation are carried out with tests at medium speed ranges (between 15 and 100 mm/s) [151] and, therefore, models at low and high velocities do not accurately reproduce the MR response. In addition, some aspects are often neglected, such as friction, rigidity or secondary dependencies of the hysteresis, which might have implication for control strategies. Generally, MR damper models can be classified into two groups, depending on the way the model is obtained: parametric/phenomenological and non-parametric/black-box models. Parametric models usually make use of such physical concepts as friction, viscosity and hysteretic behaviour, which help to describe the dynamics of the device. On the other hand, non-parametric models, such as black-box models, make use of such soft computing techniques as neural networks, machine learning

and fuzzy logic, to build a model based on experimental information. Although they have been widely used recently, their use loses knowledge of the operation of the modelled device. Therefore, the work developed in this thesis only focuses on two phenomenological models, one without hysteresis, Bingham [22], and another with hysteresis, Bouc-Wen [25, 203], as the first approach to the problem of modelling MR dampers.

2.6.2 Modelling of MR dampers

MR dampers are highly non-linear devices by their inherent nature. Their force-velocity relationship exhibits a hysteretic behaviour which depends on several factors. Thus, the hysteretic behaviour of an MR damper is current dependent, but also a function of the amplitude and frequency content of the excitation. Hysteresis can cause serious problems in controlled systems, such as instability and loss of robustness. The main challenging problem regarding MR damper numerical modelling is the accurate inclusion of the characteristic non-linear nature of these devices into the model. In the literature, different models of MR damper have been proposed. Figure 2.34 shows the most common variables of the parametric models. F is the force of the MR damper, although many models usually refer to the effective force $F - F_0$, that is, the force that results after subtracting the force contribution when the piston rod is in mid-stroke (F_0). The variable x is the displacement of the piston from the mid-stroke position, where positive values indicate the contraction of the piston. c_0 and c_∞ are damping constants that linearly define the force-velocity curves in the pre-creep and post-creep regions, respectively. The border between both regions is marked by the transition speed $\dot{x}_y$, while F_y refers to the yield strength that the chains of iron particles exert in the Bingham model (the origin of the coordinates of the line c_∞). For those models that describe the hysteresis, $\dot{x}_h$ is the velocity where the force is zero.

Among the studied parametric models, the Bingham and the Bouc-Wen based models are the most widely adopted phenomenological models. This work has focused on these models. Figure 2.35 shows the Bingham and Bouc-Wen mechanical representation.

Figures 2.36 and 2.37 show the response of the Bingham and Bouc-Wen MR damper models using the parameter model proposed by Cesar and Barros [43] for the sponge RD-1097-1 MR damper (see Figure 2.38) and a load excitation of 1.00 Hz sinusoidal with 5 mm amplitude and different input voltages. This damper has a sponge as absorbent matrix and similar ones may have an open-celled foam or fabric saturated with an MR fluid. The sponge keeps the MR fluid located in the active region of the device where the magnetic field is applied. The device is operated in a direct shear mode with a minimum volume of MR fluid. A current controller is needed. Figure 2.36 shows the signal input excitation (2.36a), the time history force response of Bingham (2.36b) and Bouc-Wen (2.36c) models, respectively. Figure 2.37 shows the damper force against the displacement and

Figure 2.34: Main parameters of an MR damper model [151].

Figure 2.35: Bingham and Bouc-Wen mechanical models [209].

velocity of the MR damper. This is the modelled damper and, its parameters are identified and then used for the experimental vibration control tests of MR-STMD in this thesis.

2.6.3 Bingham model

In 1922, Eugene C. Bingham proposed a mathematical model for visco-plastic materials that behave like a rigid body when subjected to low stresses and as a viscous fluid when they are subjected to high stresses [22]. The Bingham model has long been used to characterize MR dampers. This model consists of a Coulomb friction element placed in parallel with a viscous damper. A schematic of this model is shown in Figure 2.35a. The stress-strain behaviour of the Bingham visco-plastic model is used to describe the behaviour of MR fluids. In this model,

(a) Signal input excitation for the models.

(b) Time history of Bingham force.

(c) Time history of Bouc-Wen force.

Figure 2.36: Time response force of Bingham and Bouc-Wen MR-damper models for 1.00 Hz sinusoidal excitation with 5 mm amplitude and different input currents (0.00 A, 0.25 A and 0.50 A).

2. Vibration control

(a) Bingham force-displacement curves.

(b) Bouc-Wen force-displacement curves.

(c) Bingham force-velocity curves.

(d) Bouc-Wen force-velocity curves.

Figure 2.37: Response of Bingham and Bouc-Wen MR-damper models for 1.00 Hz sinusoidal excitation with 5 mm amplitude and different input currents (0.00 A, 0.25 A and 0.50 A).

(a) Scheme of MR damper [7].

(b) MR damper.

(c) Pieces of sponge MR damper [109].

Figure 2.38: Sponge RD-1097-1 MR damper from Lord Corporation company.

the plastic viscosity is defined as the slope of the measured shear stress versus the shear strain rate. Thus, for positive values of the shear strain rate, $\dot{\gamma}$, the total stress is given by

$$\tau = \tau_{\gamma} \operatorname{sgn}(\dot{\gamma}) + \nu \dot{\gamma}, \quad (2.25)$$

where τ_{γ} is the yield induced by the magnetic field and ν is the viscosity of the fluid. Based on this model, the Bingham model consists of a Coulomb friction element placed in parallel with a viscous damper. In this model, for nonzero piston velocities, $\dot{x}$, the force generated by the device is given by

$$F = F_c \operatorname{sgn}(\dot{x}) + c_0 \dot{x} + F_0, \quad (2.26)$$

in which c_0 is the damping coefficient, F_c is the frictional force, which is related to the fluid yield stress. An offset in the force F_0 is included to account for the nonzero mean observed in the measured force due to the presence of the accumulator. The Bingham model is a simple approach that can simulate the MR damper response, although without the typical hysteretic behaviour present in these devices. The advantage of this model stems from its simplicity as compared to the Bouc-Wen model [184].

2.6.4 Bouc-Wen model

This was initially formulated by [Bouc](#) [25] and subsequently generalized by [Wen et al.](#) [203]. This hysteresis model is one of the mathematically simplest yet effective models that can represent a large class of hysteretic behaviour. The Bouc-Wen model is extremely versatile and can exhibit a wide variety of hysteretic behaviour. A schematic of this model is shown in [Figure 2.35b](#). The force in this system is given by

$$F = c_0 \dot{x} + k_0 (x - x_0) + \alpha z, \quad (2.27)$$

where the evolutionary variable z for the hysteresis behaviour is governed by

$$\dot{z} = -\gamma |\dot{x}| z |z|^{n-1} - \beta \dot{x} |z|^n + A \dot{x}. \quad (2.28)$$

By adjusting the parameters γ , β and A of the model, the linearity in the unloading can be controlled, as well as the smoothness of the transition from the pre-yield to the post-yield region. In addition, the force F_0 due to the accumulator can be directly incorporated into this model as an initial deflection x_0 of the linear spring k_0 . The Bouc-Wen model predicts the force-displacement behaviour of the damper well, while also possessing force-velocity behaviour that more closely resembles the experimental data [184].

2.6.5 Application of MR for STMD

The research work of this thesis has focused on STMDs through MR dampers. The tuning of the semi-active device should be done differently from in a passive device. This is why the uncertainties of the structure and the non-linearities of the damper device are taken into account. Performance and optimisation analysis are carried out in [Chapter 5](#) to generate the appropriate parameters for the proposed semi-active device. The studies will be considered both analytically and experimentally. [Figure 2.39](#) shows the 2-DOF systems in which both control devices, [2.39a](#) and [2.39b](#), have an MR damper instead of linear viscous damper, which is why the variable to control the damping of the MR damper is I (constant value) for the passive case ([2.39a](#)) and $I(t)$ (time-varying value) for the semi-active case ([2.39b](#)). This is because the semi-active one (see [Figure 2.39b](#)) has the capacity to change the force of the MR damper in real-time and, with an adequate control strategy, to increase the vibration reduction of the structure.

The inertial controller physically works as a feedback control (see [Figure 2.40a](#)), but the semi-active control laws that have been studied in this thesis are, in themselves, feedforward strategies applied to the control of MR dampers (see [Figure 2.40b](#)).

Figure 2.39: Model of MR damper for control applications. Red symbol ($\nearrow$) means changing over time.

(a) Feedback control by TVA.

(b) Feedforward control strategy applied to MR damper.

Figure 2.40: Feedforward control strategy applied to MR damper.

3

Tracking modal parameters of a lightweight structure

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3.1 Introduction

This chapter describes the work carried out on a lightweight stress-ribbon footbridge including the vibration monitoring system, modal properties tracking and vibration serviceability assessment.

The main obstacle for a more widespread adoption of vibration monitoring systems is the cost associated to their installation. This chapter describes a successful attempt to reduce the cost of a monitoring system through the use of low-cost sensors to measure structural vibrations on a footbridge. Thus, a low-cost continuous vibration monitoring system using small MEMS (Micro Electro-mechanical Systems) accelerometers were developed and installed on *Pedro Gómez Bosque* (PGB) footbridge (Valladolid, Spain) in order to track its long-term dynamic performance. The system continuously measures: the acceleration (using 18 triaxial MEMS accelerometers distributed along the structure), the ambient temperature and the wind velocity and direction [62]. PGB footbridge is a singular steel-plated stress-ribbon footbridge with a single span of 85 m. The dynamic behaviour shows several low-frequency vibration modes with closely-spaced natural frequencies and low damping ratios.

A procedure to track vibration modes was programmed to carry out an automatic modal parameter estimation. This procedure has to cope with vibration modes with closely-spaced natural frequencies. In this case, sorting vibration modes using frequency criteria and removing outliers that lay out of intervals of frequency variation might not be adequate [123, 138]. This chapter describes the detailed procedure carried out to track vibration modes of PGB footbridge over time. The procedure uses three tolerances based on the Modal Assurance Criterion (MAC). This approach allows reliable monitoring of vibration modes that are maintained over time.

The chapter continues with the description of the structure and its monitoring system. Section 3.3 analyses the uncertainties associated to a single time history record. The output-only modal parameter estimation using three SSI based OMA techniques and different data blocks is carried out. Thus, uncertainties due to the selection of the SSI technique and the data block are quantified. The results of continuous dynamic analysis are described in Section 3.4. A tracking method to follow the evolution of the persistent vibration modes over time is proposed. The influence of the environmental/operational factors on the modal estimates is studied and correlation models are derived to remove the influence of these spurious factors. Finally, some conclusions are drawn.

3.2 The footbridge and its vibration monitoring

3.2.1 Structure description

PGB footbridge, sited in Valladolid (Spain), is a slender and lightweight structure that creates a pedestrian link over the Pisuerga River between a sport complex and the city centre (see Figure 3.1). This bridge, built in 2011, is a singular stress-ribbon footbridge born by a pre-tensioned catenary-shape steel band with a single span of 85 m that provides minimal impact on the surroundings. The

(a) Landscape view.

(b) Distribution of sensors.

Figure 3.1: *Pedro Gómez Bosque* Footbridge, Valladolid (Spain).

structure mainly consists of a Corten steel band of 94 m long, 3.6 m wide and only 30 mm thick which is pre-tensioned and anchored to the two abutments. The complete steel band is fabricated by welding 8-meter long sheets and a number of 110 precast concrete slabs lay on the steel band [143]. These slabs do not have bearing capacity in such a way that the only structural element is the band. The structure is completed by rubber flooring and a stainless steel glass handrail.

3.2.2 Monitoring system

A structural vibration monitoring system was devised in order to continuously estimate the modal parameters of the structure and to assess their changes under varying environmental conditions. Therefore, apart from the accelerometers needed to perform a modal analysis, sensors for the wind and environmental temperature conditions were installed. The monitoring system comprises 18 triaxial accelerometers, 9 at each side of the deck, a temperature sensor and an anemometer with a vane. Wires and acceleration sensors were installed inside the handrail so the structure aesthetic was not modified in any way. This fact introduced additional complications: (i) the installation process was a laborious task, and (ii) additional angular transformations are required to obtain the acceleration vector in the structure axes for each accelerometer.

The vibration sensor used for the monitoring system was the low-cost MEMS accelerometer ADXL327 (ANALOG DEVICES) able to measure the static acceleration of gravity. The ADXL327 is a very small, low power, 3-axis accelerometer with signal condition voltage output. The key properties of this sensor are: measurement range up to $\pm 2.5g$, sensitivity up to 500 mV/g and bandwidth up to 550 Hz . However, this sensor is not designed to transmit the signal over long distances. To overcome this problem, an ad-hoc conditioning circuit was designed to enhance its long-distance performance. First, three capacitors, one to each channel, were placed to fix the frequency bandwidth to 100 Hz . Second, since the accelerometer has to be supplied by 3.6 V to get its maximum nominal sensitivity of 500 mV/g , the power supply unit of 12 V and a voltage regulator to 3.6 V were integrated into each circuit board in order to avoid power losses by the long distance wires. Thirdly, an operational amplifier was used to reduce significantly the output impedance, achieving thus a good signal-to-noise ratio for acceleration parameter (constant sensitivity and low noise). The achieved signal-to-noise ratio was $25 \mu\text{g}/\sqrt{\text{Hz}}$, which was considered to be enough for monitoring the structural vibrations. Finally, each circuit board with all its components was covered with a plastic coating to protect it from environmental conditions. Then, the accelerometers were ready to be installed inside the tube of the handrail. The sensing system was completed by a temperature sensor (model T0110 transmitter of Comet) and a wind sentry (model 03002L of R.M. Young Company) to measure the speed and direction of the wind. The temperature sensor and wind sentry were installed on the public light tower sited closed to the structure (see [Figure 3.1b](#)).

The monitoring system then comprises 57 voltage channels that are processed continuously. The data logger CompactRIO 9076 (National Instruments, NI) with two NI 9205 with 32 analog input channels is used for real-time data acquisition. The frequency sampling for each channel was chosen to be 200 Hz , enough to identify the modal parameters of the structure and to avoid aliasing problems during the post-processing. [Figure 3.2](#) shows the data logger and the two sets of wires from the NI cards, among other monitoring devices.

Figure 3.2: Data logger, router and other monitoring devices [62].

The actual orientation of the accelerometers installed inside the handrail is unknown (see Figure 3.2a). However, the Euler angles between the accelerometer coordinate system and the structure coordinate system (see Figure 3.3) can be derived taking into account the following: (i) the longitudinal axis of each accelerometer matches with the longitudinal axis of the footbridge, and (ii) the accelerometers are able to measure acceleration due to gravity from 0 Hz (static acceleration). Therefore, the transformation matrix between both coordinate systems is obtained, and then, the acceleration in the global axes can be finally calculated. The x -accelerometer axis remains in the xz absolute plane so there is no rotation about z absolute axis ($\gamma = 0$) which simplifies the initial expression [62]. The other two angles, α and β , can be calculated using a change of basis matrix $\mathbf{Q}$ between the absolute coordinate and the accelerometer coordinate systems using the following simplified expression:

$$\underbrace{\begin{pmatrix} a_{sx} \\ a_{sy} \\ a_{sz} \end{pmatrix}}_{\mathbf{a}_s} = \underbrace{\begin{pmatrix} \cos \beta & 0 & -\sin \beta \\ \sin \beta \sin \alpha & \cos \alpha & \cos \beta \sin \alpha \\ \sin \beta \cos \alpha & \sin \alpha & \cos \beta \cos \alpha \end{pmatrix}}_{\mathbf{Q}} \underbrace{\begin{pmatrix} 0 \\ 0 \\ -9.81 \end{pmatrix}}_{\mathbf{g}},$$

with $\mathbf{a}_s$ being the static acceleration vector measured in accelerometer axis and $\mathbf{g}$ the gravity acceleration vector. Thus, the acceleration of the structure in absolute axes are calculated as follows:

Figure 3.3: Fixed system (blue) and rotated system (red) [62].

$$\mathbf{a}_{\text{stru}} = \mathbf{Q}^T \cdot \mathbf{a}_{\text{dyn}},$$

with $\mathbf{a}_{\text{dyn}}$ the dynamic acceleration (removing the static part) measured by the accelerometers and $\mathbf{a}_{\text{stru}}$ the structure acceleration in absolute axes. Before its final installation, a laboratory validation was carried out and after the installation, an in-situ validation was performed by comparing against conventional piezoelectric accelerometers. The novelty of this system is that low-cost MEMS accelerometers properly conditioned were used resulting finally in a really cheap monitoring system.

3.3 Peered analysis of one test

The uncertainties associated to the modal identification of one test, corresponding to the record measured on 01/05/2013 at 19:16 (60-minute test), are analysed herein. The dynamic behaviour of the structure is mainly governed by the vertical response (previous time-history analyses have shown that lateral and longitudinal accelerations are small and negligible, respectively, as compared to vertical ones). That is the reason why this analysis concentrates on the vertical vibration. The process followed is, firstly, the signal processing undertaken before the modal identification is presented. Afterwards, the identification from the same test (60 minutes) is performed using different OMA techniques. Then, the same technique is applied to different 20-minute time windows of the 60-minute test.

3.3.1 Data processing

The vertical response of the structure for a 60-minute test is analysed (with an initial sampling frequency of 200 Hz). The signal is filtered by a low-pass Butterworth filter of order 4 with a cut-off frequency of 5 Hz. A decimation factor of 16 is applied obtaining a Nyquist frequency of 6.25 Hz. As an example, Figure 3.4a and b show the filtered and decimated response in time and frequency domain, respectively, corresponding to the four central channels for upstream side. Besides, Figure 3.4c shows the Spectrogram using the short-time Fourier transform for channel 15. Up to ten horizontal alignments below 4 Hz can be observed in this figure. Figure 3.5 and Figure 3.6 show the changes that occur when filtering and decimating the signal, both in time and frequency domain, respectively. High frequency noise and zero-frequency (offsets of signal) are measured in such a way that the filtered, decimated and detrended signals are used for OMA.

3.3.2 Operational Modal Analysis using three SSI techniques

Three OMA techniques based on the SSI [150] and programmed in MATLAB® are used for the same test. The techniques used are: covariance-driven SSI (*SSI-cov*) [155], data-driven SSI (*SSI-data*) [165] and expectation maximization SSI (*SSI-EM*) [37]. The same criteria to define a pole (solution) of the stabilization diagram as stable is used for the three identification techniques. The criteria to define a pole as stable has to fulfil three requirements against estimates of the previous state space order: (i) the frequency variation must match within 1 % (relative), (ii) the damping ratio variation must match within 5 % (absolute), (iii) the damping ratio must match within 5 % (maximum absolute value), and (iv) the mode shapes must match within 95 %, using the MAC for comparing. Additionally, modes with identified damping ratios higher than 5 % are also rejected.

Mathematically, the criteria are expressed as follows:

$$\begin{aligned} |f_i - f_j| / f_j &\leq \varepsilon_f \\ |\zeta_i - \zeta_j| &\leq \varepsilon_{\zeta\Delta} \\ \zeta_{i_{max}} &\leq \zeta_{max} \\ 1 - MAC(\phi_i, \phi_j) &\leq \varepsilon_{MAC}, \end{aligned}$$

where f , ζ and ϕ are frequency, damping ratio and modal shape respectively. MAC shows the degree of correlation between two vectors, such as mode shapes, and it is computed as [75],

$$MAC(\phi_i, \phi_j) = \frac{|\phi_i^T \phi_j|^2}{(\phi_i^T \phi_i)(\phi_j^T \phi_j)},$$

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Figure 3.4: Processed raw data of 4 central channels for upstream side.

(a) Green signal is the raw signal.

(b) Red signal is the green signal with removed trends.

(c) Blue signal is the filtered and decimated signal.

Figure 3.5: Signal processing effects in time domain.

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(a) The red signal is the raw signal with removed trend before filtering and decimation.

(b) Blue signal is the filtered and decimated signal.

Figure 3.6: Signal processing effects in frequency domain.

where the superscript T means transpose operator. Parameters ε_ζ , ε_f , $\varepsilon_{\zeta_\Delta}$, ζ_{max} and ε_{MAC} are tolerance limits to decide if pole i calculated is stable, j subscript referring to the other previous stable mode. Their values adopted have been $\varepsilon_f = 1\%$, $\varepsilon_{\zeta_\Delta} = 5\%$, $\zeta_{max} = 5\%$ and $\varepsilon_{MAC} = 5\%$ respectively for this case.

Parameter $\varepsilon_{\zeta_\Delta}$ could be defined in a relative way, like ε_f , but it is well known that the damping is estimated with greater variability than the natural frequencies [166], so it prefers to use $\varepsilon_{\zeta_\Delta}$ in absolute terms [37]. This means that if two modes identified from different starting point have similar natural frequencies and similar mode shapes, but with damping ratios differing in, for instance, 2%, it assumes that they correspond to the same mode.

All parametric system identification methods based on the state space model require to know the model order, n_s , which, in theory, is twice the number of identified modes. In OMA it is customary to estimate the state space model for a wide range of orders and it allows to represent the identified poles associated with their frequency represented in the x -axis against the model order in the y -axis, called a stabilization diagram. Experience on a very large range of problems shows that in such analysis, the eigenfrequencies corresponding to physical modes appear at most of the used model orders, while mathematical and spurious poles tend to scatter around the frequency range. Then, from such diagram it is possible to select the valid system poles. The implementation of this strategy can be executed by initially choosing a sufficiently high order for the state space model [36]. The problem arises when choosing a right modal order capable to have poles representing all the desired modes but adding mathematical solutions, with no physical meaning, that appear for high model orders. The information of the pole solution (frequency, damping ratio and modal shape) helps to discriminate other spurious solutions, as it can be a very high damping in a structure without extra damping elements. Also to separate different modes with close identified frequencies where the modal shape might change among them. In order to cope with this issue, the stabilization diagram becomes an interesting tool. From this diagram, is possible to identify the poles with high repeatability for several model orders in a given frequency, that are more probable to represent a real resonant vibration mode of the structure than the mathematical solutions that appears as single values for high model orders. The stable poles at different model orders gradually form stable columns and the system vibration modes can be obtained by selecting clear stable poles in stabilization diagram.

The SSI-*cov*, programmed in MATLAB®, has the advantage of its conceptual simplicity and the ability to compute the probability density function of the identified system parameters. It selects the average values of the modal parameters for each column (of stable aligned poles of the stabilization diagram) with a minimum number of stable poles [155]. The SSI-*data* has been applied using MACEC (Commercial Toolbox of MATLAB® for modal analysis, [165]) and has the advantage of an optimal statistical performance when the weighting matrices are properly chosen. A statistical analysis of the stable poles of the stabilization diagram is used to choose final results [166]. The SSI-*EM* is a combination of SSI and EM algorithm. Using SSI as initial estimate, the maximum likelihood estimation requires iterations when using the EM algorithm [37]. SSI-*EM* introduces the EM algorithm [130] to maximize the likelihood function. This optimizes the lack of optimal shown in [20] and [47] improving the SSI results but does not define criteria to choose the pole in the stabilization diagram. That is, the poles are those obtained by the modal order previously selected by an experienced user. Figure 3.7 shows the diagrams of stabilization obtained with each technique and the MAC matrix calculated between selected modes.

Figure 3.8 illustrates the final selected poles for the three techniques and the averaged normalized Power Spectral Density (PSD). From the above mentioned characteristics, the recommendations given in [205] and the author's experience,

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the use of several methods simultaneously is a good way to improve the results as well as to quantify the mathematical uncertainties associated to the OMA technique (see Figure 2.9).

Figure 3.7: Diagrams of stabilization obtained and MAC matrix.

Figure 3.8: Selected poles: SSI-cov (dashed lines), SSI-data (circles) and SSI-EM (crosses).

Table 3.1 shows the estimated natural frequencies and damping ratios. A MAC correlation between the modal shapes estimated by three techniques has been carried out. Modes that exhibit a MAC value greater than 0.95 for all of the cross values are highlighted in bold. Note that even the mode at 1.05 Hz is estimated by the three techniques, it is not finally selected due to the MAC discrimination parameter. Modes with natural frequencies over 4 Hz (0.8 times the cut-off frequency of the filter) are not included. Figure 3.9 shows an example of identified modal shapes corresponding to the lowest seven modes obtained with SSI-cov.

Table 3.2 shows the summary of the selected modes. The table shows the mean values, the maximum relative error of the natural frequencies and the maximum absolute error of the damping ratios. It is well known that the damping is estimated with greater variability than the natural frequencies [164], so it is better to use absolute errors for damping estimates. The maximum errors obtained are 0.6669% and 26.67% for frequencies and damping ratios, respectively. It is worth noting the very low values obtained for the damping ratios although its estimation uncertainty is high, as usual [137].

The purpose of this subsection is to quantify uncertainties from the same time history but employing three different SSI techniques to be sure that the comparison of modal parameters of vibration modes corresponds to actually estimations of the same vibration modes. The MAC value is usually used to compare experimental and numerical modes shapes, nevertheless, this value can be used to compare any

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Table 3.1: Natural frequencies (f) and damping ratios (ζ) identified by the three SSI techniques.

Mode	SSI- <i>cov</i>		SSI- <i>data</i>		SSI- <i>EM</i>	
	f (Hz)	ζ (%)	f (Hz)	ζ (%)	f (Hz)	ζ (%)
1	0.8671	0.3352	0.8670	0.1179	0.8671	0.1006
–	1.0548	0.9622	1.0508	0.6378	1.0526	0.6807
2	1.4077	0.3792	1.4089	0.4058	1.4135	0.6459
3	1.5408	0.6762	1.5418	0.6230	1.5511	0.7600
4	1.7869	0.4483	1.7864	0.3760	1.7884	0.1983
5	1.8571	0.6381	1.8568	0.5737	1.8590	0.3884
–	–	–	2.2977	0.3730	2.2928	0.5868
–	2.3020	0.4799	2.3041	0.4690	–	–
–	2.6819	0.3206	–	–	–	–
–	–	–	–	–	2.7838	0.6067
–	–	–	2.7949	0.4447	–	–
–	2.8051	0.5159	–	–	–	–
–	–	–	2.8157	0.2338	2.8190	0.4646
–	2.8682	0.3238	2.8679	0.1601	–	–
–	–	–	–	–	2.8828	0.4393
6	3.3674	0.4485	3.3684	0.4092	3.3687	0.4361
7	3.5375	0.8723	3.5369	0.7968	3.5407	0.9700
8	3.9487	0.5106	3.9487	0.4873	3.9413	0.5285

Table 3.2: Summary of identified modes and statistical comparison for the three techniques: mean frequency ($\bar{f}$), mean damping ($\bar{\zeta}$) and the corresponding errors.

Mode	Frequency		Damping	
	$\bar{f}$ (Hz)	Error (%)	$\bar{\zeta}$ (%)	Error (%)
1	0.8671	0.0115	0.2346	18.46
2	1.4100	0.4113	0.4770	26.67
3	1.5446	0.6669	0.6864	13.70
4	1.7872	0.1119	0.3409	25.00
5	1.8576	0.1184	0.5334	24.97
6	3.3682	0.0386	0.4313	3.930
7	3.5384	0.1074	0.8797	17.32
8	3.9462	0.1875	0.5088	4.120

Figure 3.9: First seven modes shapes obtained with SSI-cov.

couple of modal shapes, as it is usually done between the poles of the stabilization diagram when different modal orders are compared. Once it is completely sure that the comparison is dealing with the same modal shape, it compares the estimated modal parameters to fill Table 3.1. Finally, this table allows to quantify uncertainties for vibration modes that are simultaneously identified by the three techniques. Figure 3.10 shows values of MAC matrix for comparison between SSI-cov and SSI-data; it is observed the vibration modes which are identified by both techniques.

Figure 3.10: MAC comparison between *SSI-cov* and *SSI-data*.

3.3.3 Operational Modal Analysis using the same SSI technique

In order to track the modal parameters of a structure, it is recommended to use the minimum test duration that leads to a successful identification. As a rule of thumb, the minimum duration of the measurement should be at least 1000 cycles of the lowest natural frequency is expected to be identified [205]. This duration is usually recommended as the minimum one for noise-contaminated signals and close vibration modes. Therefore, this duration, which corresponds approximately to 20 minutes, has been adopted here to carry out the modal tracking.

Now, three consecutive windows of 20-minute data are considered to identify the modal parameters using only *SSI-cov*. These results are compared with those obtained using the 60-minute record. Table 3.3 shows the estimates obtained from the four data blocks. Using the same criterion as in the previous case, modes that exhibit a MAC value greater than 0.95 for all of the cross values are highlighted in bold. The average temperature for each time window was 16.42, 13.84 and 11.56 °C, respectively, and 13.94 °C for the 60-minute record.

Table 3.4 presents a summary of the selected modes. The table shows the mean values for the 20-minute tests and the maximum errors compared with the results obtained with the 60-minute test. In this case, the maximum errors obtained are 0.2855 % and 61.09 % for frequencies and damping ratios, respectively. The errors in natural frequency estimates using 20 minutes as compared with 60 are negligible.

Table 3.3: Natural frequencies (f) and damping ratios (ζ) identified by SSI-*cov* and different time blocks.

Mode	60 min.		1 st 20 min.		2 nd 20 min.		3 rd 20 min.	
	f (Hz)	ζ (%)	f (Hz)	ζ (%)	f (Hz)	ζ (%)	f (Hz)	ζ (%)
–	0.8671	0.3352	0.8663	0.2526	–	–	0.8660	1.0824
–	–	–	1.0251	0.3793	1.0243	0.1433	1.0281	1.8147
1	1.0548	0.9622	1.0536	0.3755	1.0518	0.3513	1.0535	0.4883
2	1.4077	0.3792	1.4073	0.3523	1.4059	0.5874	1.4093	0.2594
3	1.5408	0.6762	1.5408	0.5343	1.5395	0.7945	1.5431	0.7658
4	1.7869	0.4483	1.7884	0.5036	1.7882	0.6227	1.7833	0.4014
5	1.8571	0.6381	1.8567	0.6340	1.8574	0.6775	1.8569	0.4995
–	–	–	1.9260	0.3870	–	–	–	–
6	2.3020	0.4799	2.3022	0.4444	2.3020	0.5353	2.3014	0.4788
–	2.6819	0.3206	–	–	2.6802	0.2709	2.6836	0.2546
–	–	–	–	–	–	–	2.7306	0.4594
–	–	–	–	–	2.7848	0.3825	–	–
–	2.8051	0.5159	2.8097	0.7520	–	–	2.8041	0.4918
–	–	–	–	–	2.8147	0.1976	–	–
–	2.8682	0.3238	–	–	–	–	2.8631	0.3248
–	–	–	2.9171	0.2887	–	–	–	–
7	3.3674	0.4485	3.3680	0.4194	3.3659	0.4449	3.3709	0.4587
8	3.5375	0.8723	3.5356	0.6417	3.5361	0.9920	3.5343	0.8376
9	3.9487	0.5106	3.9443	0.4427	3.9458	0.5759	3.9552	0.3504

3.4 Continuous dynamic analysis

The procedure carried out to track the vibration modes is depicted in [Figure 3.11](#). The results obtained from 1-year of continuous dynamic monitoring are described from now on. Firstly, a method to track the evolution of the main vibration modes over time is described. Then, the influence of the environmental/operational factors on the modal estimates is studied and correlation models are derived to remove the influence of these spurious factors.

3.4.1 Tracking of modal properties

From the acquired data, an automated OMA has been implemented using SSI-*cov*. The procedure carried out to remove the environmental/operational effects is depicted in [Figure 3.11](#). The process can be divided into the following steps:

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Figure 3.11: Tracking procedure for modal parameters.

Table 3.4: Summary of identified mode frequencies and statistical comparison for the 20-minute against 60-minute time block: mean frequency ($\bar{f}$), mean damping ($\bar{\zeta}$) and the corresponding errors.

Mode	20 min. time windows			
	Frequency		Damping	
	$\bar{f}(\text{Hz})$	Error (%)	$\bar{\zeta}(\%)$	Error (%)
1	1.0530	0.2849	0.4050	61.09
2	1.4075	0.2416	0.3997	32.80
3	1.5411	0.2336	0.6982	26.02
4	1.7866	0.2855	0.5092	22.13
5	1.8570	0.0377	0.6037	17.80
6	2.3019	0.0348	0.4862	9.090
7	3.3683	0.1484	0.4410	3.930
8	3.5353	0.0905	0.8238	35.05
9	3.9484	0.2761	0.4563	22.55

- Take the last N tests, N being the number of tests. N should be representative of the variability of the modal estimates.
- For each test, the OMA is carried out. Thus, a number of modes for each test is identified and denoted as r_i (with $i = 1 \dots N$).
- The objective is to find modal estimates that correspond to the same persistent mode over time. Each modal shape estimate is compared with all the remainder estimates within the N tests and a counter increases each time that two mode shapes match. The counter indicates the repeatability of a vibration mode. This is done for all the modal shape estimates. To carry out this step, the MAC value is used. That is, a number of $\left[\sum_{i=1}^N r_i \right]$ estimates are compared using the MAC value. Three tolerances have been defined for this purpose:
 - tol_1 : the first one is a MAC value that allows to group estimates corresponding to the same vibration mode. The repeatability is the number of estimates of a group.
 - tol_2 : the second one is included in order to reject groups that are not repeated sufficiently to be considered as significant modes. This tolerance is the lower limit of success ratio, that is, groups with low repeatability are automatically rejected.
 - tol_3 : the third one is a MAC value finally included to detect groups of estimates that are actually estimations of the same mode. Then, if more than one group correspond to the estimation of the same mode, the one with higher repeatability is selected.

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- From above procedure, the most significant modes of the dynamic response of the structure are detected for the N tests.
- Variations of these selected modes are statistically studied and corrected modal parameters are derived.

The above tracking method is applied to consecutive time-history records every 20 minutes using *SSI-cov*. A number of $N = 21\,643$ tests have been taken for the whole year 2013. The tolerances chosen to carry out the tracking were: $tol_1 \geq 0.95$, $tol_2 \geq 40\%$ and $tol_3 \leq 0.80$. Up to nine vibration modes below 4 Hz have been tracked. Table 3.5 shows the following statistics of the estimation: mean, standard deviation, absolute variation (%) and their repeatability (the success ratio is included between brackets). Note that the fourth mode at 1.79 Hz (with a damping ratio of only 0.42 %) corresponds to the highest success ratio.

Table 3.5: Summary of identified natural frequencies and damping ratios for one year monitoring and their statistics: mean frequency ($\bar{f}$), mean damping ($\bar{\zeta}$), standard deviation (Std) and the corresponding variation (ν).

Mode	Frequency			Damping			Repeatability
	$\bar{f}(\text{Hz})$	Std	$\nu(\%)$	$\bar{\zeta}(\%)$	Std	$\nu(\%)$	
1	1.0482	0.0152	14.23	0.3665	0.1710	147.89	9667 (44.7%)
2	1.4145	0.0107	35.26	0.3381	0.1513	110.74	10619 (49.1%)
3	1.5440	0.0181	27.63	0.6498	0.2357	133.62	9886 (45.7%)
4	1.7937	0.0291	20.27	0.4192	0.1502	221.88	13817 (63.8%)
5	1.8594	0.0168	6.87	0.5718	0.1605	234.74	9936 (45.9%)
6	2.3117	0.0425	15.01	0.3753	0.1474	128.54	8746 (40.4%)
7	3.3821	0.0549	42.95	0.3868	0.1191	103.96	12210 (56.4%)
8	3.5512	0.0524	51.87	0.7226	0.1884	157.48	9237 (42.7%)
9	3.9610	0.0624	8.95	0.3853	0.1185	230.82	10183 (57.8%)

It can be seen that the repeatability of modes is quite lower than the total number of tests. This is due to the fact that some of them are performed under very low vibration conditions leading to wrong estimates that, obviously, are not detected by tracking procedure. The pedestrian traffic over the structure during the day hours induces on the structure the necessary operational energy that allows successful identification. However, the lack of this excitation during the night hides the real structural response inside the signal noise. Figure 3.12 shows hourly distribution of successful identifications obtained for the fourth mode selected (see Table 3.5).

The low structural vibration conditions occur mainly at night. This is the main difference between day and night. However, success or failure in the estimation is not mainly due to the vibration amplitude, it is mainly due to frequency content excited due to the excitation. During night, sensors mainly measure noise and

Figure 3.12: Distribution per hour of the repeatability for mode 4.

while structural vibrations arise mainly during the day. [Figure 3.13](#) shows the time history and PSD results of two tests at day and night time corresponding with 15:00 and 3:00 hours respectively. Although, vibration levels are similar, the frequency contents show significant differences between both tests. It should be comment that the signal-to-noise ratio and the sensitivity of the conditioned sensors is not high (lower than the one usually used for standard OMA), anyway, when the structure is in use, structural vibrations arise and the monitoring system has success in the estimation. Remind that the monitoring system is much more cheaper than a conventional one.

The time variation of the natural frequency estimates over a year for the lowest ninth mode is shown in [Figure 3.14a](#). [Figure 3.14b](#) presents a zoom showing that the tracking method is able to identify two closely-spaced natural frequencies. Although the monitoring has been operating since January 1st 2013 up to now, some occasional stops due to minor technical problems and maintenance tasks are observed in the figure.

3.4.2 Effects of external factors

A strong seasonal and daily trends with temperature have been observed. [Figure 3.15](#) shows the time evolution of the frequency estimates of the fourth mode,

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Figure 3.13: Results at day and night test for channel 16.

in the left axis, and temperature, in the right axis. It is clearly observed the seasonal (Figure 3.15a) and daily trend of the estimation (Figure 3.15b). Figure 3.16 shows the frequency estimates versus temperature for mode 1 (Figure 3.16a) and mode 4 (Figure 3.16b) of Table 3.5. It is noted that pure vertical response behaves similar to the equivalent suspended cable as well as its thermal behavior [33]. Therefore, for these modes (such as mode 4), increasing temperature leads to an increase into the ribbon sag producing a reduction of band tension and leading to a decrease in their natural frequencies [64]. However, mode 1 (of Table 3.5 and Figure 3.9b) does not follow this pattern since this is a torsional mode.

No other visual evidence of the correlation between the frequency estimates and other factors, different from the temperature, has been found (neither with the wind velocity nor the operational values). Regarding damping ratios, no clear visual dependencies with any external factor have been found. Therefore, a proper statistical analysis will help in seeking correlations.

(a) Complete view.

(b) Zoom view of the fourth and fifth vibration mode.

Figure 3.14: Tracked frequency estimates for the whole year.

Figure 3.15: Frequency estimates and temperature recorded for mode 4.

3.4.3 Statistical analysis

A correlation analysis has been carried out in order to study the influence of the external effects on the natural frequencies and damping ratios. The technique employed to study the dependence of a dependent variable y with other independent variables or predictors has been the Multiple Linear Regression (MLR). Generally, the equation of MLR can be written as

$$y = \mathbf{Z}\boldsymbol{\beta} + \varepsilon = \hat{y} + \varepsilon, \quad (3.1)$$

$\mathbf{Z} = [1, Z_1, \dots, Z_k]$ is a row vector of $(k+1)$ terms that contains the k predictors, $\boldsymbol{\beta} = [\beta_0, \beta_1, \dots, \beta_k]'$ is a column vector of $(k+1)$ coefficients to be determined

Figure 3.16: Frequency estimates versus temperature for mode 1 and 4 (from Table 3.5).

that weight the contribution of each predictor and ε is the residual error. The variable $\hat{y} = \mathbf{Z}\boldsymbol{\beta}$ is the estimate of the statistical model for the dependent variable y .

It is important to have an indicator to provide information of the goodness of the regression fitting. The most common one is the Coefficient of Determination or R-Squared which is defined as,

$$R^2 = 1 - \frac{\sum_{i=1}^n (y_i - \hat{y}_i)^2}{\sum_{i=1}^n (y_i - \bar{y})^2} \quad 0 \leq R^2 \leq 1,$$

where $\bar{y}$ is the mean of the n observations, y_i and $\hat{y}_i$ are the observation and the estimate of i -th. This indicator represents the proportion of variation of dependent variable explained by the regression. Always R-Squared coefficient increases when it includes new independent variables in the statistical model, even when they are insignificant or have little correlation with the dependent variable. The Adjusted R-Squared solve this problem, because it takes into account the number of observations and predictors included in the model [53]. For a MLR model, this is defined in terms of R-Square as follows,

$$R_{adj}^2 = 1 - (1 - R^2) \frac{n - 1}{n - k - 1} \quad 0 \leq R_{adj}^2 \leq 1,$$

where n is the number of observations in the data set, and k is the number of independent variables. This coefficient represents the proportion of variation of

the dependent variable explained by the regression model. The increase in the Adjusted R-Squared by each additional predictor indicates the improvement provided by this predictor.

The independent variables have been considered as function of the ambient temperature, structure acceleration and wind speed. Thus, the statistical model adopted consists of linearly fitting the independent variables (that might be non linear functions of environmental/operational measurements) which are selected according to its statistical relevance. Furthermore, temperature dependence is also explored not only at current test, but also on the temperatures observed in previous tests. That is, a dynamic regression model is adopted regarding the temperature dependence.

The estimates have been considered as function of the following variables

$$\hat{y} = g(T, (\Delta T/\Delta t)_i, T_{R_j}, a_{max}, a_{RMS}, VDV, V_{max}, V_{RMS}), \quad (3.2)$$

where $\hat{y}$ is the modal estimate, T is the temperature associated of each test (the mean temperature of the test is taken), $(\Delta T/\Delta t)_i$ is the gradient of temperature between the current test and the i -times former test, T_{R_j} is the running averaged temperature using the j previous tests, a is the vertical acceleration and V is the wind velocity. Variables a_{RMS} and VDV are measures suitable to asses the vibration serviceability for pedestrian structures [95]. Contrary to peak and RMS accelerations, VDV accumulates the vibration effects rather than averaging them and increases with duration. Equation 3.2 according to Equation 3.1 is considered as

$$\hat{y} = \beta_0 + \sum_{r=1}^k \beta_r Z_r, \quad (3.3)$$

in which $Z_r \in [z_r, z_r^2, \dots, z_r^n, 1/z_r, 1/z_r^2, \dots, 1/z_r^n]$, exploring thus nonlinear relationships, with z_r being the independent variables that appear in Equation 3.2, the coefficients β_r are obtained from the linear-least square solutions and k the number of predictors for the model.

In order to extract the corrected estimates and remove the spurious effects, model of Equation 3.3 is rewritten as follows

$$\hat{y} = \underbrace{\left(\beta_0 + \sum_{r=1}^k \beta_r \overline{Z_r} \right)}_{y^*} + \sum_{r=1}^k \beta_r (Z_r - \overline{Z_r}), \quad (3.4)$$

where $\overline{Z_r}$ is the mean value of all the observations of Z_r and y^* is the corrected estimate, defined as

$$y^* = \beta_0 + \sum_{r=1}^k \beta_r \overline{Z_r}. \quad (3.5)$$

The MLR presented above has been applied to frequency and damping ratio estimates for all of the tracked modes (of Table 3.5). It is considered $i = 1, 2, \dots, 30$, $j = 50, 100, 250, 1000, 1500, 2000$ (see Equation 3.2), and $n = 3$ and $k = 4$ (see Equation 3.3 and definition of Z_r). Eventually, it has been considered 252 candidates for predictors. Thus, Table 3.6 shows the values of the Adjusted R-Squared for all the natural frequency estimates of the selected modes as the number of predictors increase in the statistical models. The predictors of those models that obtain a value of the Adjusted R-Squared greater than 0.80 are also shown in Table 3.6. The last column of the table is the maximum Adjusted R-Squared, R_{max}^2 , obtained for the maximum number of predictors, k_{max} , for each model (which is shown between brackets). It is observed that those modes with higher Adjusted R-Squared (modes 4, 6, 7 and 9), $R_{max}^2 > 0.90$, have independent variables related to the temperature as predictors. It is also observed that there are two modes, 1 and 2, both with Adjusted R-Square close to 0.80, in which serviceability parameters are used as predictors 3^{rd} and 4^{th} for their respective models (see Table 3.6).

The frequency distribution for each mode is shown in Figure 3.17. There are some modes (3, 5 and 8) whose frequencies have a very narrow distribution, indicating that: (i) they do not change significantly, (ii) they do not need any correction and (iii) they might be already used for SHM. Therefore, the variation of these modes can not be explained, leading thus to Adjusted R-Squared lower than the remainders. Figure 3.18 shows the damping ratio distribution for each mode. It can be observed that the damping of these modes of the structure are most of them very low (mean value less than 0.5 %), except for modes 3, 5 and 8, for which its mean value is greater than 0.5 %. These modes are the same which also have a very narrow distribution frequency.

Finally, very low values for the Adjusted R-Squared (smaller than 0.20) were achieved for damping ratios. Therefore, the results for the damping ratios are not presented here because there is no clear dependency of damping ratio variations.

3.4.4 Removing external factors

As an example, the model obtained for the natural frequency estimate of mode 4 is expounded in detail. This is the mode with highest success ratio and whose model achieves the highest Adjusted R-Squared. The model obtained for the statistical frequency is the following,

Figure 3.17: Overlaid distributions of the identified natural frequencies.

$$\hat{f} = \beta_0 + \underbrace{\beta_1 \cdot T + \beta_2 \cdot (\Delta T / \Delta t)_{19} + \beta_4 \cdot T^2}_{\text{Daily}} + \underbrace{\beta_3 \cdot T_{R_{1000}}}_{\text{Seasonal}}, \quad (3.6)$$

where $\hat{f}$ is the frequency estimate by statistical model. Equation 3.6 shows the dependency on the temperature, on the temperature gradient with respect to the 19-times former test ($\Delta t_{19} = 19 \cdot 20 = 380$ minutes), the running averaged temperature using 1000 previous tests and the temperature to the second power. These values have provided the higher increment into the Adjusted R-Squared. Additional predictors introduce only marginal improvements into the correlation.

In order to test this statistical model, the first 20 days of January 2014 are used as “unseen data”. Figure 3.19 shows the frequency estimates by the SSI-cov technique and by the statistical model of Equation 3.6. The maximum relative error did not exceeded 1 %.

As stated in Equation 3.5 and according to Equation 3.6, the corrected frequency is as follows

$$f^* = \beta_0 + \beta_1 \cdot \bar{T} + \beta_2 \cdot \left(\overline{\Delta T / \Delta t} \right)_{19} + \beta_3 \cdot \bar{T}_{R_{1000}} + \beta_4 \cdot \bar{T}^2. \quad (3.7)$$

Figure 3.18: Overlaid distributions of the identified natural damping ratios.

Considering Equation 3.7, Equation 3.6 can be rewritten as,

$$\begin{aligned} \hat{f} = f^* + \beta_1 \cdot (T - \bar{T}) + \beta_2 \cdot \left((\Delta T / \Delta t)_{19} - \left(\overline{\Delta T / \Delta t} \right)_{19} \right) + \dots \\ \dots + \beta_3 \cdot (T_{R_{1000}} - \bar{T}_{R_{1000}}) + \beta_4 \cdot (T^2 - \bar{T}^2). \end{aligned} \quad (3.8)$$

Equation 3.8 is now rewritten as,

$$\hat{f} = f^* + \Psi, \quad (3.9)$$

where Ψ is the sum of all terms that accompany the corrected frequency f^* in the

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Table 3.6: Adjusted R-Squared $R_{adj,k}$ and independent variable for each statistical model according to the number of predictors k .

Mode	Number of predictors k				$R_{adj,max}^2 (k_{max})$
	1	2	3	4	
1	0.627 T	0.761 $(\Delta T/\Delta t)_9$	0.777 $1/a_{RMS}$	0.785 T^2	0.839 (52)
2	0.616 $T_{R_{1000}}$	0.651 T	0.782 T^2	0.791 $1/a_{RMS}$	0.841 (45)
3	0.157 —	0.222 —	0.268 —	0.325 —	0.440 (35)
4	0.850 T	0.921 $(\Delta T/\Delta t)_{19}$	0.942 $T_{R_{1000}}$	0.955 T^2	0.967 (36)
5	0.521 —	0.537 —	0.553 —	0.565 —	0.621 (34)
6	0.874 T	0.929 $(\Delta T/\Delta t)_{18}$	0.943 $T_{R_{1000}}$	0.951 T^2	0.964 (56)
7	0.826 T	0.905 $T_{R_{1000}}$	0.925 $(\Delta T/\Delta t)_{11}$	0.943 T^2	0.953 (38)
8	0.502 —	0.546 —	0.567 —	0.577 —	0.653 (41)
9	0.799 $T_{R_{1000}}$	0.886 T	0.907 T^2	0.918 $(\Delta T/\Delta t)_{10}$	0.929 (29)

Figure 3.19: Validation of the statistical model with unseen data. (○) SSI estimates, (×) estimates from the statistical model.

right hand side of Equation 3.8. Taking into account that, in this case, the independent variable is the natural frequency estimated by SSI ($f^{(SSI)}$), Equation 3.1 can be expressed

$$f^{(SSI)} = \hat{f} + \varepsilon, \quad (3.10)$$

where ε is the residual error. Replacing Equation 3.9 into Equation 3.10, it leads to

$$f^{(SSI)} - \Psi = f^* + \varepsilon = \hat{f}^*, \quad (3.11)$$

in which $\hat{f}^*$ is the corrected frequency estimate, i.e. the corrected frequency including the residual error, which is independent of the influence of the considered predictors. Figure 3.20a and Figure 3.20b show $f^{(SSI)}$, $\hat{f}^*$ and f^* for mode 4. It can be observed that the variability range decreases from 0.1422 to 0.0446 Hz (that is, a 70 % reduction has been achieved). Finally, Figures 3.20c and 3.20d show the distribution of the residual errors for the SSI frequency estimate $f^{(SSI)}$ and the corrected frequency estimate $\hat{f}^*$ as well as their fittings to a Normal Gaussian Distribution. It is worthy to remark that the residual error for the corrected estimate fits to a Normal Gaussian Distribution, indicating that dependencies with external agents have been successfully eliminated.

3.5 Vibration Serviceability Assessment

The data collected with the low-cost vibration monitoring system have been also used to assess vibration serviceability using a general standard for human-induced vibration [40, 182]. Current codes and design guidelines for footbridges consider single-frequency excitation and rely on peak-acceleration-based criteria. However, standards ISO 2631-1:1997 [96] and ISO 10137:2007 [94], suitable for general human exposure evaluation to whole-body vibrations, are used here to obtain several vibration predictors: RMS acceleration, MTVV, VDV, CDF of peak weighted vertical acceleration and crest factor which is obtained from the ratio between the peak weighted acceleration and the total RMS value. These predictors are interesting when long-term vibration assessment is carried out since they take into account the duration of the vibration exposure as well as the effect of combined multi-axis, multi-frequency vibration.

The Vibration Serviceability Analysis (VSA) presented here is for one-year monitoring. Mode 4 is the most persistent throughout the one year of monitoring because it is the most excited and the most representative of the structure response (see Table 3.5). Because of this, the VSA presented in this section is only carried out for the accelerometer corresponding to the point of maximum displacement of this mode. This point is at one lateral side of the deck at 77.25 m from abutment.

3. Tracking modal parameters of a lightweight structure

(a) Complete view.

(b) Zoom view of 20 days.

(c) Residue before correcting.

(d) Residue after correcting.

Figure 3.20: Removing the external factors for mode 4. (x) SSI estimates $f^{(SSI)}$, (x) corrected frequency estimates $\hat{f}^*$, and (--) corrected frequency f^* .

Tests of 2-min duration have been analysed, which is the duration time estimated for a pedestrian crossing the structure, assuming speeds between 2 and 3 km/h (exposure time). The following vibration descriptors are computed using the W_k weighting acceleration a_{wk} : total RMS of each 2-min test, MTVV is calculated from the 1s running RMS acceleration, VDV and crest factor. These predictors are adopted for current standards [94, 96] as vibration predictor for general VSA.

Mean values may not be representative when long-time monitoring is considered. Figure 3.21 shows these predictor values over the year in an hourly distribution circular charts in which the mean value per hour of several predictors is represented. The latter plots provide a quick idea of how the structure is behaving during the day. It should be noted that for processing these plots, time histories with a very low signal-to-noise ratio have been removed from the VSA. Although the average value during the day of the crest factor is around 6 (see Figure 3.21d), this value is often exceeded. This fact means that a normal evaluation method based on total RMS values is not enough. MTVV and VDV are more appropriate predictors in this case (see Figures 3.21b and 3.21c).

CDF curves provide a really valuable information for long-term VSA. Figure 3.22 shows the mean CDF curve obtained for weighted vertical acceleration. This curve represents the percentage of time that an acceleration level is not exceeded (cumulative distribution). It can be observed a spreading area that illustrates the difference between extreme CDF curves. It is worth mentioning that although the average CDF curve may seem appropriate, there are many tests in which very high amplitudes are achieved. The experience of the authors in the field of the footbridges dynamics allows them to provide that the comfort values registered in the standards are not conservative and, for the comfort level obtained, the structure might experience excessive vibrations for pedestrians.

3.6 Conclusions

A low-cost vibration monitoring system based on MEMS accelerometers has been successfully installed on a singular stress-ribbon footbridge. It has been demonstrated that these low-cost sensors, carefully conditioned, can be a competitive alternative to the traditional ones. Thus, using this system, this chapter has focused mainly on the time evolution of the modal parameters and their correlation against environmental/operation factors. A simple and efficient way of removing environmental effects has been carried out based on dynamic MLR. A correction of these modal parameters has been proposed in such a way that the corrected frequencies might be used as a potential damage index in a SHM system. It should be taken into account that the presented methodology for the tracking of the modal parameters and removing the influence of external factors can be applied to other bridges following the same steps but considering that predictors may be different.

3. Tracking modal parameters of a lightweight structure

Figure 3.21: Mean value per hour of several predictors.

For this particular stress-ribbon structure, it has been demonstrated that it is highly-sensitive to temperature variation (frequency changes of more than 20%). Although only one ambient temperature sensor was installed, an interesting statistical model for the frequency estimates has been derived. This model makes use not only of the temperature, but also the gradient of temperature and the running averaged temperature, explaining more than 95% of the frequency variation over time of two of the vibration modes. A procedure to obtain the corrected frequency (removing the influence of external factors) has been proposed and the residual error in this case has been shown to be fitted to a narrow Normal Gaussian Distribution. This feature makes modal estimates more appropriate to be used as damage detectors. Furthermore, the uncertainties due to the estimation process (using different SSI techniques and different data blocks) have been studied. The corrected frequency has a closer range of variability than the original one. However, the variability for the corrected frequency has a wider range than the errors quantified from the estimation process.

Figure 3.22: Mean curve of likelihood of non-exceeded for weighted vertical acceleration. The shaded area indicates the range between the minimum and maximum curves for one-year monitoring.

A VSA of an in-service footbridge for vertical direction has been presented, employing a group of vibration predictors, such as RMS acceleration, MTVV, or crest factor, over one-year monitoring. Therefore, a comfort level for the structure could be properly established. The different predictors for general VSA are computed.

4

Study of semi-active implementable strategies

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4.1 Introduction

The in-service steel-plated stress-ribbon PGB Footbridge presented in the previous chapter was analysed and the influence of the environmental factors on modal parameter estimation was therefore extensively studied. It was found that this structure has the most perceptible vibration mode at approximately 1.8 Hz (within the normal range of walking). Through a peered analysis of a full-year monitoring, it was obtained that this mode changes up to 20 % during the year with both seasonal and daily trends and that these changes were mainly explained by temperature variation. Furthermore, the structure has other vibration modes

Figure 4.1: Distribution density. The line shows the normal distribution with the same mean and standard deviation as the original distribution density. Extracted from Figures 3.17 and 3.18.

with natural frequencies smaller than 4 Hz that might also be excited by human-induced vibrations. Thus, this chapter studies the performance of the two STMD strategies presented in Chapter 2, called STMD 1 and STMD 2, as compared to a classical TMD for this structure. The chapter continues with a description of two different models of the structure considering: firstly, only the aforementioned mode (SDOF model) and secondly, all the modes with natural frequencies smaller than 4 Hz (MDOF model), over time. The two semi-active control laws are considered to be implementable in practice. The application of an active control has also been simulated, using an AMD based on the use of a commercial electrodynamics shaker commanded by velocity feedback control law. To obtain the structure response, a novel load excitation based on a modulated chirp signal is proposed to evaluate the functioning of the different control strategies. Finally, the results obtained are discussed at the end of the chapter.

4.2 Structure models

The fourth mode of Table 3.5 was shown to be the most critical one for the VSA. Figure 4.1 shows the distribution of the natural frequency and damping ratio estimates of mode 4 for the full year. As it can be observed, the damping estimates fit to a normal distribution whereas the frequency estimates do not fit to it. Besides, the average value and the most repeated one are clearly different for frequency estimates.

Two structure models are considered to study the performance of the control strategies. Firstly, a SDOF model corresponding to mode 4 is considered (Table 3.5). It is assumed that the control device is placed at the point of maximum displacement of mode 4 (at one lateral side of the deck at 77.25 m from abutment).

Table 4.1: Summary of modal participation factors, α_i , for each vibration mode at the maximum amplitude node of mode 4.

Mode	$\bar{f}$ (Hz)	α (kg ⁻¹)
1	1.0482	$3.26 \cdot 10^{-6}$
2	1.4145	$1.30 \cdot 10^{-5}$
3	1.5440	$1.13 \cdot 10^{-5}$
4	1.7937	$2.00 \cdot 10^{-5}$
5	1.8594	$1.32 \cdot 10^{-5}$
6	2.3117	$1.19 \cdot 10^{-5}$
7	3.3821	$5.11 \cdot 10^{-6}$
8	3.5512	$2.10 \cdot 10^{-5}$
9	3.9610	$2.51 \cdot 10^{-6}$

Secondly, the effect of other modes at that point is included in such a way that a MDOF system is considered. The frequency and damping for each vibration mode will change according to estimates over the year. Table 4.1 shows the modal participation factors, α_i , at such location which will be considered fixed values and have been obtained from an updated FEM [42]. Thus, the TF between the structure acceleration and the applied force in the Laplace domain is as follows:

$$G(s) = \sum_{i=1}^N \frac{\alpha_i s^2}{s^2 + 2\zeta_i \omega_i s + \omega_i^2}, \quad (4.1)$$

in which $s = j\omega$, $\omega = 2\pi f$ being the circular frequency (rad/s) and f the natural frequency (Hz), α_i modal participation factor (taken from Table 4.1), ω_i is the circular frequency and ζ_i is the damping ratio. The values of these varying modal parameters are those estimated experimentally for the full-year (in the footbridge and represented by their mean value in Table 3.5). N is the number of vibration modes taken into account in the model ($N = 1$ and $N = 9$ for the SDOF and MDOF models, respectively).

Figure 4.2a shows the 13817 models considered for the case of SDOF system. Figure 4.2b shows the 1721 models considered for the case of MDOF system. This is the number of estimations for which the ninth first modes are estimated simultaneously.

4.3 Loading cases

The force used to evaluate the vibration control strategies has been a chirp waveform whose frequency increases at a linear range with time, as follows

$$F(t) = F_0(f) \cdot \sin(\psi(t)), \quad (4.2)$$

(a) SDOF.

(b) MDOF.

Figure 4.2: Magnitude of Transfer Functions of the models considered over the year.

in which $\psi(t) = 2\pi f(t)t$ is the angle of the chirp signal, t being the time, which is defined depending of the instantaneous frequency:

$$\begin{aligned} f(t) &= f_i + kt, \\ k &= \frac{f_f - f_i}{T_f}, \end{aligned} \tag{4.3}$$

where k is the rate of frequency change, f_f is the final frequency, f_i is the starting frequency and T_f is the time it takes to sweep from f_i to f_f . The frequency range has been chosen in order to excite the nine vibration modes below 4 Hz. The final time, T_f , has been 500 s and 1000 s, for the SDOF and MDOF system, respectively. These values were chosen to ensure that sweeping at each frequency is sufficiently slow so that the results do not depend on the final time.

The amplitude of the force has been frequency weighted according to dynamic load factors for walking induced vertical forces [30, 192]. The following definition for $F_0(f)$ has been taken here,

$$F_0(f) = \begin{cases} 17.09 f, & f < 1.2 \text{ Hz} \\ 316.25 f - 358.98, & 1.2 \text{ Hz} \leq f < 2.4 \text{ Hz} \\ -2.56 f - 51.28, & 2.4 \text{ Hz} \leq f < 20 \text{ Hz} \\ 0, & f \geq 20 \text{ Hz} \end{cases} . \tag{4.4}$$

Figure 4.3 shows $F_0(f)$ for the considered frequency range. Note that the amplitude is representative of the load imparted by a pedestrian walking at different frequencies. Besides, this input force can be easily generated by a shaker to assess control devices designed for cancelling human-induced vibration.

4.4 Vibration control strategies

4.4.1 Passive and semi-active control

The TMD mass is fixed as a fraction of the modal mass (which is the inverse of the modal participation factor α_i , Equation 4.1) of the targeted vibration mode; the stiffness of the springs is selected to obtain the optimum TMD frequency, and the viscous dampers ensure the operation of the TMD in a range of frequencies around the tuning frequency. The primary system is the structure modelled as a SDOF system, which is composed of a mass m_s , a spring of constant k_s and a viscous damper of constant c_s . These physical parameters are equivalent to the modal parameters of Equation 4.1 with $N = 1$: $\alpha_s = 1/m_s$, $k_s = \omega_s^2/\alpha_s$ and $c_s = (2\zeta_s\omega_s)/\alpha_s$.

Figure 4.3: Weighted force amplitude, $F_0(f)$.

The TMD has been designed using the [Asami and Nishihara \[12\]](#) formulas (see [Equation 2.11](#) and [Equation 2.13](#) from [Chapter 2](#)) and only mode 4 is considered for TMD design ($i = 1$ in [Equation 4.1](#)) with a mass ratio of 1 % ($m_r = 500$ kg). The TMD might detune when variations into the modal parameters of the structure take place, as already mentioned in [page 19](#), and this is the cases studied in this chapter. For the nominal tests, the tuning process is trivial, both for the SDOF and MDOF cases. Nevertheless, the tuning process for the full-year tests of SDOF and MDOF cases is different. The most-repeated modal properties of 4th vibration mode, frequency and damping ratio, are 1.82 Hz and 0.37 %, respectively (see [Figure 4.1](#)). These values have been adopted to tune the passive control device for the full-year tests in order to be optimally tuned as long as possible throughout the full year of estimates. The passive solution is chosen in this study as a benchmark damper ([Figure 1.12b](#)), as it is also done, for instance, in [\[196\]](#). Meanwhile, the laws of semi-active control, STMD 1 (see [Figure 2.31a](#) and [Equation 2.17](#)) and STMD 2 (see [Figure 2.31b](#) and Equations from (2.16) to (2.24)), explained in [Chapter 2](#) are used for the upgrading of the passive TMD ([Figure 1.12b](#)).

4.4.2 Active control

In addition, of the passive and semi-active control strategies, an active control system is also studied in order to have a numerical reference of its operation. The inertial mass actuator, APS Dynamic Model 400, presented in [Chapter 2](#) (see [Figure 2.16](#)), is used here. This proof-mass actuator generates inertial forces in the structure on which it is placed without the need for a fixed reference. The

TF between the inertial force applied to the structure $F_A(t)$ and the input voltage $V(t)$ can be closely described as a linear second-order system as follows [160]:

$$G_A(s) = \frac{K_A s^2}{s^2 + 2\zeta_A \omega_A s + \omega_A^2}, \quad (4.5)$$

in which K_A is the actuator gain (N/V), $\omega_A = 2\pi \cdot 0.9$ rad/s is the natural frequency associated with the suspended moving mass and $\zeta_A = 0.5$ is the damping ratio (including electrical and mechanical effects). The natural frequency ω_A must be sufficiently below the first natural frequency of the structure ω_s , in such a way that the phase distortion introduced by the proof-mass actuator does not affect significantly the efficacy of the AVC system at the structure frequency. Typically, it is recommended that ω_A less than half of ω_s [69]. The actuator has a moving mass which is of $m_A = 30.4$ kg and its maximum force in state steady conditions is of 400 N. Note that the resonance of the shaker can be adapted to the structure necessities through the use of a local (or inner) feedback control as it was shown in [70].

The feedback control scheme considered is shown in Figure 4.4. In this case, velocity feedback control has been adopted together with a saturation non-linearity included to avoid actuator overloading [71]. That is, the initial control voltage is clipped up to ± 2 V. The compensator is a lossy integrator and a control gain as follows

$$C(s) = K_c \left(\frac{\omega_c}{s + \omega_c} \right), \quad (4.6)$$

with $\omega_c = 2\pi \cdot 0.4$ rad/s and $K_c = 1174$ which is the control gain obtained from the root locus method (see Figure 4.5) for the nominal structure model in such a way that the damping of the closed-loop poles corresponding to the actuator are approximately $\zeta_A = 0.2$, which avoid stroke saturation due to low frequency noise. The root locus represents the poles of the total closed-loop TF once the control is applied. Actually, these poles are the vibration modes of the the closed-loop system. The structure increases its damping but the actuator reduces it. Note that if the value of K_c is increased, the imparted damping to the structure is higher, however, the control loop system is more sensitive to stroke saturation for low frequency noise (see the root locus of Figure 4.5).

4. Study of semi-active implementable strategies

Figure 4.4: Active control scheme.

Figure 4.5: Root locus. × open-loop poles, ■ closed-loop poles.

4.5 Results with TMD, STMD 1 and STMD 2

Four cases are analyzed, SDOF and MDOF systems for nominal and full year analysis:

- SDOF system

- Nominal case

A SDOF model of the structure with mean frequency and mean damping (see [Figure 4.1](#)) of the full year is adopted.

- Full year case

13 817 SDOF models estimated for the full year are simulated. [Figure 4.2a](#) illustrates the models. Control strategy degradation from nominal case is studied in this case.

- MDOF system

- Nominal case

A MDOF model with the ninth first vibration modes is considered. Mean values for frequency and damping (see [Figure 3.14](#)) are used for the nominal model. The influence of additional modes on the control performance is studied against SDOF nominal case.

- Full year case

A value of 1 712 MDOF models are obtained making the intersection between the modal estimates of the 9 tracked vibration modes over the full year monitoring. The effects of additional time-varying modes is studied. [Figure 4.2b](#) illustrates the model variations over time.

The loading force described in [Section 4.3](#) is applied to the structural model for the uncontrolled case and for the three control strategies described previously. The control performance of these three strategies are compared in time and frequency domain:

- In time domain, the MTVV calculated from the 1s running RMS acceleration is considered [\[95\]](#).
- In frequency domain, the H_∞ and H_2 values are used to evaluate the control performance [\[200\]](#).

The vibration reduction for each case is computed as follows. For nominal cases, the vibration reduction is directly obtained as (uncontrolled value – controlled value)/uncontrolled value. For the full year cases, the CDF curves, explained in [Chapter 2](#), are computed for the MTVV, H_∞ and H_2 . Thus, the vibration

reduction is obtained from the total area between the CDF curve and y -axis, denoted by subscript “area”. This is because the smaller this area is, lower is also the level of vibration experienced. That is, this solution is computed as follows:

$$\text{Vibration Reduction} = \frac{\text{Uncontrolled CDF}_{\text{area}} - \text{Controlled CDF}_{\text{area}}}{\text{Uncontrolled CDF}_{\text{area}}}, \quad (4.7)$$

Through this procedure, a single value can be derived to quantify the reduction performance for full year analysis.

4.5.1 Single degree of freedom system

4.5.1.1 Nominal case

Figure 4.6 shows the acceleration responses and their corresponding frequency spectra. It is observed that both STMDs, 1 and 2, improve marginally the results as compared to the TMD. The vibration reductions for each case are 79.04%, 84.08% and 83.78% (MTVV), 90.23%, 92.69% and 92.69% (H_∞) and 34.57%, 42.28% and 42.75% (H_2) for TMD, STMD 1 and STMD 2, respectively.

4.5.1.2 Full year tests

The TMD might detune when variations into the modal parameters of the structure take place, as it is the case studied here. As described in Subsection 4.4.1, these values have been adopted to tune the control device. Figure 4.11 shows the MTVV obtained for the uncontrolled case and the three control strategies (TMD, STMD 1 and STMD 2).

Figure 4.8 shows CDF curves for MTVV, H_∞ and H_2 obtained for the full year distribution response. It is worth mentioning that, in order to observe the difference between the semi-active laws, it is necessary to zoom in. The difference in H_2 is negligible, however for MTVV and H_∞ there are differences for higher amplitudes. As it can be obtained, the results of STMDs are low-dependent on structure model

The reduction for each case is 70.57%, 77.22% and 78.74% (MTVV), 88.18%, 91.20% and 91.71% (H_∞) and 31.04%, 38.90% and 38.96% (H_2) for passive TMD, STMD 1 and STMD 2, respectively.

(a) Time domain, time histories.

(b) Frequency domain, frequency spectrum.

Figure 4.6: Response of SDOF system.

4.5.2 Multi-degree of freedom system

4.5.2.1 Nominal case

Figure 4.9 shows the accelerations responses and their corresponding frequency spectra. The reduction for each case is 18.64%, 61.19% and 64.03% (MTVV), 28.28%, 69.11% and 71.01% (H_∞) and 29.73%, 51.59% and 52.81% (H_2) for passive TMD, STMD 1 and STMD 2, respectively. Both semi-active devices have a very similar performance and much better than the passive one.

(a) Complete view.

(b) Zoom view.

Figure 4.7: MTVV vs Frequency.

4.5.2.2 Full year tests

As for the SDOF system for full year tests, the TMD properties are tuned to the most repeated structural properties. Figure 4.10 shows CDF curves obtained for this case. The reduction for each case is 10.82%, 42.85% and 44.20% (MTVV), 33.85%, 69.83% and 70.92% (H_∞) and 19.71%, 38.04% and 37.56% (H_2) for passive TMD, STMD 1 and STMD 2, respectively. Again, both STMDs have similar performance and much better than the passive one. The inclusion of multiple modes degrades drastically the performance of control strategies. This effect is much more apparent for the TMD.

Figure 4.8: CDF curves for SDOF case.

4.6 Results with TMD, STMD 1 and AMD

Figure 4.11 and Figure 4.12 show the MTVV obtained for the uncontrolled case and the three control strategies using the 13817 SDOF models for both loading cases, chirp of 400 N and 1 500 N amplitudes. As it is observed, the STMD 1 always improves results as compared to the TMD even for the nominal model for which the TMD was designed, being the results very similar in that case. It is also observed that the AMD considered (moving mass of 30.5 kg, the TMD and STMD mass was of 500 kg) can cope with low-amplitude vibration (chirp of 400 N) very efficiently; however, it is not able to cope with vibrations coming from a stronger force (chirp of 1 500 N). The amplitude level of MTVV obtained is directly proportional to the damping of the structure for each case (see Figures 4.11b and 4.12b).

4. Study of semi-active implementable strategies

(a) Time domain, 2D view.

(b) Frequency domain, frequency spectrum.

Figure 4.9: Response of MDOF system.

Finally, [Figure 4.13](#) shows the percentage of reduction distribution for the whole year and for both loading cases. Again, the STMD performances better than the TMD: the reduction is higher and the distribution band is narrower so that its behavior is low-dependent on the structure model. The AMD for the 400-N force is totally independent of the structure model and is able to reduce up to 98% of the uncontrolled case. However, for the case of 1 500 N force, the AMD is not able to cancel them due to its limitation in the maximum force.

Figure 4.10: CDF curves for MDOF case.

4.7 Discussion

A complete study has been carried out in order to explore different practical semi-active control strategies to reduce vibrations in a frequency-varying real structure over a full year monitoring taking into account only one mode or several closed-frequency modes. This study helps in making a decision about the more convenient semi-active control technique to be considered for implementation as well as to quantify their improvements with respect to the passive system.

Table 4.2 summarizes the results described in Section 4.5. Thus, the following conclusions can be drawn:

4. Study of semi-active implementable strategies

Figure 4.11: Results for 400-N chirp amplitude.

- The structure analysed shows plenty of vibration modes with closed-space frequencies in a critical short frequency range (below 4 Hz). The modal properties of these modes change over the year mainly due to temperature variations. This fact together with the human-induced vibrations that overcome recommended serviceability limits, makes this structure an ideal candidate to implement a semi-active control strategy.
- A comprehensive study using SDOF and MDOF systems and accounting for full year tests has been carried out. A chirp function amplitude-weighted has been employed as excitation. The amplitude weighting accounts for the different loading factor depending on the excitation frequency. Several vibration reduction indexes, in frequency and time domain, have been computed in order to assess the performance of the control strategies. Interesting indexes based on MTVV, H_∞ and H_2 values obtained from CDFs over the year have been used.

Figure 4.12: Results for 1500-N chirp amplitude.

- The TMD, even though it behaves quite well, degrades its performance significantly when model frequencies of the targeted vibration mode (SDOF) move away from the nominal model. Besides, it worsens abruptly when different modes of vibration are excited (MDOF).
- Both STMDs, 1 and 2, have been shown to be quite insensitive to its initial tuning and able to cancel effectively the vibration independently of the modal properties of the tuned vibration mode (SDOF) and, also other vibration modes (MDOF) over the full year.
- The study demonstrates that two control algorithms coming from different conceptual approaches have similar performances in terms of control operations and response reduction. There is a slightly better performance with the STMD 2 but it turns out to be insignificant for the study carried out

4. Study of semi-active implementable strategies

(a) 400-N chirp amplitude.

(b) 1500-N chirp amplitude.

Figure 4.13: Distribution of % of vibration reduction for the three strategies.

through this chapter as compared to the simplicity that STMD 1 has for practical implementation.

- The commercial shaker considered as AMD (30 kg moving mass) is extremely effective for low amplitude vibration but it is not able to cope with large amplitude due to extreme loading, since maximum control force delivered by it is not able to cope with large amplitudes due to extreme loading as it is shown in [Section 4.6](#).

Table 4.2: Summary of vibration reduction (%) for each case.

Magnitude	Control	SDOF	MDOF	Case
MTVV	TMD	79.04	18.64	Nominal
	STMD 1	84.08	61.19	
	STMD 2	83.78	64.03	
H_∞	TMD	90.23	28.28	
	STMD 1	92.69	69.11	
	STMD 2	92.69	71.01	
H_2	TMD	34.57	29.73	
	STMD 1	42.28	51.59	
	STMD 2	42.75	52.81	
MTVV CDF area	TMD	70.57	10.82	Full year
	STMD 1	77.22	42.85	
	STMD 2	78.74	44.20	
H_∞ CDF area	TMD	88.18	33.85	
	STMD 1	91.20	69.83	
	STMD 2	91.71	70.92	
H_2 CDF area	TMD	31.04	19.71	
	STMD 1	38.90	38.04	
	STMD 2	38.96	37.56	

5

Optimal tuning of semi-active TMD

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5.1 Introduction

In the previous chapter, it has been studied the performance of the two STMD strategies presented in [Chapter 2](#). But several issues have to be tackled to achieve

a successful implementation for broad frequency-band vibration: i) the design of the control law to implement, ii) the frequency tuning in passive mode, iii) the damping in passive mode, iv) the definition of a PI able to represent the performance in a broad frequency band and v) practical issues such as filtering and switching on-off function. Some of these issues have been partially addressed previously in other research works [79, 113, 199].

An MR damper can be used within control systems. However, MR dampers have a strongly non-linear nature that has to be modelled and included in control design. Otherwise, the simulated performance may differ considerably from the experimental performance.

Although sensitivity studies of semi-active control law parameters have been accomplished, as well as optimal tuning proposals [79, 140], the main purpose of the semi-active device has not considered for its tuning: be effective in a broad frequency-band spectrum. Cases such as modal properties varying with time due to external factors (temperature, level of occupancy or ageing) and structures with several vibration modes with closely-spaced frequencies prone to vibrate are common in practice (see Chapter 3). Under these circumstances, much greater robustness than that provided by the TMD is required and a design approach that considers both uncertainties in structure modal parameters and an MR damper is desirable.

Thus, the tuning of the semi-active system should take into account its broad spectrum of operation and the significant non-linearities, such as the control law itself, as well as the operation of the MR damper. Then, a new approach to tuning STMDs is carried out in this article and compared to the passive performance. Necessary aspects to implement the control law are also described and taken into account within the optimization problem. Although the problem is stated in a general way, this is clearly geared to highly resonant structures (for instance, lightweight footbridges) and the PI to be minimised is focused on user comfort. The following issues are considered: i) the velocity of the inertial mass should be estimated from its acceleration, ii) a continuous trigger function should be implemented in order to avoid unnecessary damper actions when structure acceleration is of low amplitude, and iii) acceleration of the structure should be carefully filtered to avoid spillover effects and unnecessary damper action when this signal is crossing over zero.

The chapter continues with a description of the semi-active inertial controller and the vibration control strategy adopted in Section 5.2. A sensitivity analysis of the tuning parameters of the control device is carried out in Section 5.3. The effect of including damper force saturation and an MR damper model instead of a linear-viscous damper are also analysed. Lastly, an optimization procedure to tune the semi-active device is proposed. Section 5.4 shows the experimental results conducted on a steel beam and carefully explains the implementation of the semi-active control and the practical issues that have been considered. Finally, some conclusions are given in Section 5.5.

5.2 Semi-active Tuned Mass Damper

5.2.1 Tuned Mass Damper

The TMD has been designed using the approximate solution provided by [Asami and Nishihara \[12\]](#), based on H_∞ optimization for primary systems with vanishing damping. The expressions by [Asami and Nishihara](#) minimise the acceleration under harmonic excitation which is related with potential applications in footbridges and also to assess the pedestrian comfort. This solution is the initial solution for the optimization process presented in [Subsection 5.2.3](#).

5.2.2 Semi-active control strategy

The phase control strategy for the TMD damping presented in [Chapter 2](#) and studied in [Chapter 4](#), called STMD 1, is used here. Thus, the adopted control law when ideal viscous damping is assumed is defined in [Equation 2.15](#). When an MR damper is assumed (see [Figure 2.39b](#)), the law of [Equation 2.15](#) is rewritten as follows:

$$\begin{cases} \ddot{x}_S \cdot \dot{x}_T \leq 0 & \Rightarrow & I = I_{\min} & \text{(normal functioning)} \\ \ddot{x}_S \cdot \dot{x}_T > 0 & \Rightarrow & I = I_{\max} & \text{(blocking action),} \end{cases} \quad (5.1)$$

in which I is the current applied to the MR damper. The maximum and minimum value, $I_{\max}$ and $I_{\min}$, depend on the MR damper employed. This strategy is fairly simple and implementable as it uses only two magnitudes easily measured. However, a few aspects must be taken into account before the practical implementation. These are described in [Subsection 5.4.3](#).

5.2.3 Optimal control design

The purpose of this section is to provide a procedure to find an optimal tuning for STMD control devices in structures with uncertainty in their modal parameters and/or with several critical vibration modes. The term closed-loop control implies the use of a feedback controller to bring the output of the system (movement of structure) to a desired value. The controller changes the dynamics of the system. The general feedback control scheme is shown in [Figure 5.1](#), in which the inertial controller changes in real time following the semi-active control law.

The optimization will minimise a PI over a broad frequency band which is geared to structures that show resonant behaviour. The PI is defined according to the

Figure 5.1: Simplified control scheme with an inertial controller.

following expression:

$$\text{PI} = \max \left(\sum_{i=1}^N \hat{S}_i(\omega) \right) \quad \text{with} \quad \hat{S}_i = \frac{1}{F_0} S_i(\omega), \quad (5.2)$$

in which F_0 is the force amplitude of the excitation force $F(t)$, $S_i(\omega)$ is the spectrum of the acceleration response of the structure ($\ddot{x}_s(t)$), $\hat{S}_i(\omega)$ is denoted as the normalized spectrum and N may be the number of vibration modes to be cancelled or a set of models representative of a particular uncertain vibration mode. This set may be derived considering a normal distribution of the modal parameters or may come from experimental measures as in [Chapter 4](#). Note that the $\hat{S}(\omega)$ is used instead of the frequency response function due to control system non linearities. The optimization problem consists of minimizing the PI defined in [Equation 5.2](#) subjected to the following constraints:

$$\min_{f_T, \zeta_T} (\text{PI}) \quad \text{subject to} \quad \begin{cases} f_T > 0 \\ \zeta_T > 0 \\ |x_T| \leq \text{maximum stroke} \end{cases}, \quad (5.3)$$

in which f_T and ζ_T are the frequency and damping ratio of the controller device. The variable $|x_T|$ is the displacement of the inertial mass that must be limited by the maximum stroke and mass ratio μ is considered fixed. Note that in [Equation 5.3](#), the damping ratio ζ_T , equivalent to $c_{\min}$ in [Equation 2.15](#), is substituted by $I_{\min}$ (see [Equation 5.1](#)) when the MR damper model is included in the design.

The perturbation force $F(t)$ used to evaluate the structural response is a chirp waveform whose frequency increases at a linear range with time as explained in [Section 4.3](#) and Equations from (4.2) from (4.4). The frequency range is chosen to excite a broad frequency band and the duration is chosen to ensure that sweeping at each frequency is sufficiently slow.

5.3 Simulation results

Two kinds of analysis, described in [Subsection 5.3.1](#) and [Subsection 5.3.2](#), are carried out for the four configurations model shown in [Figure 5.2](#) jointly. The first analysis is a sensitivity analysis of the tuning parameters of the control device. The second one consists of running the optimization procedure defined in [Equation 5.3](#).

Figure 5.2: Model of the 2-DOF systems studied. Red symbol ($\nearrow$) means changing over time.

5.3.1 Sensitivity analysis

The following parameters have been adopted to undergo the sensitivity analysis:

- The excitation has been applied using a chirp signal (see [Equation 4.2](#)) with a constant amplitude of 50 N, linear-varying frequency from 0.1 and 6 Hz and a sampling frequency of 500 Hz for a record of 500 s.
- The structure parameters are: $m_s = 500$ kg, $f_s = 2$ Hz and $\zeta_s = 0.05$.
- The inertial controller parameters are variables: $f_T \in (0.1 \cdot f_s, 2 \cdot f_s)$ Hz and $\zeta_T \in (0.01, 0.20)$ for TMD and STMD with viscous damping ([Figure 5.2a](#) and [Figure 5.2b](#), respectively), and $I \in (0, 0.5)$ A for MR-TMD and MR-STMD ([Figure 5.2c](#) and [Figure 5.2d](#), respectively).
- The mass ratio is constant, $\mu = 0.02$.

[Table 5.1](#) shows the parameters used in the sensitivity analysis. The PI is studied for the tuning frequency ratio η and the damping ratio ζ_T . A total of 2601 simulations have been carried out. The inclusion of the MR damper model aims to study the effect of degradation in the operation of the control device, passive and semi-active.

Table 5.1: Parameters used for the sensitivity analysis.

Parameter	Value
Mass, m_s (kg)	500
Damping ratio, ζ_s (%)	0.5
Frequency, f_s (Hz)	2.0
Mass ratio, μ	0.02
Inertial mass, m_T (kg)	10
Maximum damper force saturation, $F_{D_{\max}}$ (N)	75
Chirp amplitude, F_0 (N)	50
Chirp initial frequency, f_i (Hz)	0.1
Chirp final frequency, f_f (Hz)	6.0

5.3.2 Control design parameters

Having completed the sensitivity study, the optimization problem is run. The stroke of the device has been limited to 0.3 m and the value of the mass ratio has been set again at $\mu = 2\%$. Note that both variables are related, the higher the mass ratio is, the smaller the stroke achieved.

The PI minimization has been computed using the `patternsearch` function from MATLAB software [129]. The pattern-search method seeks a minimum based on an adaptive mesh that, in the absence of linear constraints, is aligned with the coordinate directions [4]. Several optimization algorithms have been compared and pattern-search has always been the most efficient in achieving the best solution.

The optimization is carried out assuming that the parameters involved are variables following a normal distribution randomly generated. Each varying parameter has the following characteristics: $f_s \sim \mathcal{N}(2, 0.2)$ Hz, $\zeta_s \sim \mathcal{N}(1, 0.5)\%$, $m_s \sim \mathcal{N}(500, 100)$ kg and $F_0 \sim \mathcal{N}(50, 25)$ N, in which $\mathcal{N}(\nu_i, \sigma_i^2)$ indicates the normal or Gaussian distribution with ν_i and σ_i^2 being the mean and the standard deviation, respectively. Each randomly generated value of each parameter is grouped with those that occupy the same position in the generation sequence, obtaining a set of randomly generated models for the same structure. The PI has been computed from 100 models, that is, $N = 100$ in Equation 5.2.

5.3.3 Ideal viscous damper

The sensitivity analysis defined in Subsection 5.3.1 is carried out for TMD (Figure 5.2a) and STMD (Figure 5.2b). For the latter, only the $c_{\min}$ is tuned since $c_{\max}$ is assumed to be enough for blocking (see Equation 2.15). The PI of Equation 5.2 is computed for each case. Figure 5.3 shows the contour plots including isolines of same PI for ζ_T and η . It is clear that the STMD is more robust than the TMD. That is, taking a value of ζ_T , the frequency band for the same PI is much broader for the STMD.

How the PI is obtained is explained here. Firstly, the set of randomly-generated models is obtained (Figure 5.4a). Then, Figure 5.4b shows the result of Equation 5.3, in which the maximum value of the sum of the spectra is the PI.

Figure 5.3: Contour plot of the PI for TMD and STMD with linear-viscous damper.

Figure 5.5 shows the envelope curve of the PI with the initial solution together with the optimized one obtained for passive (5.5a) and semi-active (5.5b). Although the maximum value of the curve (PI) is the value to be optimized, it must be appreciated that all the points of the curve in the semi-active case are below that of the passive case. Table 5.2 shows a summary of the results associated to Figure 5.5a and Figure 5.5b. On the one hand, the optimum design of the TMD improves the PI obtained from the classical approach (see Equation 2.11). To achieve this, the TMD damping should be much greater in order to deal with the system uncertainty. On the other hand, for the considered control law, the optimal design parameters achieved significantly improves the performance reduction, and interestingly, the unblocking damping of the STMD should be much slower than the one obtained from the classical approach.

Figure 5.4: Uncontrolled case. Obtaining the PI.

Figure 5.5: Envelope curve and PI. Optimization for the case of viscous damping. The legend shows the PI values for these cases: (---) uncontrolled, (- - - -) initial solution and (—) optimized solution.

5.3.4 Effect of damper force saturation

Now, a maximum damper force is added to the model: the force saturation. All the cases shown in Figure 5.3 are repeated with different force saturation values in order to evaluate the performance degradation of the device since its capacity to block is limited, that is:

$$F_D = c_{\max} (\dot{x}_T - \dot{x}_S) \leq F_{D_{\max}}. \quad (5.4)$$

Table 5.2: Results from the optimization procedure for TMD and STMD. The improvement is computed from the initial solution of each device.

Control device	f_T (Hz)	ζ_T (%)	PI	Red. (%)	Impr. (%)
Uncontrolled structure	–	–	3.54	–	–
TMD (Asami)	1.98	8.65	1.52	57.06	–
TMD (Optimal)	2.02	13.25	1.39	60.73	↑ 3.67
STMD (Asami)	1.98	8.65	1.15	67.51	–
STMD (Optimal)	2.08	3.80	0.89	74.86	↑ 7.35

The MTVV is now used and CDF curves for the MTVV are plotted in Figure 5.6a for several values of $F_{D_{\max}}$ (without control and with the TMD are also included for comparison). For the case of the TMD, $F_{D_{\max}}$ has no effects since demanded forces are always lower than that value. For the STMD, increasing the values of $F_{D_{\max}}$ improves the performance, since the blocking capacity is enhanced. In order to have a single value to compare the results with different $F_{D_{\max}}$, the total area between the CDF curve and the y -axis is obtained (see Equation 4.7), resulting in Figure 5.6b. Thus, this figure shows the performance of each case referred to the case of 500 N (which is considered as perfect blocking). Initially, small $F_{D_{\max}}$ increments produce important performance improvements; however, from a certain $F_{D_{\max}}$ value, the increments do not result in significant performance improvements. With only $F_{D_{\max}} = 75$ N, 95% of the maximum performance is achieved.

5.3.5 Effect of considering an MR damper model

In order to consider the MR damper model in the controller design, a model identification of the MR damper used for the experimental test was carried out. The identification study conducted by [43] has been taken as reference research. Bingham and Bouc-Wen models were identified. The objective is to study how the TMD and STMD performances degrade when an MR damper model is considered (MR-TMD, Figure 5.2c and MR-STMD, Figure 5.2d) instead of an ideal viscous damper.

5.3.5.1 Experimental tests

Figure 5.7 shows the experimental setup together with the MR damper to be identified: sponge-type magneto-rheological damper from the Lord Corporation company. A steel frame combined with an electromagnetic actuator (APS Dynamics Model 400), an accelerometer, an LVDT and a load cell have been used in order to measure the damper acceleration, the displacement, and the force, respectively. Each test was done with a sinusoidal excitation with fixed frequency, fixed shaker gain (amplitude displacement is measured by an LVDT) and fixed damper current. This process has been repeated for several combinations. Therein, the

5. Optimal tuning of semi-active TMD

(a) CDF curves of the MTVV for several $F_{D_{\max}}$.

(b) Percentage of the maximum performance for several $F_{D_{\max}}$.

Figure 5.6: Performance in terms of the CDF of the MTVV for several damper force saturations $F_{D_{\max}}$.

damping ratio is changed by the input current of the MR damper. The selected range of frequencies, shaker gains and current supplies involved in the experimental procedure are specified in Table 5.3. A total of 252 tests were completed.

Table 5.3: Variation range of the input parameters for the MR damper identification.

Parameter	Range
Shaker gain (V)	0.05 to 0.8
Frequency (Hz)	1 to 7
Input current I (A)	0 to 0.5

Figure 5.7: Experimental setup for MR damper model identification.

Figure 5.8: Summary of the experimental results conducted for the identification of the MR damper model.

Figure 5.8a shows, in a single plot, all the experimental variables involved in the identification process. The amplitude of the force, displacement and acceleration in the steady-state response of each test are shown. Figure 5.8b shows that the dependency of the acceleration with the displacement and the frequency is much lower than that with the input current, in such a way that the Bingham model parameters are considered to be dependent only on the current I . Thus, the parameters of the model

$$F = F_c \operatorname{sgn}(\dot{x}) + c_0 \dot{x} + F_0,$$

presented in Subsection 2.6.3. They were calibrated using `lsqcurvefit` MATLAB routine (least-square method) minimizing the error between the force time history curves of the experimental force and the force predicted by the models. The relationship between the model parameters and the operating current are defined by the following polynomial functions:

$$F_c(I) = 216.08I^5 - 200.09I^4 + 4.7248I^3 - 20.422I^2 + 2.2397I + 1.8008 \quad [\text{N}]$$

$$c_0(I) = 17958.0I^5 - 24636.0I^4 + 6838.0I^3 - 2013.9I^2 + 58.367I + 117.55 \quad [\text{N}\cdot\text{s}/\text{mm}]$$

$$F_0(I) = 0 \quad [\text{N}].$$

The same process was followed by the Bouc-Wen model:

$$F = c_0 \dot{x} + k_0 (x - x_0) + \alpha z,$$

where the evolutionary variable z for the hysteresis behaviour is governed by

$$\dot{z} = -\gamma |\dot{x}| z |z|^{n-1} - \beta \dot{x} |z|^n + A \dot{x},$$

presented in Subsection 2.6.4. The results are:

$$c_0 = 2128.5I^5 - 2325.0I^4 + 557.44I^3 + 44.179I^2 - 2.7683I + 43.445 \quad [\text{N}\cdot\text{s}/\text{mm}]$$

$$k_0 = -486694.0I^5 + 401093.0I^4 - 38592.0I^3 - 9657.09I^2 + 2554.52I + 1655.06 \quad [\text{N}/\text{mm}]$$

$$A = 3516.9I^5 - 6681.6I^4 + 2539.6I^3 - 647.13I^2 - 96.528I + 87.002 \quad [-]$$

$$\alpha = -1140.9I^5 + 704.52I^4 - 562.46I^3 + 403.27I^2 + 88.389I + 101.28 \quad [\text{N}/\text{mm}]$$

$$\beta = -130567.0I^5 + 152277.0I^4 - 43708.7I^3 - 2606.44I^2 + 1600.62I + 478.221 \quad [\text{mm}]^{-2}$$

$$\gamma = 9905.0I^5 - 11295.0I^4 + 2532.8I^3 + 472.65I^2 - 71.096I + 149.55 \quad [\text{mm}]^{-2},$$

Figure 5.9 shows a comparison between the corresponding experimental data and the predicted responses using Bingham and Bouc-Wen models in an example with 0.1 A of damper current and 2 Hz of excitation sinusoidal force (with 0.18 V of shaker gain, around 7 mm of amplitude). The force-displacement behaviour of

the MR damper is well-predicted for both models. However, the force-velocity behaviour is closer in the case of Bouc-Wen model.

(a) Force versus Time.

(b) Force versus Displacement.

(c) Force versus Velocity.

Figure 5.9: Comparison between experimental response and Bingham and Bouc-Wen models predicted responses.

The initial condition for the optimization process will be assumed for $I = 0$ A. For the semi-active case, it is necessary to optimize the minimum damper current, because the current to block the inertial mass is the maximum value for the on-off control law. Figure 5.10 shows the degradation obtained when the Bingham MR damper model is used for the sensitivity analysis with the MR-TMD and the MR-STMD. The following aspects are observed from Figure 5.10 as compared to Figure 5.3: i) in both cases, the performance has degraded clearly when the MR model is considered and ii) the MR-STMD behaves much better than the MR-TMD, showing a greater robustness, i.e., it is more effective for a broader frequency band.

Figure 5.10: Contour plot of the PI for MR-TMD and MR-STMD with the Bingham MR damper model.

5.3.5.2 Equivalent damping ratio of MR damper in off state

It is interesting to analyse the equivalent damping factor in the off state of the device. For highly non linear system, such as MR dampers, it is possible to derive an equivalent damping [48, 75]. An effective or equivalent viscous damping coefficient for any phenomenon or device can be approximated by the linear one which provides the same energy loss per cycle as the real one in the force-displacement curve, that is, an equivalent viscous damping of the structure could be obtained equalling the experimental energy dissipated in a vibration cycle to the one dissipated by an equivalent viscous system. Figure 5.11 shows two cases of force vs displacement and force-velocity curves at 2 and 6 Hz operating in off state (0 A). The behaviour is close to linear at higher frequencies, so the value of a equivalent damping coefficient c_D can be obtained directly from the force-velocity curve in the case of 6 Hz, but it cannot be done for 2 Hz.

The equivalent c_D value is dependent on frequency and amplitude and it has been calculated for each TMD frequency f_T (simulation and experimental test cases,

Figure 5.11: MR damper experimental results for 2 and 6 Hz in off state (0 A).

Subsection 5.3.1 and Subsection 5.4.2, respectively), $c_{D,\text{freq}}$, using the characteristic amplitude values experienced by the device for each case. From $c_{D,\text{freq}}$, the equivalent damping ratio, $\zeta_{\text{eq},\text{freq}}$, is derived using the following equation:

$$c_{D,\text{freq}} = 2m_T \zeta_{\text{eq},\text{freq}} \omega_T = 4\pi m_T \zeta_{\text{eq},\text{freq}} f_T \quad \rightarrow \quad \zeta_{\text{eq},\text{freq}} = \frac{c_{D,\text{freq}}}{4\pi m_T f_T}.$$

From Figure 5.11a and Figure 5.11c it has derived the $c_{D,\text{freq}}$ coefficient. For 2 Hz, it is obtained from the force-displacement curve, $c_{D,2} = 28.54 \text{ N}\cdot\text{s}/\text{m}$. For 6 Hz, c_D is derived from force-displacement curve, $c_{D,6} = 13.44 \text{ N}\cdot\text{s}/\text{m}$, and from

force-velocity, $c_{D,6} = 13.14 \text{ N}\cdot\text{s/m}$ (verifying the previous value). For the case of 2 Hz, it has been considered the simulated case carried out in the chapter (Subsection 5.3.1), the TMD mass of 10 kg, then $\zeta_{\text{eq},2} = 0.1147 = 11.47\%$. For the case of 6 Hz, it has been considered the experimental test (Subsection 5.4.2) with a TMD mass of 3 kg, then $\zeta_{\text{eq},6} = 0.0549 = 5.49\%$.

5.3.5.3 Optimization process of MR-TMD and MR-STMD

Due to the similar results obtained for the Bingham and Bouc-Wen models (both passive and semi-active), the optimization process is performed only for Bingham cases. The optimization analysis for designing the control parameters is carried out for MR-TMD and MR-STMD with Bingham model. Figure 5.12 shows the PI obtained for both cases. The PI of the MR-TMD is almost the same as that of the TMD (see Figure 5.5a). However, the PI of MR-STMD is worse than the STMD (see Figure 5.5b), so the inclusion of the MR model significantly degrades the effectiveness of the semi-active control.

Figure 5.12: Envelope curve and PI. Optimization for the case in which the Bingham MR damper model is included. The legend shows the PI values for these cases: (---) uncontrolled, (- - -) initial solution and (—) optimized solution.

Table 5.4 shows the initial tuning and that obtained from the optimization process. Interestingly, the tuning frequency for the MR-STMD is over the structure frequency when a broad frequency band is considered. Besides, the optimal minimum current is $I_{\min} = 0 \text{ A}$, which indicates that the frictional force F_c (in Bingham model) of the device is excessive at 0 A and the results would improve significantly for lesser values of F_c .

Table 5.4: Results from the optimization procedure for the MR-TMD and the MR-STMD. The improvement is computed from the initial solution of each device.

Control device	f_T (Hz)	I (A)	PI	Red. (%)	Impr. (%)
Uncontrolled structure	–	–	3.54	–	–
MR-TMD (Asami)	1.98	0.00	1.54	56.50	–
MR-TMD (Optimal)	2.02	0.05	1.40	60.45	↑ 3.95
MR-STMD (Asami)	1.98	0.00	1.18	66.67	–
MR-STMD (Optimal)	2.08	0.00	1.10	68.93	↑ 2.26

Figure 5.13: Description of the experimental test.

5.4 Experimental results

5.4.1 Structure description and experimental setup

The structure is a simply-supported steel beam of 5.10-meter span with UPN-200 cross-section. The structure is excited at mid-span, where the first vibration mode has its maximum sag, by the same actuator used for the MR damper identification setup (see [Figure 5.7](#)). Two piezoelectric accelerometers mounted on a magnet-plate are attached to the beam and the inertial mass, respectively. The controller hardware is a CompactRIO from NI using the LabVIEW real-time module: cRIO-9066 equipped with analog output card NI-9203 and IEPE conditioning card NI-9234 to measure the structure response. In [Figure 5.14c](#), it can be seen the CompactRIO, amplifier and a computer recording the results. To carry out the control law (see [Equation 5.1](#)), the inertial mass velocity has to be estimated from its acceleration. Therefore, the control law programming comprises the velocity estimation, the proper control law and other important elements needed for the practical implementation that will be developed from here.

The nominal case consists of the structure without changes (case 2). Two other cases are studied to evaluate the MR-STMD performance. Case 1 consists of adding four masses of 10 kg close to the mid-span, as shown in [Figure 5.14a](#). Case 3 consists of reducing the span by moving the supports (reducing 20 cm at each side, a total reduction of 40 cm of the span) and removing the masses previously added (see [Figure 5.14b](#)). [Table 5.5](#) shows the modal parameters of the beam for the three cases. Note that the MR-TMD and MR-STMD are tuned to case 2, and their ability to cancel vibration modes with lower and higher frequencies is studied.

(a) General view.

(b) Support view.

(c) Acquisition system and amplifier.

Figure 5.14: Description of the experimental test.

Table 5.5: Modal parameters of the beam for the three cases studied.

	Case 1	Case 2	Case 3
Frequency f_s (Hz)	5.67	6.65	7.67
Damping ratio ζ_s (%)	0.65	0.72	1.53
Modal mass m_s (kg)	115.15	76.87	63.80

5.4.2 Mechanical design of the Tuned Mass Damper

The Asami and Nishihara [12] solution shown in Equation 2.11 serves as the initial estimate of the stiffness for the springs. Therefore, a physical model has been designed using two springs with a given stiffness of 2.5 kN/m each one and the MR damper identified in Subsection 5.3.5. The inertial mass is set at 3 kg (mass ratio of 4% and tuned frequency of 6.49 Hz). A final tuning was carried out from

the normalized spectrum using an excitation chirp from 3 and 8 Hz. This device is studied experimentally, functioning as passive control as well as semi-active control.

5.4.3 Implementation of the control law

The practice implementation of the control law is described in this Section. [Figure 5.15](#) shows the components of the inertial controller feedback (see [Figure 5.1](#)). The control force is $F_C = F_S + F_D$, F_S being the force of the springs and F_D the force of the damper. This control scheme has been implemented using a sampling frequency of 1000 Hz and assigned First-In First-Out conditions.

Figure 5.15: Control scheme for practical implementation of the semi-active control law.

5.4.3.1 Switching-on/off function

A switching-off function has been included to the output of the semi-active law to avoid continuous unnecessary operation of the control device. When the running RMS acceleration computed each second is under a certain value, the control law is off [71]. The switching-off function adopted here is $a_{\text{RMS},1} < 0.05 \text{ m/s}^2$.

5.4.3.2 Low-pass filter

In order to avoid multiple unnecessary blocking/unblocking actions (instabilities coming from the on-off semi-active law operation) due to the noise of the acceleration of the structure at zero crossing, this signal is filtered by a low-pass

Butterworth filter of 4th order with a cut-off frequency of 100 Hz. This frequency should be high enough to prevent phase shifts in the structure acceleration that may spoil the operation of the semi-active control law. Figure 5.16 shows the filtered acceleration for two cut-off frequencies of the low-pass filter. It is shown that the phase shift introduced by the 100 Hz cut-off frequency filter is negligible, whereas the high frequency noise is removed. However, when a filter of cut-off frequency at 20 Hz is considered, the phase shift introduced is not negligible at all, and the semi-active law is then corrupted. Figure 5.17 shows the inertial mass acceleration together with the damper current as a result of applying the semi-active control law. In this figure, it can be seen that the blocking control action is applied uninterruptedly for a short time interval, that is, the damper brakes the inertial mass until tuning phase conditions are achieved.

Figure 5.16: Original and low-filtered acceleration for two cut-off frequencies.

Figure 5.17: Braking the inertial mass due to the blocking force of the MR damper. The gray area represents the damper force turned on.

5.4.3.3 Estimation of the inertial mass velocity

There are different ways to compute the estimated velocity of the inertial mass:

- i) a simple numerical differentiation of the displacement,
- ii) a Kinematic Kalman

Filter (KKF) using the displacement and the acceleration [100, 202], and iii) a Leaky-Integrator Filter (LIF) using only the acceleration [93]. In the first case, differentiating a signal has the drawback that the noise levels can potentially be increased at the output, so this method is directly discarded. Also, it needs to measure the displacement while it is easier to measure accelerations. On the other hand, KKF is a more robust method to estimate the velocity, but it has the disadvantage that both displacement and acceleration must be measured. Finally, an LIF is also a robust procedure for a harmonic signal (as usual for resonance response) and it only requires the acceleration to be measured. In addition, for the on-off control law, only the velocity phase is actually important. The results of KKF and LIF have been experimentally compared, with similar results, making the LIF the most convenient method for the velocity estimation experimentally within the control law. Thus, the following integrator filter is adopted [71]:

$$G(s) = \frac{s}{s^2 + 2\zeta_I \omega_I s + \omega_I^2}, \quad (5.5)$$

in which $s = j\omega$, ω being the angular frequency and ω_I and ζ_I are, respectively, the natural circular frequency and damping ratio of the integrator filter. The parameter $\omega_I = 2\pi f_I$ should be chosen sufficiently smaller than ω_s to avoid significant phase shift at frequencies of interest. The parameters selected were $f_I = 0.5$ Hz and $\zeta_I = 1\%$. Then, $G(s)$ is transformed into the discrete domain using the zero-order hold with a sampling frequency of 1000 Hz, resulting in:

$$G(z) = \frac{0.0009969z - 0.0009969}{z^2 - 1.994z + 0.9937}, \quad (5.6)$$

where z is here the discrete variable. Thus, this filter obtains the estimation of the velocity for frequencies greater than f_I .

5.4.4 Experimental tests

In this case, passive and semi-active vibration control is applied to the cases presented in Subsection 5.4.1. The purpose of these tests is to prove the results obtained numerically for tuning the semi-active device. The damper current applied is 0.11 A for passive and 0.0 A for semi-active, since they have shown to be the optimal values for the nominal case.

Figure 5.18 shows the experimental results obtained for the three cases of Table 5.5. It shows the frequency domain response, magnitude (5.18a, 5.18c, and 5.18e) and phase (5.18b, 5.18d and 5.18f) between the structure acceleration and the excitation force for the structure without control and those controlled with the MR-TMD and with the MR-STMD. With respect to the magnitude, the MR-STMD shows a much better performance when out of the nominal case, although it also behaves better in the nominal case. It is really interesting to look at to the phase plots. The semi-active control law tries to correct the phase to keep

the inertial mass velocity tuned to the structure acceleration (this results in a 90° delay between the structure acceleration and the control force). Note that the phase correction in case 3 is less effective than in case 1. That is, when the frequency ratio $\eta > 1$, the STMD frequency is higher than that of the structure and the semi-active control law is more effective, since it is easier to brake the inertial mass of the control device in this case (as opposed to $\eta < 1$). This result matches the results obtained in Table 5.2 and Table 5.4 from the optimization process for broad frequency-band vibration. It was demonstrated that the STMD frequency should be tuned at higher frequencies of the nominal structure frequency to cope with broad frequency-band vibrations.

Table 5.6 shows the reduction improvements, H_∞ in dB, obtained in the experimental tests. The improvement of the semi-active device with respect to the passive is also shown for each case. It can be seen that, although the semi-active device always improves the performance of the passive one, the greatest difference is obtained for case 1, in which the control device has a greater frequency than the structure. Finally, it should be mentioned that these results have come out as expected, according to the results obtained in the simulations carried out including the MR damper model.

Table 5.6: Experimental vibration reduction (dB) for MR-TMD and MR-STMD for the three cases studied.

Control device	Reduction (dB)		
	Case 1	Case 2	Case 3
MR-TMD	23.94	49.01	8.20
MR-STMD	38.14	52.13	18.02
Improvement	↑ 14.20	↑ 3.12	↑ 9.82

5.5 Conclusions

This chapter presents a methodology for the tuning of STMDs to reduce vibrations in broadband-frequency-varying structures. The methodology is clearly geared to implementation since several practical issues are carefully considered. The methodology is convenient to both structures with an uncertain mode that may change over time or a structure with several close-in-frequency modes.

The following conclusions can be drawn:

- For the sensitivity analysis:
 - The STMD is much less sensitive to structure parameter uncertainties than the TMD, thus showing a better performance for a given frequency band.

Figure 5.18: Experimental vibration results in frequency domain comparing the uncontrolled structure, MR-TMD and MR-TMD for the three cases of [Table 5.5](#). The dashed red line (---) in the phase plots indicates 90° delay between the structure acceleration and the control force, which means perfect tuning.

5. Optimal tuning of semi-active TMD

- Regarding the damper force saturation, for the passive case, the force requirements are merely to dissipate the transferred energy, while the peak force needed is lower. In the semi-active case, higher requirements are needed.
- For the optimization analysis:
 - The STMD should be tuned at a higher frequency with a lower damping ratio than the TMD. That is, the TMD needs more damping to get a certain level of robustness. The higher frequency tuning of the STMD is due to the fact that it is easier for the semi-active controller to decelerate the inertial mass when its frequency is higher than the frequency of the structure.
 - If a model of an existing MR damper is included instead of a linear viscous damper, the performance is clearly degraded, although the STMD remains more robust and effective.
 - Table 5.7 shows a summary of all the results obtained for the different cases jointly: TMD, STMD, MR-TMD and MR-STMD. The aspects mentioned in the previous items are clearly appreciated in this Table.

Table 5.7: Summary of the results obtained numerically for TMD, STMD, MR-TMD and MR-STMD.

Control device	f_T (Hz)	PI	Reduction (%)
Uncontrolled structure	–	3.54	–
TMD	2.02	1.39	60.73
STMD	2.08	0.89	74.86
MR-TMD	2.02	1.40	60.45
MR-STMD	2.08	1.10	68.93

- For practical implementation:
 - To carry out practical implementations, three elements have been added to the semi-active control law: a switching-off function, a low-pass filter and an integrator filter. These elements increase the stability and reliability of the control system.
 - The effect of time delay for the MR damper to apply the semi-active control law can be considered negligible since the structure dynamics are much slower than the MR damper response, which always happen in lively low-frequency structures.
 - The experimental results support the previous numerical results. The phase control degrades faster for structures with a higher frequency than that of the STMD, which is why the tuned frequency of STMD should be higher than the nominal case.

6

Conclusions

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The present chapter summarizes the most relevant conclusions conducted throughout this thesis, followed by suggestions for future research and the publications derived from this thesis.

6.1 Conclusions

This thesis covers from the analysis of the dynamics of pedestrian structures whose modal parameters change over time up to the vibration control using an state-of-the-art strategy based on semi-active control. In between, it has been considered the existence of more than one problematic vibration mode, modal parameter uncertainties, non-linearities in the control design and, finally, a few of interesting conclusions are achieved through the thesis. From the research conducted within the scope of this PhD thesis, according to the goals outlined in [Chapter 1](#) and separately from the particular conclusions of each chapter, the following general conclusions can be drawn:

- A review of VSA under human loading, sources of uncertainty in the modal parameter extraction, vibration control strategies and MR dampers has been carried out in [Chapter 2](#). Issues studied in this chapter serve as sustenance

for later chapters of this thesis. Special care has been taken to mode frequency variability and the implications of that into the tuning and performance of TMDs. Indeed, it has been shown that footbridge may have several vibration modes into the human excitation range, which actually makes the control system design more challenging. Thus, the thesis has focused on STMD as a solution to structures with several problematic modes whose modal properties may indeed change over time. Finally, the thesis has built on STMD with phase control, showing that properly design and conditioned simple control laws can make the STMD succeeded in this challenging problem.

- The variability of the modal parameters that characterise a structure (frequencies, damping ratios and modal shapes) has been analysed in [Chapter 3](#) using three different SSI OMA techniques to estimate the uncertainties associated to the OMA techniques. Moreover, the uncertainty associated to the selected raw data and the time block selected for a particular OMA techniques are studied. The errors in natural frequency estimates using 20 minutes as compared with 60 are negligible. It does not happen with the damping ratio estimates, which experience greater variability.
- A novel methodology for tracking the modal parameters of a structure which may change significantly over time is proposed and tested in [Chapter 3](#) for an in-service stress-ribbon footbridge. Unless natural frequency estimates are well-separated, a sufficient number of spatially located sensors will be needed to associate correctly a particular estimate to its mode shape (through the MAC as the first indicator). The application of this methodology has allowed to analyse and remove the influence of the external factors on the post-process (i.e. temperature). Also an ad-hoc statistical model is developed to remove the influence of external factors on the final estimates. Thus, the frequency estimates might be used within SHM system.
- Several easy-to-implement semi-active strategies for inertial controllers are studied in [Chapter 4](#). They have been compared with other vibration control strategies as passive or active ones under broad frequency-band excitations. Focusing on the semi-active strategies and their implementability within constructable devices, it has been concluded that the on-off semi-active phase control is easy to implement, robust to the MR dynamics and it has shown an excellent capacity to cope with broad band-frequency vibrations.
- A design methodology, through an optimization of the parameters of the semi-active inertial controller, for structures with modal parameter uncertainties and considering a wide frequency range of operation, has been proposed in [Chapter 5](#). In this chapter, the influence of the tuning parameters of the STMD has been studied. Important conclusions have been derived about the tuning of STMDs: 1) very low damping for off stage of the MR is recommendable; 2) the tuning of STMD should be for a higher frequency than the one is usually used (mean frequency) for TMDs designed considering broad frequency-band vibrations; and 3) simple but necessary actions

such as filtering the inertial mass acceleration, switching-off function or low pass filtering should be included into the control law and even in the design methodology.

- Two phenomenological models, Bingham and Bouc-Wen models, have been identified through experimental tests with an MR damper. It has been shown that the non-linear behaviour of MR dampers should be taken into account to obtain more reliable results. It has been demonstrated that the Bingham model (the simplest one) is enough in order to account for the MR non-linearities into the design methodology. This is due to the fact the control is on-off, so the hysteresis does not play an important role into the STMD performance. Obviously, for other control laws, more accurate models might be required.
- The design methodology has been applied and experimentally tested into a lab beam structure in [Chapter 5](#). The different steps for its materialization have been described. Finally, its robustness has been compared to the passive case for two types of frequency detuned cases, above and below detuning. It has been demonstrated the STMD capacity of phase correction keeping the inertial controller tuned for a broad frequency band.

6.2 Perspectives for future work

Firstly, the STMD studied and developed has been tested on a laboratory beam. Efforts should be made in order to get the STMD interesting to be considered in future structures. To achieve that, the following actions have been planned from the research carried out through this thesis:

- A modular TMD will be designed in order to be easily adaptable to a range of structures and also easy to incorporate MR dampers to upgrade it to an STMD.
- Both the TMD and STMD will be tested on a lively in-service footbridge.
- In order to keep the STMD under reasonable cost, a low-cost instrumentation will be included to both, carrying out the semi-active control law and monitor the VS continuously. To this point, an instrumentation based on Raspberry Pi (+ Arduino) and low-cost accelerometers (MEMS) properly conditioned will be considering (see [Figure 6.1](#)).

All these actions are focused on getting affordable STMDs to be incorporated from the design stage or as remedial action to structure which comfort is wanted to be enhanced.

Figure 6.1: Hardware and external cell of the low-cost prototype.

Figure 6.2: MR damper feedback control scheme.

Directly from this research, it has been observed that the friction force of current MR dampers may be too high if one wants to really take advantage of the theoretically achievable STMD performance. In this sense, two interesting possible actions are stated:

- Going into the physical design of current MR dampers and setting clearly the points to act in order to reduce dry friction of these devices.
- Incorporating into the developed design methodology the MR physical properties (using phenomenological models) in order to find the optimal properties to cope with broadfrequency-band vibrations.

Finally, regarding the semi-active action, it is clear that incorporating a feedback action will improve the STMD and may alleviate/compensate the problem of the dry friction, among others such as instabilities due to on-off actions (see Figure 6.2). In this issue, two actions are proposed:

- A broadfrequency-band identification process of MR damper should be developed, giving guidelines to face it. Both displacement and force controlled excitation will be used. Thus, accurate models for broadfrequency-band excitation might be incorporated into the control law and error compensation might even be included. Figure 6.3 shows the assembly experimental setup to carry out this type of test in the future.
- The inclusion of an LVDT and force cell to monitor the MR real-time action. These measures will be used by the feedback controller.

Figure 6.3: Controlled displacement assembly for MR damper identification tests.

6.3 Publications

This section lists the submitted contributions to JCR-indexed journals and conferences, as a result of the research work carried out through this Doctoral Thesis. Publications have been numbered in order of publication (starting with the most recent one).

6.3.1 JCR journal papers

Four journal papers have been published up to the thesis submission in high quality journal in the field of ENGINEERING, CIVIL:

4. **Soria J.M.**, Díaz I.M., García-Palacios J.H. *Further steps towards the tuning of inertial controllers for broadband-frequency-varying structures*. Structural Control and Health Monitoring 2019. doi: <https://doi.org/10.1002/stc.2461>. (ENGINEERING: CIVIL, Q1).
3. Iban N., **Soria J.M.**, Magdaleno A., Casado C., Díaz I.M., Lorenzana A. *Ad-hoc vibration monitoring system for a stress-ribbon footbridge: from design to operation*. Smart Structures and Systems 2018;22:13–25. doi: <https://doi.org/10.12989/sss.2018.22.1.013>. (ENGINEERING: CIVIL, Q1).
2. **Soria J.M.**, Díaz I.M., García-Palacios J.H. *Vibration control of a time-varying modal-parameter footbridge: study of semi-active implementable strategies*. Smart Structures and Systems 2017;20:525–537. doi: <https://doi.org/10.12989/sss.2017.20.5.525>. (ENGINEERING: CIVIL, Q1).

1. **Soria J.M.**, Díaz I.M., García-Palacios J.H., Ibán N. *Vibration Monitoring of a Steel-Plated Stress-Ribbon Footbridge: Uncertainties in the Modal Estimation*. Journal of Bridge Engineering (ASCE) 2016;21:1–13. doi: [https://doi.org/10.1061/\(ASCE\)BE.1943-5592.0000830](https://doi.org/10.1061/(ASCE)BE.1943-5592.0000830). (ENGINEERING: CIVIL, Q2).

6.3.2 Conference proceedings

The communications in conferences directly related to the thesis content are listed here:

18. Magdaleno, A., Casado, C., **Soria J.M.**, Pereira, E., Díaz, I.M., Lorenzana, A. *Comparison of passive and active mitigation devices for vibration control in slender structures*. IX ECCOMAS Thematic Conference on Smart Structures and Materials (SMART 2019), Paris, France: 2019.
17. Renedo C.M., Díaz IM, **Soria J.M.**, García-Palacios J.H. *Designing a footbridge with integrated TMD: improvement assessment*. Congresso de Métodos Numéricos em Engenharia, Guimarães, Portugal: 2019.
16. Barrera C.A., **Soria J.M.**, Wang X., García-Palacios J.H., Díaz I.M. *On the search of TMD configurations for a vibration mode with changing modal properties*. I Conferencia de Dinámica Estructural (DinEst 2018), Madrid, Spain: 2018.
15. **Soria J.M.**, Díaz I.M., García-Palacios J.H., Zanuy C., Wang X. *Tuning a phase-controlled smart TMD for broad-band-frequency-varying vibration modes*. I Conferencia de Dinámica Estructural (DinEst 2018), Madrid, Spain: 2018.
14. Bukovics M., **Soria J.M.**, Díaz I.M., Arroyo J., Calvo J. *Enhancement of vibration pedestrian comfort of a footbridge via Tuned Mass Damper*. I Conferencia de Dinámica Estructural (DinEst 2018), Madrid, Spain: 2018.
13. Casado C.M., Fernández J., Pereda J., **Soria J.M.**, Castaño J. *Design of vibration damping systems using simplified models for lightweight footbridges*. 4th International Conference on Mechanical Models in Structural Engineering (CMMoST 20117), Madrid, Spain: 2017.
12. Marinas-Sanz G., Díaz I.M., **Soria J.M.**, García-Palacios J.H., Zanuy C. *Construcción, análisis dinámico y cancelación de vibraciones de un forjado mixto ligero*. 4th International Conference on Mechanical Models in Structural Engineering (CMMoST 2017), Madrid, Spain: 2017.
11. Díaz I.M., García-Palacios J.H., **Soria J.M.**, Pereira E. *Vibration control strategies for pedestrian structures*. 4th International Conference on Mechanical Models in Structural Engineering (CMMoST 2017), Madrid, Spain: 2017.

10. Canales Sánchez-Castro F., **Soria J.M.**, Díaz I.M., Marinas-Sanz G., García-Palacios J.H. *Identifying a TMD and its final tuning once installed on a lively composite slab*. 4th International Conference on Mechanical Models in Structural Engineering (CMMoST 20117), Madrid, Spain: 2017.
9. **Soria J.M.**, Díaz I.M., García-Palacios J.H., Lorenzana A. *Long-term vibration serviceability assessment of a steel-plated stress-ribbon footbridge*. Footbridge 2017, Berlin, Germany: 2017, p. 384–393.
8. Díaz I.M., García-Palacios J.H., García-Cruz A., **Soria J.M.** *Computing serviceability predictors for an in-service footbridge*. Footbridge 2017, Berlin, Germany: 2017, p. 426–435.
7. **Soria J.M.**, Díaz I.M., García-Palacios J.H., Marinas-Sanz G., Pereira E. *Designing a smart TMD for a composite steel-concrete slender slab*. VIII EC-COMAS Thematic Conference on Smart Structures and Materials (SMART 2017), Madrid: 2017, p. 764–775.
6. **Soria J.M.**, Díaz I.M., Pereira E., García-Palacios J.H., Wang X. *Exploring vibration control strategies for a footbridge with time-varying modal parameters*. MOVIC & RASD 2016, Southampton, England: 2016, p. 118–131.
5. García-Palacios J.H., Tirado-Andrés F., **Soria J.M.**, Díaz I.M., Araujo A. *Effects of time synchronization on operational Modal analysis*. 6th International Operational Modal Analysis Conference, IOMAC 2015, Gijón, Spain: 2015, p. 249–258.
4. **Soria J.M.**, Díaz I.M., García-Palacios J.H., Ibán N., Lorenzana A. *A procedure to track vibration modes under changing external factors: application to a pedestrian bridge*. 3rd International Conference on Mechanical Models in Structural Engineering, CMMoST 2015, Sevilla, Spain: 2015, p. 41–57.
3. Ibán N., Castaño J., **Soria J.M.**, Díaz I.M., Lorenzana A. *Dynamic characterization, modelling and model updating of a lively footbridge*. 3rd International Conference on Mechanical Models in Structural Engineering, CMMoST 2015, Sevilla, Spain: 2015, p. 235–244.
2. García-Palacios J.H., Díaz I.M., **Soria J.M.**, Cara F.J., García S. *Uncertainty optimization of Automated Operational Modal Analysis applied to Structural Health Monitoring techniques*. 6th World Conference on Structural Control and Monitoring, Barcelona: 2014, p. 1420–1427.
1. Díaz I.M., **Soria J.M.**, Ibán N., Cacho M., García-Palacios J.H. *Vibration monitoring of a steel-plated stress-ribbon footbridge: effects of environmental and operational factors on modal properties*. 6th World Conference on Structural Control and Monitoring, Barcelona: 2014, p. 457–466.

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